Pumping Station Design
Revised Third Edition

Editor-in-Chief

GARR M. JONES, PE

Senior Vice President, Design
Brown and Caldwell Consultants
Walnut Creek, California

Co-Editors

ROBERT L. SANKS, Ph.D., PE

Consulting Engineer and Professor Emeritus
Montana State University
Bozeman, Montana

GEORGE TCHOBANOGLOUS, Ph.D., PE

Consulting Engineer and Professor Emeritus
University of California at Davis
Davis, California

BAYARD E. BOSSERMAN, II, PE

Engineering Consultant
Mission Viejo, California

AMSTERDAM • BOSTON • HEIDELBERG • LONDON • NEW YORK • OXFORD
PARIS • SAN DIEGO • SAN FRANCISCO • SINGAPORE • SYDNEY • TOKYO
Butterworth-Heinemann is an imprint of Elsevier

ELSEVIER

Butterworth–Heinemann is an imprint of Elsevier
30 Corporate Drive, Suite 400, Burlington, MA 01803, USA
Linacre House, Jordan Hill, Oxford OX2 8DP, UK

Library of Congress Cataloging-in-Publication Data
Pumping station design/editor-in-chief, Garr M. Jones; co-editors, Bayard E. Bosserman, Robert L. Sanks, George Tchobanoglous. – Rev. 3rd ed.
p. cm.
Includes bibliographical references and index.
ISBN 978-1-85617-513-5 (alk. paper)
1. Pumping stations–Design and construction. 2. Water treatment plants. 3. Sewage disposal plants. I. Jones, Garr M. II. Bosserman, Bayard E. III. Sanks, Robert L. IV. Tchobanoglous, George.
TD485.P86 2008
628.1'44–dc22 2008019632

British Library Cataloguing-in-Publication Data
A catalogue record for this book is available from the British Library.

For information on all Butterworth-Heinemann publications visit our Web site at www.elsevierdirect.com

Printed in the United States of America
08 09 10 11 12 10 9 8 7 6 5 4 3 2 1

Preface to the First Edition

This book, *Pumping Station Design*, is unique in the following ways. It was written by consultants for consultants so as to be of the greatest practical use for designers. Each author is an expert whose writing is based mostly on personal experience. Little of it was obtained from, or based on, the existing literature. To make the book more usable and understandable, over 370 illustrations are included together with 59 design examples. Most design examples and all formulas are given in both SI and U.S. customary units. The book is complete enough for the novice and advanced enough to be useful to experienced designers and to those who direct or may be associated with design (such as utility managers, city engineers, or equipment suppliers). It is the only text available that deals comprehensively with the entire subject of how to design pumping stations. Finally, the book is unique in the number and expertise of its authors and contributors and in the meticulous care exercised during the seven years of its preparation (as described in the following paragraphs) to make it as easy to read as possible.

The first eleven chapters contain the fundamentals essential for effective design and include hydraulics, piping, water hammer, electricity, and theory and descriptions of pumps. The middle third is devoted to system design, including pump and driver selection and general piping layouts for water, wastewater, and sludge pumping. The last ten chapters contain supporting disciplines and subjects such as instrumentation and design, heating and ventilating, noise and vibration, comparisons of types of pumping stations and pumps, blunder avoidance, contract documents, detailing, and cost analyses. The appendices contain useful physical data, lists of codes and specifications, design checks, start-up checks, and addresses of all publishers given in the references. The tables of flow and headloss in pipes are compiled in a useful form not heretofore published. All of the work is extensively cross-referenced.

Perhaps never before has such a large, talented group of professionals been gathered to produce a book. The 132 expert contributors to this text provide broad and encompassing viewpoints gained from an aggregate of 20 centuries of practical experience. Each author was selected on the basis of specialized knowledge, past performance, experience, and commitment to the profession. Each produced one or more chapters (or parts thereof) based on detailed outlines suggested by the editorial board and improved by author and board as the rest of the book was developed. The other contributors, also selected on the basis of experience and competence, helped in the peer reviews and by supplying information.

Typically, I rewrote (or at least heavily edited) each chapter to conform to a uniform style and then sent it to from three to seven peer reviewers whose collected comments would be rephrased and given to the author with my own comments added. Following the author's reply, a second rough draft would be prepared and sent to author and reviewers. The returned comments would be recast into a third draft and again sent to the author. The fourth draft, usually called "final draft one," was sent to the co-editors. George Tchobanoglous checked every chapter for construction, clarity, and style. Garr M. Jones checked every chapter for practicality and good design practice. The other co-editors reviewed selected chapters for completeness and accuracy. Improvements, integration with other chapters, and nuances of wording often required as many as four subsequent "final drafts" until the chapter satisfied author, reviewers, and editors—a process that has taken seven years. As the book neared completion, new material was added and various subjects were

sometimes shuffled between chapters for more logical presentation and cross-referencing. Alterations and improvements were continued through February 1989. Some idea of the effort taken can be appreciated by realizing that over 50,000 pages of review drafts have been distilled into this book. The result is considered to represent the state of the art (as of early 1989)—practical, authoritative, and essentially timeless. Consulting firms will find that this book can sharply reduce the time for an inexperienced engineer to become a competent pumping station designer. Project leaders will find the comprehensiveness, the checklists, and the list of blunders to be of great help. Utility managers will discover that selective reading of a few chapters will provide insights for directives that can produce better pumping stations for lower overall costs of construction, maintenance, and repair.

The work on this book was begun with a conference on pumping station design and a detailed proceedings outline, which served as a first approximation for the textbook to follow. Proceedings authors were selected on the basis of their experience records and were assigned chapters (or sections thereof) in strict adherence to the outline. The resulting *Proceedings,* published in 1981 in 4 volumes (1576 pages), are still available and valuable as an adjunct tools for design *[out of print in 1996 but still available through interlibrary loan from Montana State University—Ed.]*. Although the purpose of the conference was to make this new material immediately available to the profession, it also enabled us to find a group of experts and to gather resources for this book.

What prompted this project was the lack of a complete textbook about pumping station design in the United States (or in the English language insofar as we knew.) Of course, there were many books about pumps and pumping machinery and a few short manuals for designing pumping stations but there was no comprehensive, authoritative text or reference book dealing specifically with the design of all phases of water and wastewater pumping stations. Indeed, the literature about pumping station design has been fragmented, often superficial, sometimes wrong, and generally incomplete. One expert stated that 95 percent of all pumping stations he has seen contain serious design mistakes and that they occur in every category; if so there was a need for a book written by practicing engineers for consultants and others involved in decision making. Knowledge about the subject has been largely confined to consulting engineers, a few large public utilities and to equipment manufactures, so the overall purpose of this project was to gather, codify, and preserve the knowledge (much of which has never been printed) for the benefit of the public and the profession.

Carl W. Reh was the first co-editor appointed and, until his death in 1983, my chief proponent and supporter. The other co-editors, George Tchobanoglous, Donald Newton, B. E. Bosserman II, and Garr M. Jones (in order of appointment) have made this work possible. As technical advisor, Earle C. Smith provided much invaluable guidance and critiqued a large part of the work. All the authors and contributors have given a great deal of time to the project with no thought of reward beyond a desire to be of service to the profession.

Several consulting firms made extraordinary contributions of time, effort, and finances to the project, as follows: Greeley and Hansen Engineers, Chicago—six authors, including one editor, wrote four chapters, a part of another, and two appendices; Brown and Caldwell Consultants, Walnut Creek, California—three authors, including one editor, wrote six chapters and one appendix; Boyle Engineering Corporation, Newport Beach and Bakersfield, California—two authors, including one editor, produced five chapters and one appendix. Several firms, listed in Chapter 29, contributed cost data, an onerous task. Sincere appreciation is extended to all for this help, and, indeed, the engineering profession is indebted to all the contributing firms and personnel.

Mary C. Sanks patiently typed draft after draft and checked grammar, readability, punctuation, and spelling, and she assisted with galley and page proofs. Edimir Rocumback, student in architecture, drafted most of the figures. The entire project was made possible by the financial support of Montana State University. Officers directly involved included Theodore T. Williams, formerly Head, Department of Civil Engineering and Engineering Mechanics; Byron J. Bennett, formerly Dean, College of Engineering; and Lawrence T. Kain, formerly Administrator of Grants and Contracts.

Robert L. Sanks
Bozeman, Montana
March 1989

Contents

Chapter 4
Piping 4.1

BAYARD E. BOSSERMAN II, JAMES C. DOWELL, ELIZABETH M. HUNING, AND ROBERT L. SANKS

Chapter 5
Valves 5.1

CARL N. ANDERSON, BAYARD E. BOSSERMAN II, AND CHARLES D. MORRIS

Chapter 6
Fundamentals of Hydraulic Transients 6.1

BAYARD E. BOSSERMAN II AND WILLIAM A. HUNT

Chapter 7
Control of Hydraulic Transients 7.1

BAYARD E. BOSSERMAN II

Chapter 8
Electrical Fundamentals and Power System Principles 8.1

STANLEY S. HONG, PHILIP A. HUFF, AND PAUL C. LEACH

Chapter 9
Electrical Design 9.1

STANLEY S. HONG, PHILIP A. HUFF, AND PAUL C. LEACH

Chapter 19 System Design for Sludge Pumping 19.1

CARL N. ANDERSON AND DAVID J. HANNA

Chapter 20 Instrumentation and Control Devices 20.1

ROBERT S. BENFELL AND ALAN VAUSE

Chapter 21 Instrumentation and Control Applications 21.1

ROBERT S. BENFELL

Chapter 22 Vibration and Noise 22.1

JERRY G. LILLY AND WILLIAM D. MARSCHER

Chapter 23 Heating, Ventilating, and Cooling 23.1

EARL L. HECKMAN, PERRY L. SCHAFER, AND PHILIP WOLSTENHOLME

Chapter 24 Designing for Easy Operation and Low Maintenance 24.1

BAYARD E. BOSSERMAN II, GEORGE JORGENSEN, GARY ISAAC, AND ROBERT L. SANKS

Chapter 25 Summary of Design Considerations 25.1

GARR M. JONES, ROBERT A. ''RANDY'' NIXON, RANDALL R. PARKS, AND ROBERT L. SANKS

Chapter 26 Pumping Station Design Examples 26.1

GARR M. JONES, GARY S. DODSON, AND THEODORE B. WHITON

Chapter 27 Avoiding Blunders 27.1

ROBERT L. SANKS

Chapter 28 Contract Documents 28.1

JOHN E. CONNELL AND THOMAS M. FLEGAL

Chapter 29
Costs 29.1

STEFAN M. ABELIN, MARC T. PRITCHARD, AND ROBERT L. SANKS

Appendix A
Physical Data A.1

Appendix B
Data for Flow in Pipes, Fittings, and Valves B.1

ROBERT L. SANKS

Appendix C
Typical Specifications for Pumps and Drivers C.1

DAVID L. ELSENHAUER, THOMAS M. FLEGAL, AND GARR M. JONES

Appendix D
Common Blunders D.1

EARLE C. SMITH

Appendix E
Checklist for Project Reviews E.1

LEROY R. TAYLOR

Appendix F
Start-Up F.1

ERIK B. FISKE, GEORGE FRYE, LOWELL G. SLOAN, AND SAM V. SUIGUSSAAR

Appendix G
Suction Specific Speed G.1

GARR M. JONES

Index I.1

Chapter 2

Nomenclature

MAYO GOTTLIEBSON
CARL W. REH
ROBERT L. SANKS

CONTRIBUTORS
Robert S. Benfell
Earl L. Heckman
Stanley S. Hong

2-1. Abbreviations

The abbreviations given here are the same for both singular and plural. The style follows common American usage. Abbreviations for some special disciplines are given near the beginning of the chapter about that subject.

A	Ampere (also amp).
A	Area, usually the cross-sectional area of a pipe.
a	Acceleration.
a	Acoustic wave speed.
@	At.
abs	Absolute.
ac	Alternating current.
AC/h	Air changes per hour.
acre-ft	Acre-foot (43,560 ft^3).
AF	Adjustable frequency.
AFD	Adjustable-frequency drive (for electric motors).
AM	Ante meridian (morning).
amp	Ampere (also A).
amp-h	Ampere-hour.
AN	As noted.
AOR	Allowable operating region of a pump's H-Q curve.
API	American Petroleum Institute.
ASHRAE	American Society of Heating Refrigerating and Air Conditioning Engineers.
atm	Atmosphere (= 14.7 lb/in.^{2}).
avg	Average.
AWG	American wire gauge.

AWWA	American Water Works Association.
B	Bulk modulus of elasticity of water.
BEC	Best efficiency capacity. Flow rate at BEP.
BEP	Best efficiency point.
BHN	Brinell hardness number (on the C-scale except as otherwise noted).
bhp	Brake horsepower.
bhp-h	Brake horsepower-hour.
BHRA	British Hydromechanics Research Association.
bkW	Brake kilowatts.
Btu	British thermal unit (the heat required to raise the temperature of 1 lb of water by 1°F).
c	One hundred.
©	Copyrighted.
C	Hazen–Williams roughness coefficient.
°C	Degree Celsius (centigrade) equals (°F – 32)5/9.
c–c	Center to center.
C_L–C_L	Centerline to centerline.
CADD	Computer-aided drafting and design.
CFD	Computational fluid dynamics. A numerical means to predict flow patterns based on solving the governing equations in three dimensions at discrete nodes that represent the flow space. See Section 3-13.
cfm	Cubic feet per minute (also ft^3/min).
cfs	See ft^3/s.
ckt bkr	Circuit breaker.
cm	Centimeter.
cm^2	Square centimeter.
cm^3	Cubic centimeter.
CMOM	Capacity, Management, Operation, Maintenance—acronym for regulations intended to become part of the Clean Water Act.
coef	Coefficient.
conc	Concentration.
cond	Conductivity.
const	Constant.
COP	Coefficient of performance (a ratio of electrical energy input to heat energy output).
CPRV	Chlorine pressure reducing valve.
C/S	Constant speed.
CSO	Combined (wastewater and storm water) sewer overflow.
CU	Coefficient of utilization used in lighting calculations.
cu ft	See ft^3.
cu in.	See $in.^3$.
cu yd	See yd^3.
cwt	One hundred pounds.
d	Day.
D	Diameter.
dB	Decibel.
dBA	Decibel-A scale.
dc	Direct current.
dia	Diameter.
DIP	Ductile iron pipe.
Do	Ditto.
DOL	Direct on line (method of starting electric motors).
E	Exponent (e.g., 3.6 E–6 = 3.6×10^{-6}).
E	Modulus of elasticity (see Table A-10, Appendix A).
E	Efficiency (sometimes followed by a subscript to indicate pump, motor, etc.).
ea	Each.
ed	Edition.

Ed.	Editor.
eff	Efficiency.
e.g.	For example.
EGL	Energy grade line.
El	Elevation (also Elev.).
emf	Electromotive force.
ENRCCI	Engineering News-Record Construction Cost Index.
E_p	Pump efficiency.
EPA	See U.S. EPA.
Eq	Equation.
est	Estimated.
et al.	And others.
ETM	Elapsed time meter.
$\boldsymbol{F}$	Froude number.
°F	Degrees Fahrenheit.
ff	Following pages.
FFT	Fast Fourier transform (analyzer) for vibration analysis.
FLA	Full load amperes.
FLI	Full-load electrical current (see also I_{FL}).
FM	Force main.
fob	Freight on board.
fpm	See ft/min.
fps	See ft/s.
FRP	Fiberglass-reinforced plastic.
FS	Factor of safety.
FSK	Frequency shift modulation.
ft	Foot or feet.
ft^2	Square feet.
ft^3	Cubic feet.
ft · cd	Foot-candles.
ft · lb	Foot-pound.
ft/min	Feet per minute.
ft/s	Feet per second.
ft^3/s	Cubic feet per second.
FVNR	Full voltage (nonreversing for magnetic motor starters).
g	Acceleration due to gravity ($= 9.81\ m/s^2 = 32.17\ ft/s^2 = 386\ in./s^2$).
g	Gram.
g_c	$32.17\ ft \cdot lb_m/lb_f \cdot s^2$.
ga	Gauge.
gal	Gallon.
gal/d · cap	Gallons per day · capita.
gal/min	Gallons per minute.
GFCI	Ground fault circuit interrupter (a safety device).
GFI	See GFCI.
gpcpd	Gallons per capita per day (also gpcd; see gal/d · cap).
gpm or GPM	See gal/min.
Grd	Ground.
GTO	Gate turn-off thyristor.
h	Hour.
H	Hedstrom number. See Equation 19-5.
H or h	Head [usually in meters (feet) of water].
h_L	Headloss.
HDPE	High-density polyethylene (pipe).
HGL	Hydraulic grade line.
HOA	Hand-off-automatic (switch).
hp	Horsepower.

hp · h	Horsepower-hour.
H-Q	Head-capacity.
hr	Hour (see h).
HV	Heating and ventilating (also H&V).
HVAC	Heating, ventilating, and air conditioning (also HV&AC).
H–W	Hazen–Williams (Equations 3-8 and 3-9).
HWL	High water level.
Hz	Hertz (cycles per second).
I	Electrical current.
I_{FL}	Full-load current (amperes).
I_L	Line current (amperes).
I_θ	Phase current (amperes).
I&C	Instrumentation and control.
ID	Inside diameter [mm (in.)].
i.e.	That is.
IEEE	Institute of Electrical and Electronics Engineers.
IGBT	Insulated-gate bipolar transistor (sometimes found in AF drives).
I/I	Infiltration and inflow into sewers.
Imp gal	Imperial gallon.
in.	Inch.
in.^{2}	Square inches.
in. · lb	Inch-pound.
IR	Indicating and recording (instrument).
ISA	Instrument Society of America.
J	Joule.
J	Polar moment of inertia.
K	Bulk modulus of elasticity of water (B in Chapter 22).
K	Torsional rigidity (in Chapter 22).
K	Degree Kelvin (= °C + 273.2).
kc	Kilocycles.
kcal	Kilocalorie.
kcmil	A unit of conductor area equal to the diameter of a wire in mils squared.
kg	Kilogram.
kip	Kilopound (1000 lb).
km	Kilometer.
kPa	Kilopascals (1000 N/m^2).
ksi	Kilopounds (1000 lb) per square inch.
kV	Kilovolt.
kVA	Kilovolt ampere.
kVAR	Reactive kilovolt ampere.
kW	Kilowatt (1000 watts).
kW · h	Kilowatt-hour.
L	Liter.
lb	Pound (means lb_f in this volume).
lb_f	Pound force = $lb_m\ g/g_c$; numerically (not dimensionally) equals lb_m. In this volume lb_f is written lb.
lb_m	Pound mass (= $lb_f \cdot g_c/g$).
lb/ft^2	Pounds per square foot.
lb/ft^3	Pounds per cubic foot.
lb/in.^{2}	Pounds per square inch.
lb/in.^{2} abs	Pounds per square inch absolute (above vacuum).
lb/in.^{2} ga	Pounds per square inch gauge (above atmospheric pressure).
L/d · cap	Liters per day · capita.
LEL	Lower explosion limit.
LF	Linear feet (also lin ft).
ln	Natural logarithm.

log	Logarithm to base 10.
LPG	Liquid petroleum gas (either propane or butane; specify which).
LWL	Low water level.
m	Meter(s).
m^2	Square meters.
m^3	Cubic meters.
mA	Milliampere.
max	Maximum.
MCC	Motor control center.
MCM	Now kcmil. A unit of conductor area equal to the diameter of a wire in mils squared.
MCP	Motor circuit protector.
Mfr	Manufacturer.
mg	Milligram.
Mgal	Million gallons.
Mgal/d	Million gallons per day.
mgd or MGD	See Mgal/d.
mg/L	Milligrams per liter.
mi	Mile.
mi/h	Miles per hour.
mil	1/1000 in. or 0.025 mm or 25 microns.
min	Minimum.
min	Minute.
ml	Milliliter.
mm	Millimeter.
mol	Mole.
mph	See mi/h.
mV	Millivolts.
N	Newton (the force required to accelerate 1 kg of mass $1\,m/s^2$).
n	Manning roughness coefficient (Equation 3-17).
NA	Not applicable.
NC	Normally closed (electrical switch).
n.d.	No date.
NEC	National Electrical Code.
NEMA	National Electrical Manufacturers Association.
NO	Normally open (electrical switch).
No.	Number.
NPS	Nominal pipe size.
NPSH	Net positive suction head (q.v. Section 2-2).
NPSHA	Net positive suction head available (q.v. Section 2-2).
NPSHR	Net positive suction head required (q.v. Section 2-2).
NRS	Nonrising stem (for valves).
N_s	Specific speed in U. S. customary units of rev/min, gal/min, and ft.
n_s	Specific speed without regard to units.
n_q	Specific speed in SI units of rev/min, m^3/s, and m.
NTU	Nephelometric turbidity units.
OC	Overcurrent.
OD	Outside diameter.
O&M	Operation and maintenance.
OSHA	Occupational Safety and Health Administration.
OS&Y	Outside screw and yoke (indicates a rising stem in a valve).
oz	Ounce.
p	Pole (electrical).
p, pp	Page, pages.
Pa	Pascal (N/m^2).
PAM	Pulse amplitude modulated.

pcf	See lb/ft^3.
Pf	Power factor.
P&ID	Process and instrumentation diagram.
PIR	Peak influent rate.
PLC	Programmable logic controller.
PM	Post meridian (afternoon).
POR	Preferred operating region of a pump's H-Q curve.
ppb	Parts per billion.
ppm	Parts per million.
PRV	Pressure reducing valve.
PS	Pumping station.
psi	See $lb/in.^2$.
psia	See $lb/in.^2$ abs.
psig	See $lb/in.^2$ ga.
PTC	Positive temperature coefficient thermistor.
PVC	Polyvinyl chloride.
PWM	Pulse width modulated.
Q	Flow rate, capacity, or discharge.
q.v.	Which see.
R	Degrees Rankine (= °F + 491.7).
R	Electrical resistance (in ohms).
R	Hydraulic radius (wetted area/wetted perimeter).
$\boldsymbol{R}$	Reynolds number.
®	Registered trademark (see also ™).
R_m	Hydraulic radius modified by Escritt's assumption.
rad	Radian.
RCP	Reinforced concrete pipe.
RCR	Room cavity ratio used in lighting calculations.
rev/min	Revolutions per minute.
rev/s	Revolutions per second.
RMPDR	Recommended minimum pump discharge rate.
RMS	Root mean square (an average of 70.7% of peak value in a sinusoidal waveform).
rpm or RPM	See rev/min.
rps or RPS	See rev/s.
RS	Raw sewage (the older term, "sewage," is now replaced by "wastewater").
RTP	Reinforced thermosetting polyester (the modern acronym for fiberglass or FRP).
RVNR	Reduced voltage nonreversing (for magnetic motor starters).
RW	Raw wastewater.
s	Second.
SCADA	Supervisory control and data acquisition.
scfm	Standard cubic feet per minute.
Sch	Schedule (for pipe).
SCR	Silicon controlled rectifier.
SDR	Average outside diameter of a pipe divided by minimum wall thickness.
sec	Second (see s).
SF	Service factor.
shp	Shaft horsepower.
SI	Systéme International (for metric units).
sp gr	Specific gravity.
sq	Square.
sq ft	See ft^2.
sq in.	See $in.^2$.
ss	Stainless steel.
SSO	Sanitary sewer overflow.
SSRV	Solid state reduced voltage (for motor starters).
Std	Standard.

t	Thickness.
t	Time.
TDH	Total dynamic head (= static head + friction head + other losses).
TDS	Total dissolved solids.
TEFC	Totally enclosed, fan cooled (motor).
temp	Temperature.
TIR	Totalizing, indicating, and recording (instrument).
™	As a superscript, trademark (see also ®).
Typ	Typical, typically.
UL	Underwriter's Laboratory.
URV	Upper range value.
U.S.	United States or U.S. customary units.
U.S. EPA	United States Environmental Protection Agency.
V	Volts.
V	Volume.
V_{L}	Line-to-line voltage.
V_{θ}	Phase voltage.
v	Velocity.
VA	Volt-ampere.
VAR	Reactive volt ampere.
VCP	Vitrified clay pipe.
VFD	Variable-frequency drive [also AFD (adjustable-frequency drive)].
viz.	Namely.
V/S	Variable speed.
vs.	Versus.
VTSH	Vertical turbine, solids-handling (pump).
W	Watt.
WC	Water column [pressure in meters (or feet) of water].
WEF	Water Environment Federation.
W · h	Watt-hour.
W · hM	Watt hour meter.
whp	Water horsepower.
wkW	Water kilowatts.
WPCF	Water Pollution Control Federation (now Water Environment Federation).
WR^2	Inertia of moving parts in pump and driver.
wt	Weight.
XS	Extra strong (pipe).
XXS	Double extra strong (pipe).
yd	Yard.
yd^2	Square yards.
yd^3	Cubic yards.
yr	Year.
γ	Specific weight.
ε	Absolute pipe roughness (inches or millimeters).
μ	Dynamic viscosity.
μ	Poisson's ratio.
ν	Kinematic viscosity.
ρ	Density.
Σ	Sum, summation.
τ	Shearing stress.
ϕ	Phase (electrical current).
ω	Rotational speed (usually rev/min).
Ω	Ohms (resistance).
$\geq$	Equal to or greater than.
$\leq$	Equal to or less than.
$>$	Greater than.

$<$	Less than.
$\approx$ or $\cong$	Approximately equals.
$\approx$ or $\sim$	Approximately (prefix).
$\pm$	Approximately (suffix).
¢	Cent.
\$	Dollar.
$'$	As a superscript, feet.
$''$	As a superscript, inches.
%	Percent.
$\cdot$ or $\times$	Times (multiplication).

2-2. Definitions

The following list of terms is not intended to be complete but to contain only commonly used terms that might be unfamiliar to some readers. Some terms for special disciplines are contained at the beginning of the chapter about that subject.

Absolute pressure: Gauge pressure plus standard atmospheric pressure.

Affinity laws: A set of equations or laws governing the operation (discharge, head, power) of a single centrifugal pump operated at different speeds or, by close approximation, governing the relationship of operation of pumps or impellers of different sizes operated under dynamically similar conditions (see Section 10-3).

Air break: An air gap in a liquid system intended to ensure against back siphonage. The discharge into free atmosphere must be located well above the highest feasible water level. Air breaks are often required for positive protection of potable water.

Ampacity: Current-carrying capacity in amperes.

Approach pipe: A pipe laid at a slope of 2%± from high to low water level to avoid a free fall of entering water and to add storage capacity to a wet well with constant speed pumps.

Archimedes screw: See helical screw pump.

Axial flow pump: A pump in which the impeller (somewhat similar to a motorboat propeller) moves the fluid parallel to the pump shaft.

Backflow preventer: A device installed in potable water piping to prevent the reverse flow of non-potable water back into a potable system.

Background: The ambient condition before a change is made, such as the apparent concentration of a substance before the substance is added as a tracer.

Back head (also stuffing box head): The portion of the pump casing attached to the volute that houses the stuffing box and bearings on an end-suction pump.

Barrel (can) pump: A pump inside a pipe.

Best efficiency point: The discharge rate at which an impeller of a given diameter rotating at a given speed operates at maximum efficiency.

Bingham plastic: A material in which flow occurs in a straight-line relationship with shear stress, but only after a certain amount of shear stress is overcome.

Blank: A solution containing all the ingredients except the substance to be measured. Also the instrumental reading obtained from the foregoing.

Booster: A pump that takes suction from a pressurized piping system and discharges, at a higher pressure, to a second, isolated piping system.

Boss: Projection on a pipe fitting to provide extra thickness for a smaller threaded pipe.

Bowl: The casing that contains the impeller of a vertical turbine or vertical axial flow pump.

Brake horsepower: The power delivered to the pump shaft.

Bump: Momentary energization of a motor that causes it to rotate a few revolutions.

Bus: A bar (usually copper) that serves as the common connection for three or more electrical circuits.

Caisson: A heavy, thick-walled enclosure built in stages (or lifts) above ground and sunk into place by its weight and by excavating the interior up to and underneath the cutting edge.

Casing: The impeller housing that directs the fluid into the impeller and thence to the discharge flange (see also scroll).

Cavitation: Vapor bubbles formed on a solid surface (often an impeller) in contact with the liquid. The vapor bubbles occur when the pressure in the liquid falls below the vapor pressure.

Centrifugal pump: A rotodynamic pump in which the fluid is displaced radially by the impeller. Colloquial: Any rotodynamic pump in which the fluid is displaced radially, axially, or by a combination of both.

Characteristic curve: A plot of head versus flow rate for a pump.

Clean water: See clear water.

Clear water: Water that neither forms scum nor deposits under stagnant conditions.

Clear well: A basin of filtered water in a water treatment plant.

Close coupled: A motor mounted close to the pump and in which the pump impeller is mounted on a specially lengthened motor shaft.

Closed impeller: An impeller with vanes enclosed by shrouds on both sides.

Closed system: A liquid piping system that does not include storage open to the atmosphere.

Coating: A liquid that cures after application to a surface. Example: paint (see Lining).

Column: The vertical piping through which a suspended pump discharges.

Comminutor: A machine designed to cut and shred stringy materials and large solids. It is sometimes installed before the wet well of a wastewater pumping station to prevent clogging of the pumps.

Condition point: A point on the pump characteristic curve at which the pump delivers a given value of flow at a given head.

Conduit fill: The amount of wires allowed in a conduit. The allowable is 40% of the internal cross-sectional area.

Confined space: A space large enough and so configured that a person can bodily enter and perform assigned work; may have limited or restricted means for entry or exit [e.g., tanks, vessels, silos, storage bins, hoppers, vaults, pipes, trenches greater than 1.2 m (4 ft) deep, and not designed for continuous employee occupancy]. (See permit-required confined space.)

Control room: The superstructure of a pumping station, which usually includes electrical equipment and may also include motors for extended shaft or suspended pumps.

Corresponding point: One of two or more points related to one another by all of the affinity laws applied simultaneously.

Critical depth, flow, or velocity: The depth, flow, or velocity for which specific energy is the minimum.

Cutter pump: A centrifugal pump with a cutting blade (attached to the impeller) designed to cut stringy materials into short segments before entrance into the impeller eye.

Cutwater: The leading edge of the volute wall that separates the volute from the discharge nozzle. Minimum clearance between impeller and volute occurs at the cutwater.

Dashpot: A device in which a fluid escapement decreases the speed of closure (as of a valve disc).

Demand: A schedule of water requirements.

Developed head: The difference between the suction head and discharge head of an operating pump.

Diffusers: Stationary vanes (within the casing) to provide gradually expanding paths to change the direction of flow and convert liquid velocity into pressure.

Dirty water: Water that contains solids such as silt or sand.

Discharge flange: The flange at which liquid is expelled from the pump.

Discharge head: Static head plus friction headloss (in discharge piping) plus velocity headloss at the end of the discharge piping.

Discharge manifold: The piece or section of pipe to which the individual discharge pipes from all the pumps are connected.

Double suction: A centrifugal water pump in which the liquid enters the impeller through eyes on both sides and thereby reduces the force on thrust bearings.

Downstream: In the direction of flow.

Drawdown: The vertical distance over which the surface of the pumped liquid is lowered during the pumping cycle.

Dresser® coupling: See sleeve coupling.

Dry pit: See dry well.

Dry well or dry pit: The below-grade structure of a pumping station that contains the pumps, drive shafts, valves, and piping and in which there is no liquid outside the pumps and piping (i.e., the structure is "dry" and workers often occupy the space).

Dry well pump: A pump designed to be mounted in a dry well and requiring a flooded suction pipe.

Ductile iron: A cast iron alloyed with magnesium and heat treated (to concentrate the carbon into nodules) to obtain ductility and high tensile strength. Uncontrolled high heating makes ductile iron revert to brittle gray cast iron.

Duty: Utilization and specific application of equipment, such as operation or emergency.

Duty cycle: Time-based utilization and specific application of equipment.

Effluent: Liquid discharged from a pumping station.

Escritt assumption: The addition of half the width of the free water surface to the wetted perimeter to obtain a modified hydraulic radius, Rm, for improving the accuracy of the Manning equation.

Extended shaft: Vertical pump and motor separately supported and connected by means of open shafting.

Fillet: Concrete in the bottom of the wet well shaped to smooth liquid flow into the pump suction openings and to prevent the accumulation of solids.

Finished water: Potable, treated water.

Firm pumping capacity: Capacity of the pumping station with the largest pump out of service or on standby.

Fix: A device or correction that prevents the occurrence of an undesirable condition.

Flexible coupling: A device to couple two shafts at a small angle.

Flexible pipe coupling: A coupling often used at walls, and sometimes elsewhere, to couple two slightly misaligned pipes.

Flooded suction: Condition in which the pump volute is below the low water level of the wet well.

Foot-candle: The amount of light falling on 1 sq ft of a spherical surface at a distance of 1 foot from a standard candle.

Force main: Piping, external to the station and filled with liquid under pressure, through which the station discharges.

Frazil ice: Crystals of ice formed in turbulent waters.

French drain: A rock-filled pit or trench designed to allow water to seep into surrounding soil.

Friction head: Equivalent head necessary to overcome friction losses through the piping, fittings, and valves.

Gauge pressure: Pressure above standard atmospheric pressure.

Gen-set: Engine generator set.

Gray iron: A brittle cast material principally of iron with 2 to 4% carbon and 1% silicon by weight and an ultimate tensile strength of 140,000 to 410,000 kPa (20 to 60 ksi), compressive strength 3 to 5 times greater than tensile strength, with negligible plastic deformation.

Grinder pump: A centrifugal pump with grinder assembly on an extension of the shaft, designed to grind solids larger than a fraction of an inch before they contact the recessed impeller.

Grooved-end pipe coupling: A pipe coupling used to make pipe fitting and disassembly easier, often called a Victaulic® coupling (a trademark name).

Groundwater: Raw water obtained from wells.

Hazardous location: Defined in Article 500 of National Electric Code (see also Section 25-3).

Head: Pressure expressed in terms of the height of a column of water in meters or feet.

Headloss: The drop in pressure due only to friction and turbulence as fluid flows through a section of conduit or piping. It is expressed in terms of an equivalent loss of height of a water column in meters or feet.

Heat tracing: An electric heat tape wrapped around a pipe or tube (usually beneath insulation) to prevent freezing.

Hedstrom's number: A dimensionless number defined by Equation 19-5.

Helical screw pump (also Archimedes screw): A positive displacement, inclined, semi- or fully enclosed screw for lifting liquid and in which both suction and discharge are open to the atmosphere.

Hertz: Frequency in cycles per second.

High service pump: A water pump that discharges to a pressurized piping system, which may include non-isolated, elevated storage.

Home runs: Wires running directly back to the power source from field devices such as lights.

Horizontal pump: A pump mounted with its shaft horizontal.

Hydraulic radius: Cross-sectional area of water divided by the wetted perimeter of a conduit.

Impeller: A circular casting mounted on a rotating shaft with vanes to accelerate the fluid.

Impeller eye: An opening in the impeller of a centrifugal pump through which liquid enters the impeller.

Impeller hub: The outside portion of the impeller of a centrifugal pump that encloses the eye.

Impeller shroud: The outside part of the impeller of a centrifugal pump to which the vanes are attached.

Inrush (current): The temporary surge of electrical current to a motor that occurs with an across-the-line starter.

Intake: A structure from which the pumps take suction.

Intake basin: See wet well.

Invert: The inside bottom of a pipe.

Jockey pump: A small pump that maintains pressure in a piping system during periods of low demand.

Knee: A reduction in the gradient of a pipeline or a change from a positive to a negative gradient.

Laminar flow: Flow with straight streamlines, i.e., without eddying or turbulence.

Laying length: The length of a piece of pipe or a fitting in a piping system.

Lining: A covering applied or attached to a surface. Examples: T-Lock® and Linabond®.

Lumen: The rate at which light from a one-candle source flows through a unit solid angle ($1\,m^2$ at 1 m or $1\,ft^2$ at 1 ft).

Lux: A measure of luminous intensity equal to 1 lumen per square meter or 0.0929 foot-candles.

Manifold: See discharge manifold.

Mechanical bar screen: A screen of bars, through which the influent of a pumping system flows before entrance to the pumps, with a motorized cleaning mechanism that delivers the screenings to a collector.

Mechanical seal: An assembly consisting of one or two pairs of polished surfaces, one of each pair mounted on (and turning with) the shaft and the

other connected to the casing to inhibit leakage of liquid between the casing and the shaft.

Micron: One thousandth of a millimeter (1×10^{-6}m).

Mil: Twenty-five microns (0.001 in.).

Mixed-flow pump: A pump that produces a combination of centrifugal and axial flow (see Figure 10-8).

Multistage: Two or more impellers mounted on a common shaft housed within casings for each impeller and designed to deliver liquid from the discharge of one casing to the suction of the other.

Net positive suction head (also NPSH): Absolute (total) dynamic head of the pumped liquid at the suction eye of a pump (see also Section 10-4).

Net positive suction head available (also NPSHA): The NPSH at which the pump in a given system operates at a given discharge rate.

Net positive suction head required (also NPSHR): The minimum NPSH at which a pump can properly operate for a given discharge rate.

Newton: A unit of force in SI units that accelerates a 1-kg mass $1\ \mathrm{m/s^2}$.

Newtonian fluid: A fluid (water, for example) in which shear flow is proportional to shear stress.

Nonclog pump: A centrifugal pump designed to pump liquids containing suspended solids and stringy material. (Ed. note: In spite of the name, they often do clog.)

Nonoverloading: The motor rating that is not exceeded by the brake horsepower required at any point on the characteristic curve of a pump.

Open impeller: An impeller with vanes on only one side of a single shroud plate.

Open shaft: See extended shaft.

Open system: A liquid piping system that includes nonisolated storage open to the atmosphere.

Operating point (for pumps): The head and discharge at which a pump operates in a system; the intersection of the pump characteristic curve and the system curve.

Overloading: The motor rating that is exceeded by the brake horsepower at some point on the characteristic curve of a pump.

Packing: Semiplastic material installed in a stuffing box to seal the shaft opening in the casing to restrict the leakage of liquid from the casing along the shaft.

Packing gland: See stuffing box.

Pascal: A unit of pressure or stress in SI units ($= \mathrm{N/m^2}$).

Permit-required confined space: A confined space with one or more of the following characteristics: (1) contains or has the potential to contain a hazardous atmosphere such as lack of oxygen (<19.95%) or explosive or toxic atmosphere; (2) contains a material that has the potential for engulfing an entrant; (3) has an internal configuration such that an entrant could be trapped or asphyxiated by inwardly converging walls or by a floor that slopes downward and tapers to a smaller cross-section; or (4) contains any other recognized serious safety or health hazard.

Pig: A device for cleaning the inside of a pipe.

Point: Percentage efficiency (of a pump) (e.g., 2 points equals 2%).

Poppet valve: A spring-loaded valve that operates automatically to relieve excessive pressure.

Potable water: Drinkable, finished water.

Pressure surges (also transients or water hammer): Rapid fluctuations in pressure in a closed piping system caused by a sudden change in the velocity of liquid in the pipe.

Primary element: An element that quantitatively converts measured variable energy into a form suitable for measurement.

Prime: Pump casing and suction piping completely filled with liquid.

Progressing cavity: A positive displacement pump with a helix rotor in a flexible, double helix casing for handling liquid with large amounts of solids in suspension.

Propeller pump: An axial flow pump suitable for pumping large volumes of liquids without stringy solids at low heads.

Pump: A machine that imparts kinetic and potential energy (from an external energy source) to a liquid to force a discharge from the machine.

Pump-down: An operation in which the water level in a sump is lowered as far as possible by the main pumps—generally to about $D/4$ above the last pump intake of diameter D.

Pumping station: A structure housing pumps, piping, valves, and auxiliary equipment.

Punch list: An itemized list of work the contractor must do to complete the job according to the plans, specifications, and change orders before the owner pays for substantial completion. Warranty repairs are not considered part of the punch list.

Radial vane: Impeller with vanes oriented radially to the shaft.

Raw sewage: See raw wastewater.

Raw wastewater: Wastewater that has not been settled or treated in any manner (except, possibly, for screening or grinding).

Raw water: Nonpotable water that must be treated to become finished water.

Reynolds number: A dimensionless number defined by Equation 3-12.

Root area: The cross-sectional area of a bolt calculated from the diameter at the root of the threads.

Rotodynamic pump: A mechanical device that pressurizes a fluid by means of a rotating impeller.

Runout: The operating condition for the maximum possible flow rate. For a pump, runout occurs at the least possible dynamic head. Hence, the pump usually operates beyond its BEP.

Sail switch: A device used to indicate failure of the ventilation system. A light paddle (within an air duct) is kept pushed up by a normal flow of air, but when air ceases to flow, the paddle falls, thereby allowing a limit switch to open and trigger an alarm.

Salient pole: A rotor (in an electric machine) consisting of projecting elements surrounded by coils for energization.

Scroll: The casing enclosing the rotor of a water turbine.

Secant piles: A double (or triple) ring of large-diameter, drilled, cast-in-place, reinforced concrete piles touching each other and having grouted interstices to ensure a water-tight wall.

Secondary element: An element that converts the output of a primary element (q.v.) into an electrical or pneumatic signal for monitoring or control, using switches for discrete signals and a transmitter for analog signals.

Self-cleaning: Ability to eject both settleable and floating solids on pump-down.

Self-priming pump: A centrifugal pump that normally retains prime or automatically reprimes itself when drawing suction from a liquid pool whose surface is below the casing of the pump.

Sequent depth: Depth following a hydraulic jump.

Service factor: A multiplier that, when applied to the rated horsepower, gives the load at which an electric motor can be operated.

Sewage: Domestic wastewater. May contain industrial wastewater.

Shaft sleeve: A replaceable cylinder mounted on (and rotating with) the pump shaft and extending through the stuffing box to protect the shaft from wear caused by the packing.

Sheet piles: Long, narrow steel plates with interlocking ball and socket edges that form a watertight wall. Wood is sometimes substituted for steel in shallow, dry excavations.

Shooting flow: Super-critical flow.

Shut-off head: The head developed by a centrifugal pump at zero discharge rate (against a closed discharge valve).

Sleeve coupling: A device to couple two plain-end pipes together (see Figure 4-3).

Sludge: Residue remaining after settling or treatment of wastewater and raw water. Usually about 3% or more solids concentration in suspension.

Slug: The unit of mass that is accelerated 1 ft/s^2 by a force of 1 lb ($\sim 32.17\,lb_m$).

Slug: The instantaneous addition of a known (usually large) amount of a tracer added to a fluid.

Slurry wall: Excavation by means of a clamshell of a short (~6 m or 20 ft long) series of trenches through a slurry of bentonite and water to support the walls. Cages of reinforcing bars can be lowered through the slurry. Concrete can be deposited by tremie.

Snubber: A device (such as a micropore filter) used to reduce pressure fluctuation to make a gauge indicate time-averaged pressure.

Soffit: The underside of a beam or the inside surface at the top of a pipe.

Soft foot: A machine foot (or support) with a bottom surface not coplanar with those of other feet and requiring compensation in shimming to avoid twisting the machine.

Soft start: Motor starting in which the inrush current is reduced.

Soldier beam: A vertical beam used to resist earth pressure.

Solids-bearing water: Water that develops either scum or deposits under stagnant conditions.

Solids capability: The diameter of the largest sphere that can pass through a pump from the suction flange to the discharge flange.

Special: Any piece of pipe (except a full-length straight section), such as short pieces, manifold sections, elbows, reducers, adapter sections with special ends, sections with outlets.

Specific energy: Depth plus velocity head.

Specific speed: A characteristic number for a geometrically similar series of pumps operating under similar conditions. Sometimes called the "type number" in Europe (see Table 10-1 and Figure 10-8).

Speed: Usually specific in this text for the velocity of propagation of a pressure wave.

Spike: A known addition to the concentration of a substance being analyzed; also a sudden change in a variable.

Split case: A two-piece casing bolted together for ease in providing access to the rotating assembly and all internal passages.

Spool: A section of flanged pipe shorter than a standard length.

Springline: A line on the inside wall of a pipe at its greatest width.

Static discharge head: The vertical distance from the center of the pump casing to the surface of the liquid at the point of discharge.

Static head: See total static head.

Static suction head: The vertical distance from the centerline of the pump casing to the surface of the liquid being pumped; the casing must be below the liquid level (h_s in Figure 10-2).

Static suction lift: The vertical distance from the centerline of the pump casing down to the surface of the liquid being pumped; the casing must be above the liquid level (see h_s in Figure 10-3).

Station losses: The headlosses (fluid friction) occurring in the station piping.

Station piping: The piping contained within the pumping station. By convention, it is regarded as the piping from the wet well or suction pipe, through the piping associated with the individual pumps, and ends at the connection to the pumping station discharge pipeline outside the building.

Steady bearing: A bearing between the pump and the motor for the lateral support of extended shafting.

Stuffing box: An assembly containing packing or a mechanical seal through which the pump shaft passes.

Sub-critical flow, velocity: Flow at velocity less than critical velocity.

Submersible pump: A pump or pump and motor suitable for fully submerged operation.

Suction elbow: Any elbow that is connected to the suction flange of a pump or an elbow, bolted to the floor of a wet well, that engages a submersible pump.

Suction flare or bell: A flaring entrance fitting to the pump suction piping.

Suction head: See static suction head and net positive suction head.

Suction lift: See static suction lift.

Sump: See wet well.

Super-critical flow, velocity: Flow at velocity greater than critical velocity.

Surface water: Raw water obtained from streams, lakes, or reservoirs.

Suspended: A pump in which the impeller is suspended below the liquid level on a vertical shaft supported from a motor above the liquid.

System head curve: Curve of total dynamic head (q.v.) versus flow for all flow rates within the capability of the pumping station.

Therm: 100,000 Btu.

Total dynamic head (TDH): The total head at which a pump operates at any given discharge rate (see Figure 10-2).

Total static head: The difference in elevation between the surface of the pool from which the pump draws water and the surface of the pool into which the outlet discharges (see Figure 10-2).

Tranquil flow: Sub-critical flow.

Trash basket: A basket with bars or screens through which influent of a pumping station flows before entering the liquid pool in the wet well to prevent large solids from entering the pumps.

Tremie: A movable metal tube (expanding into a funnel at the top) used for depositing concrete under water.

Tubercles: Rough deposits (formed inside of a pipe) that greatly increase resistance to flow.

Turbulent flow: Flow in which the streamlines are unsteady and the motion at a fixed point vanes in a random manner.

Type number: See specific speed.

Umbrella: An extension fastened to a pipe flare to form an entrance similar to a trumpet bell.

Upstream: Opposite to the direction of flow.

Varmeter: A wattmeter corrected for power factor.

Velocity: In pipe flow, the average velocity of the fluid.

Velocity head: The kinetic energy in a moving liquid; the average is $v^2/2g$.

Vertical pump: A pump with a vertical shaft.

Vertical turbine: A centrifugal pump usually with multistage impellers and bowls in a vertical column. It is suitable for pumping from wells and sumps.

Victaulic® coupling: See grooved-end pipe coupling.

Volute: A centrifugal pump casing that provides a gradually expanding liquid path to change the direction of flow and convert velocity to pressure.

Vortex impeller: A recessed impeller that creates a vortex in the casing to induce flow. The liquid flows over the impeller, not through it.

Wastewater: Water-borne wastes including those from households, businesses, and industry.

Water hammer: Rapid, severe, and often destructive changes in pressure in a piping system caused by a sudden change of liquid velocity.

Water horsepower: The power delivered by the pump to the liquid being discharged (= pound of fluid per minute × feet of head/33,000).

Water lance: A long, light pipe used for directing a water jet at the desired point of delivery.

Water seal: Clear water supplied under pressure to the packing or mechanical seal of a pump to lubricate, cool, and flush the packing or seal. (Grease or oil is sometimes used in place of water.)

Wear rings: Removable, replaceable metallic rings on the casing and/or impeller at the impeller hub.

Wet pit: See wet well.

Wet well: The below-grade compartment of a pumping station into which the liquid flows and from which the pumps draw suction. Also "pump intake basin" and "sump."

Wet well pump: A pump designed to be directly immersed in the liquid.

Withstand rating: Capability of an electrical device to withstand a short circuit without damage long enough to trip the protective devices.

Wrought pipe fittings: Pipe fittings formed by forging, piercing, hammering, welding, rolling, bending, pressing, or extruding (or by a combination of these).

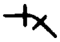
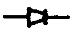
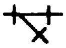
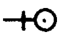
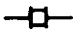
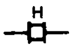
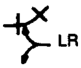
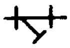
2-3. Symbols

Some of the symbols shown in Tables 2-1 to 2-6 are standard; others are typical, though not necessarily universal. Except for Table 2-4, which is reproduced from drawings by Brown and Caldwell, the tables are reproduced here by the courtesy of Greeley and Hansen Engineers.

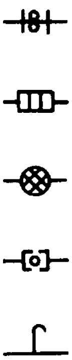
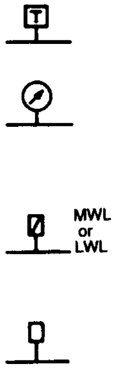
2-4. Supplementary Reading

Lyons, J. L., and C. L. Askland, Jr., *Encyclopedia of Valves,* Van Nostrand Reinhold, New York (1975).

Table 2-1. Single-Line Piping Symbols

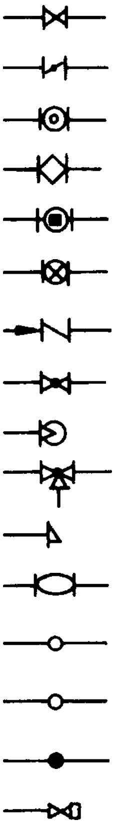
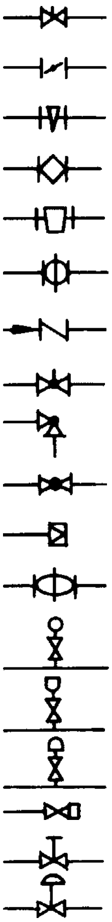
Pipe fittings			
Symbol	Description	Symbol	Description
	Cross		Tee
	Cross (branch up)		Tee (branch up)
	45° elbow		Tee (branch down)
	45° elbow (up)		Side outlet tee (up)
	45° elbow (down)		Reducer
	Lateral		Eccentric reducer (elevation)
	90° elbow		Union, screwed
	90° elbow (up)		Sleeve coupling
	90° elbow (down)		Sleeve coupling (harnessed)
	45° elbow (long radius)		Meter (identify type)
	90° elbow (long radius)		Venturi meter
	Base elbow		Expansion joint, metal bellows
	Side outlet elbow (up)		Expansion joint, rubber bellows
	Side outlet elbow (down)		Strainer

Pipe fittings

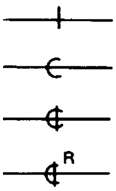
Symbol	Description	Symbol	Description
	Duplex strainer		Thermostat
	Flame trap		Pressure gauge
	Lube oil filter		Water level alarm
	Scale trap		Thermometer
	Vent		

Valves

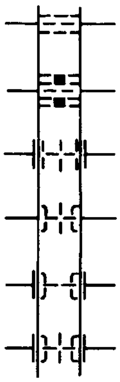
Stem relative to plane		
Perpendicular	Parallel	Description
		Gate
		Butterfly
		Eccentric plug
		Eccentric plug (alternate)
		Cone
		Ball
		Check, swing
		Globe
		Angle
		Three way
		Flap
		Diaphragm
		Auto air and vacuum release
		Auto air release
		Auto vacuum release
		Hose (bibb)
		Valve, manual operation
		Control valve with hydraulic, pneumatic, or electric actuator

Pipe joints



Symbol	Description
	Flange
	Bell and spigot
	Mechanical joint
	Mechanical joint, restrained
	Bell and ball

Wall fittings



Symbol	Description
	Wall sleeve, caulked
	Wall sleeve, annular seal
	Wall pipe, flange & flange, intermediate collar
	Wall pipe, bell & bell, intermediate collar
	Wall pipe, mechanical joint
	Wall pipe, mechanical joint, intermediate collar

Continued

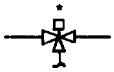
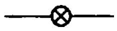
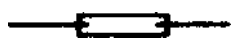
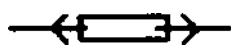
Table 2-2. Mechanical Symbols[a]

Fixture symbols

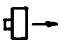
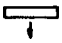
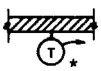
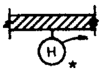
Symbol	Description	Symbol	Description
FD	Floor drain	WH	Water heater
AD	Area drain	L	Lavatory
RD	Roof drain	S	Sink
A	Compressed air outlet	SH	Shower stall
G	Gas outlet	U	Urinal
VAC	Vacuum outlet	WC	Water closet
HB	Hose bibb	WC	Water closet, wall hung
DF	Drinking fountain	EWC	Electric water cooler

Piping symbols

Symbol	Description	Symbol	Description
ST	Storm	F	Fire protection
S	Soil	SP	Fire sprinkler
W	Waste	Plan DT El DT	Drum trap
D	Drain		P-trap
V	Vent	RT RT	Running trap
AW	Acid waste		Backflow preventer
AV	Acid vent	CO	Cleanout (exposed pipe)
SW	Service water	CO	Cleanout (floor or grade)
CW	Cold water	HWS	Heating water supply
EW	Effluent water	HWR	Heating water return
HW	Hot water (potable)	CWS	Cooling water supply
HWC	Hot water circulating (potable)	CWR	Cooling water return
A	Compressed air	CHS	Chilled water supply
VAC	Vacuum	CHR	Chilled water return

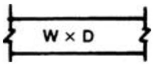
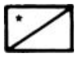
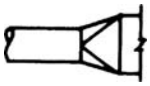
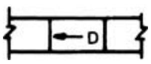
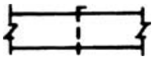
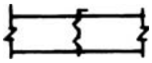
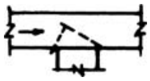
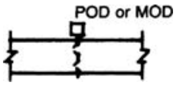
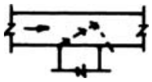
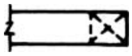
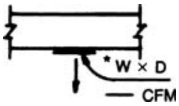
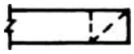
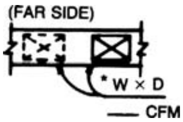
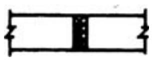
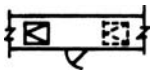
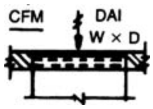
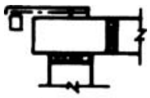
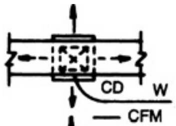
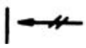
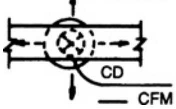
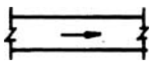
Symbol	Description	Symbol	Description
LPS	Low-pressure steam		Three-way control valve * (P): pneumatic * (E): electric
LPC	Low-pressure condensate		
MPS	Medium-pressure steam		Control valve * (P): pneumatic * (E): electric
MPC	Medium-pressure condensate		
HPS	High-pressure steam		Pipe guide
HPC	High-pressure condensate		Pipe anchor
FOS	Fuel oil supply		Float and thermostatic trap
FOR	Fuel oil return		Thermostatic trap
BF	Boiler feed		Expansion joint
NG	Natural gas		Expansion joint (harnessed)
LPG	Liquefied petroleum gas		Pipe flow direction
RL	Refrigerant liquid	DN	Pipe pitch down with respect to flow
RS	Refrigerant suction	UP	Pipe pitch up with respect to flow
CPD	Condensate pump discharge		Pipe capped

Miscellaneous symbols

Symbol	Description	Symbol	Description
	Unit heater	INV EL	Invert elevation
	Convection	℄ EL	Centerline elevation
	Cabinet heater	CFM	Cubic feet per minute
	Thermostat * (P): pneumatic * (E): electric	MBH	Thousand Btu/hour
		VD	Volume damper
	Humidostat * (P): pneumatic * (E): electric	TC	Temperature control
BD/EL	Bottom of duct elevation	NTS	Not to scale
TD/EL	Top of duct elevation	P	Pneumatic
		E	Electric

[a]After Greeley and Hansen Engineers.

Table 2-3. Heating, Ventilating, and Air Conditioning Symbols[a]

Symbol	Description	Symbol	Description
	Duct width × depth (first dimension is dimension seen)		Negative pressure duct section; asterisk represents service designation
	Inclined rise (with respect to air flow)		Round duct with transition to rectangular duct
	Inclined drop (with respect to air flow)		Splitter damper
	Volume damper (shaft parallel to paper)		Turning vanes
	Volume damper (shaft perpendicular to paper)		Deflecting damper with rod and lock
	Automatic opposed blade control damper (POD) pneumatic; (MOD) electric		Extractor with rod and lock
			Supply duct down (positive pressure)
	Register or grille (face perpendicular to paper); asterisk represents equipment designation		Exhaust or return duct down (negative pressure)
	Register or grille (face parallel to paper); asterisk represents equipment designation		Flexible duct connection
	Equipment designations		Access doors in duct
R	Register		
G	Grille		Outside air intake
CR	Ceiling register		
CG	Ceiling grille		
			Single inlet fan (plan view)
	Rectangular ceiling diffuser (size given refers to neck size)		Outlet air direction
			Inlet air direction
	Round ceiling diffuser (size given refers to neck size)		Direction of air flow in duct
			Service designations
	Positive pressure duct section; asterisk represents service designation	DA	Outside air
		S	Supply air
		E	Exhaust air
		R	Return air

[a]After Greeley and Hansen Engineers.

Table 2-4. Process and Instrumentation Diagram Symbols[a]

Miscellaneous mechanical equipment symbols

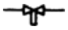
Symbol	Description	Symbol	Description	Symbol	Description
	Centrifugal pump		Rotary lobe compressor		Spiral heat exchanger
	Submersible sump pump		Liquid ring blower or compressor		Shell-and-tube-type heat exchanger
	Vertical pump		Silencer	RAG	Right angle gear
	Gear pump		Inlet air filter silencer		Tank
	Rotary lobe pump		Mixer		Waste gas burner
	Progressive cavity pump	ASD	Adjustable-speed drive		Engine
	Diaphragm pump		Generator		Horizontal surface mixer
	Boiler		Grinder pump		Sight glass
	Blower or fan		Plate-type heat exchanger		
	Compressor				

Valve and actuator symbols

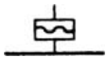
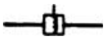
Symbol	Description	Symbol	Description	Symbol	Description
N.O.	Normally open				
N.C.	Normally closed		Diaphragm valve	F.O.	Three-way valve (with typical fail position)
N.O. N.C.	Gate valve		Needle valve		
N.O. N.C.	Globe valve		Balancing cock	F.O. F.O.	Four-way valve (with typical fail position)
N.O. N.C.	Plug valve		Knife gate valve		Pressure-reducing valve (flow to right)
N.O. N.C.	Ball valve		Circuit-balancing valve		Back-pressure-reducing valve (flow to left)
	Butterfly valve		Telescoping valve		Valve with hand actuator
	Check valve		Relief valve	S	Solenoid-operated valve
	Ball check valve		Float valve	M	Electrical-motor-operated valve
	Pinch valve	F.O.	Fail open		
	Angle valve	F.C.	Fail closed		Piston-operated valve

Continued

Valve and actuator symbols

Symbol	Description	Symbol	Description	Symbol	Description
	Diaphragm-operated valve		Thermostatically controlled valve		Sluice gate (normally closed)
	Electrohydraulic-operated valve		Sluice gate (normally open)		

Instrumentation symbols for primary elements

Symbol	Description	Symbol	Description	Symbol	Description
	Orifice plate		Flume		Flow-straightening vanes
	Venturi or flow tube		Weir		Chemical seal
	Averaging pitot tube		Variable-area flow meter (rotameter)		Concentric chemical seal
	Propeller or turbine meter		Rupture disc		Temperature well

Process and signal line symbols

Symbol	Description	Symbol	Description	Symbol	Description
	Main process flow (with typical direction of flow shown)		Electric signal (analog)		Electromagnetic or sonic signal (unguided)
	Secondary process flow		Pneumatic signal (discrete)		Software or data link
	Instrument supply, process taps		Electric signal (discrete)		Mechanical link
	Pneumatic signal (analog)		Capillary tube		Hydraulic
			Electromagnetic or sonic signal (guided)		

[a]Courtesy of Brown and Caldwell.

Table 2-5. Electrical Engineering Diagram Symbols[a]

SYMBOL	DESCRIPTION
	DC MOTOR WITH SERIES FIELD
	REACTOR (NONMAGNETIC CORE)
	REACTOR (MAGNETIC CORE)
	HALF WAVE RECTIFIER
+ −	FULL WAVE RECTIFIER
+ −	BATTERY
	CAPACITOR
	DIODE
R	RESISTOR
R	RESISTOR OR RHEOSTAT VARIABLE
R, M	RHEOSTAT-MOTOR OPERATED
	3-PHASE WYE (UNGROUNDED)
	3-PHASE WYE (GROUNDED)
	3-PHASE WYE (RESISTANCE GROUNDED)
	3-PHASE DELTA (UNGROUNDED)
	3-PHASE DELTA (GROUNDED)
	TWO MACHINES DIRECT CONNECTED
	MECHANICAL CONNECTION
	MECHANICAL INTERLOCK
	LIGHTNING OR SURGE ARRESTER

CONTROL AND INSTRUMENT SWITCHES	
CS	CONTROL SWITCH
SC	SECONDARY CONTROL
GS	GOVERNOR SWITCH
SCS	SPEED CONTROL SWITCH
SYS	SYNCHRONIZING SWITCH
DT	DUTY TRANSFER SWITCH
VS	VOLTMETER SWITCH
AS	AMMETER SWITCH
SS	SELECTOR SWITCH
VR	VOLTAGE REGULATOR
ANN	ANNUNCIATOR
POT	POTENTIOMETER

SYMBOL	DESCRIPTION
SWITCHES	

PROTECTIVE RELAYS	
27	UNDERVOLTAGE
32	AC DIRECTIONAL OVERCURRENT
46	PHASE BALANCE
47	PHASE SEQUENCE VOLTAGE
48	INCOMPLETE SEQUENCE
49	THERMAL
50	INSTANTANEOUS OVERCURRENT
51	AC TIME OVERCURRENT
67	AC DIRECTIONAL OVERCURRENT
68	BLOCKING
81	FREQUENCY
86	LOCK OUT
87	DIFFERENTIAL
GSR	GROUND SENSING RELAY

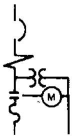
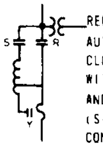
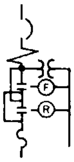
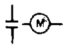
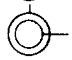
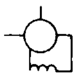
CONTROL DEVICES AND SWITCHES	
AR	AUXILIARY RELAY
CR	CONTROL RELAY
LS	LIMIT SWITCH
LL	LIQUID LEVEL SWITCH
DP	DIFFERENTIAL PRESSURE SWITCH
F	FLOAT SWITCH

SYMBOL	DESCRIPTION
CONTROL DEVICES	
FW	FLOW SWITCH
P	PRESSURE SWITCH
V	VACUUM SWITCH
T	THERMOSTAT
TQ	TORQUE SWITCH
TR	TIMING RELAY
VIB	VIBRATION SWITCH

METERS AND INSTRUMENTS	
A	AMMETER
V	VOLTMETER
W	WATTMETER
WH	WATTHOUR METER
DM	DEMAND METER
VAR	VARMETER
PF	POWER FACTOR METER
F	FREQUENCY METER
TG	TACHOMETER GENERATOR
TA	TACHOMETER
SY	SYNCHROSCOPE
TT	TOTAL TIME OR ELAPSED TIME METER
TMR	TIMER
T°	TEMPERATURE
GD	GROUND DETECTOR
SI	SPEED INDICATOR

* – INDICATING LIGHT (*-INDICATES COLOR)
G – GREEN INDICATES NON OPERATING CONDITION
R – RED INDICATES OPERATING CONDITION
B – BLUE INDICATES TROUBLE CONDITION
Y – YELLOW INDICATES OPERABLE CONDITION
A – AMBER
C – CLEAR
O – ORANGE
W – WHITE

Continued

SYMBOL	DESCRIPTION
	ACROSS-THE-LINE MAGNETIC CIRCUIT BREAKER OR MOTOR CIRCUIT PROTECTOR COMBINATION STARTER WITH CONTROL TRANSFORMER AND OVERLOAD RELAYS (M-OPERATING COIL)
	REDUCED VOLTAGE STARTER AUTOTRANSFORMER TYPE. CLOSED CIRCUIT TRANSITION WITH CONTROL TRANSFORMER AND OVERLOAD RELAYS (S-START CONTACTOR, R-RUN CONTACTOR, Y-WYE CONTACTOR)
	ACROSS-THE-LINE MAGNETIC CIRCUIT BREAKER OR MOTOR CIRCUIT PROTECTOR COMBINATION REVERSING STARTER WITH CONTROL TRANSFORMER AND OVERLOAD RELAYS (F-FORWARD, R-REVERSE)
	OIL FUSE CUTOUT (RATING)
	TRANSFORMER
	AUTOTRANSFORMER
	POTENTIAL TRANSFORMER
	CURRENT TRANSFORMER
	MANUAL CONTACTOR
	MAGNETIC CONTACTOR WITHOUT OVERLOAD PROTECTION (M-OPERATING COIL)
	THERMAL OVERLOAD ELEMENT (OL)
	SQUIRREL CAGE MOTOR
	WOUND ROTOR INDUCTION MOTOR
	SYNCHRONOUS MOTOR OR GENERATOR (G INDICATES GENERATOR)
	DC MOTOR WITH SHUNT FIELD OR DC GENERATOR (G INDICATES GENERATOR)

[a]Courtesy of Greeley and Hansen Engineers.

Table 2-6. Electrical Power and Lighting Symbols[a]

SYMBOL	DESCRIPTION
	SERVICE ENTRANCE (OVERHEAD)
	UNDERGROUND CABLE
	UNDERGROUND CONDUIT OR DUCT
	CONCEALED CONDUIT IN BUILDING
	EXPOSED CONDUIT OR DUCT IN BUILDING
	CONDUIT WITH ONE NEUTRAL AND ONE OR MORE HOT WIRES (LONG LINE IS FOR NEUTRAL, SHORT LINE IS FOR HOT)-LIGHTING
	HOME RUN TO PANELBOARD
	GROUNDING CONDUCTOR
	FLEXIBLE CONDUIT
	CONDUIT SEAL
	CONDUIT (UP)
	CONDUIT (DOWN)
	CONDUIT CAPPED
JB JB	JUNCTION BOX
PB PB	PULL BOX
TB	TERMINAL BOX
	GROUNDING ROD
	LIGHTING PANEL
	EQUIPMENT ENCLOSURE
	MANUALLY OPERATED SWITCH OR STARTER
M	MAGNETICALLY OPERATED STARTER FULL VOLTAGE
R	MAGNETICALLY OPERATED STARTER REDUCED VOLTAGE
C	MAGNETIC CONTACTOR
	THREE PHASE MOTOR (*-HP)
	SINGLE PHASE MOTOR (*-HP)
	PUSH BUTTON STATION
L	PUSH BUTTON WITH LOCKOUT ATTACHMENT
	CEILING OR PENDANT LIGHT
	BRACKET LIGHT
	COLUMN LIGHT
	EMERGENCY LIGHT
	EXIT LIGHT(BRACKET, PENDANT)
	FLOOD LIGHT
	FLUORESCENT LIGHT
	FLUORESCENT STRIP LIGHT
	FLUORESCENT LIGHT WITH EMERGENCY POWER SUPPLY UNIT
EM	EMERGENCY BATTERY UNIT
	EMERGENCY BATTERY LIGHTING HEAD
	UNIT HEATER
	RECEPTACLE (SINGLE AND DUPLEX)
XP	EXPLOSION-PROOF
WP	WEATHERPROOF

SYMBOL	DESCRIPTION
	RECEPTACLE - 3 PHASE
C	CLOCK
	SWITCH - SINGLE POLE
2P	SWITCH - 2 POLE
PL	SWITCH WITH PILOT LIGHT
T	SWITCH WITH THERMAL OVERLOAD PROTECTION
3W, 4W	SWITCH - 3 OR 4 WAY
	PLANT INTERCOM
	PUBLIC TELEPHONE
	PHOTOELECTRIC RELAY
SV	SOLENOID VALVE
K	SWITCH-KEY OPERATED
M	SWITCH-MOMENTARY CONTACT
Ø	POWER POLE

SCHEMATIC AND WIRING DIAGRAM SYMBOLS

SYMBOL	DESCRIPTION
*	CIRCUIT BREAKER (*-TRIP RATING)
	OPERATING COIL
	CONTACT - NORMALLY OPEN
	CONTACT - NORMALLY CLOSED
TC	NORMALLY OPEN CONTACT WITH TIME DELAY CLOSING
TO	NORMALLY CLOSED CONTACT WITH TIME DELAY OPENING
	3 - POSITION SELECTOR SWITCH
	NORMALLY OPEN PUSH BUTTON
	NORMALLY CLOSED PUSH BUTTON
	DOUBLE CIRCUIT PUSH BUTTON
	DOUBLE CIRCUIT MAINTAINED TYPE PUSH BUTTON
IC TO	NORMALLY OPEN CONTACT INSTANTANEOUS CLOSING AND TIME DELAY OPENING
	HORN OR SIREN
	BUZZER
	BELL
	CONDUCTORS CONNECTED
	GROUND CONNECTION

SYMBOL	DESCRIPTION
*	BUS (*-RATING)
	INCOMING FEEDER
	OUTGOING FEEDER
	POTHEAD
	DRAWOUT TYPE DEVICE
*	FUSE (*-RATING)
*	DRAWOUT TYPE FUSE (*-RATING)
*	SINGLE-THROW DISCONNECT SWITCH (*-RATING)
*S/*F	FUSED DISCONNECT SWITCH (*S-SWITCH RATING, *F-FUSE RATING)
	DOUBLE THROW SWITCH (*-RATING)
	LOW VOLTAGE AIR CIRCUIT BREAKER WITHOUT TRIP DEVICE NON-AUTO (*-FRAME RATING)
*F T	LOW VOLTAGE DRAWOUT TYPE AIR CIRCUIT BREAKER (*F-FRAME RATING *T-TRIP RATING)
F T	LOW VOLTAGE TYPE AIR CIRCUIT BREAKER (F-FRAME RATING T-TRIP RATING)
F S S	LOW VOLTAGE AIR CIRCUIT BREAKER (*F-FRAME RATING, S-SOLID STATE SENSOR RATING)
	LOW VOLTAGE DRAWOUT TYPE AIR CIRCUIT BREAKER ELECTRICALLY OPERATED
	LOW VOLTAGE DRAWOUT TYPE AIR CIRCUIT BREAKER WITH ONE NORMALLY OPEN AND ONE NORMALLY CLOSED AUXILIARY CONTACT
52	MEDIUM AND HIGH VOLTAGE CIRCUIT BREAKER
52	MEDIUM AND HIGH VOLTAGE CIRCUIT BREAKER DRAWOUT TYPE
52	MEDIUM AND HIGH VOLTAGE CIRCUIT BREAKER WITH DISCONNECT SWITCH

[a]Courtesy of Greeley and Hansen Engineers.

Chapter 3
Flow in Conduits

ROBERT L. SANKS

CONTRIBUTORS
Appiah Amirtharajah
Joel E. Cahoon
Alfred B. Cunningham
William F. H. Gros
George E. Hecker
James J. McCormick
Rhys M. McDonald
Charles D. Morris
Dennis R. Neuman
Constantine N. Papadakis
Robert E. Phillips
Sanjay P. Reddy
Arnold R. Sdano
Earle C. Smith
Thomas M. Walski
Gary Z. Watters
William Wheeler

This chapter is a primer for those with little knowledge of hydraulics, a review for those with experience, and a presentation of unique pipe flow tables, curves, and computer programs for everyone. Even experts will find some of this material new and useful.

Standards and specifications, such as ANSI B36.10, are commonly (and sufficiently) identified by the call number and, hence, are not referenced in Section 3-14. Double designations, such as ANSI/AWWA 104/A21.4, means that AWWA 104 is the same as ANSI A21.4.

3-1. Fundamentals of Hydraulics

The analysis of water flow in closed and open conduits depends on three fundamental principles—the conservation of (1) mass, (2) energy, and (3) momentum. Each principle is considered in terms of the equation(s) derived from its application.

Continuity Equation for Mass

Based on the conservation of mass, a material balance for steady, continuous flow in any conduit means that the flow rate of material weight in equals the flow rate of material weight out, or

$$\rho_1 Q_1 = \rho_2 Q_2 = \text{constant} \qquad (3\text{-}1)$$

where ρ is density and Q is flow rate or discharge. However, Q equals average velocity times cross-sectional area, and for water, $\rho_1 = \rho_2$, so

$$Q = A_1 v_1 = A_2 v_2 = \text{constant} \qquad (3\text{-}2)$$

Energy Equation

Energy per unit mass in flowing water, expressed in kg · m/kg (or ft × lb/lb), is the summation of three quantities:

- Elevation head (potential energy) above a datum (an assumed level), usually termed z and expressed in meters (or feet);
- Pressure head (also potential energy). To put this into the same units as z (elevation), the pressure (p) is divided by the specific weight (γ) and is expressed in meters (or feet). Pressure head is shown by the height to which the fluid would rise in a piezometer (an open tube), as shown in Figure 3-1;
- Velocity head (kinetic energy). To put this into the same units (meters or feet), it is expressed as $v^2/2g$, where v is velocity in meters per second (feet per second) and g is acceleration due to gravity, $9.81\,\mathrm{m/s^2}$ ($32.2\,\mathrm{ft/s^2}$). More exactly, velocity head is $\alpha v^2/2g$, where α is a term to convert velocity heads in each streamline to average velocity head for the section. For regular channels, α averages 1.15, but velocity head is small compared to pressure head, and the difference between velocity head on the right and left sides of the equation is smaller yet, so the assumption that $\alpha = 1.0$ produces all of the practical accuracy required.

In equation form, for either open channels or closed and pressurized conduits, the total energy head is given by

$$E = z + \frac{p}{\gamma}\cos\theta + \alpha\frac{v^2}{2g} \tag{3-3}$$

In open channel flow, if the datum plane is taken at the invert (bottom of conduit) and if the slope of the invert, θ, is zero, then z becomes zero, and $\cos\theta$ becomes unity. As explained above, α can be assumed to be unity with little error. Defining E as the specific energy, E_s,

$$E_s = \frac{p}{\gamma} + \frac{v^2}{2g} = y + \frac{v^2}{2g} \tag{3-4}$$

Conservation of Energy

The energy of water flowing between two points in either an open channel or a closed conduit is derived from Equation 3-3 and becomes the Bernoulli equation

$$z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} = z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + h_f + \Sigma h_m \tag{3-5}$$

where h_f is the energy per unit mass dissipated by friction between points 1 and 2 and Σh_m is the summation of headlosses due to turbulence in pipe fittings. The datum plane can be assumed at any level in Equation 3-5 because as z increases, p/γ decreases by the same amount. The application of Equation 3-5 to pipes is shown in Figure 3-1. Note how the pressure

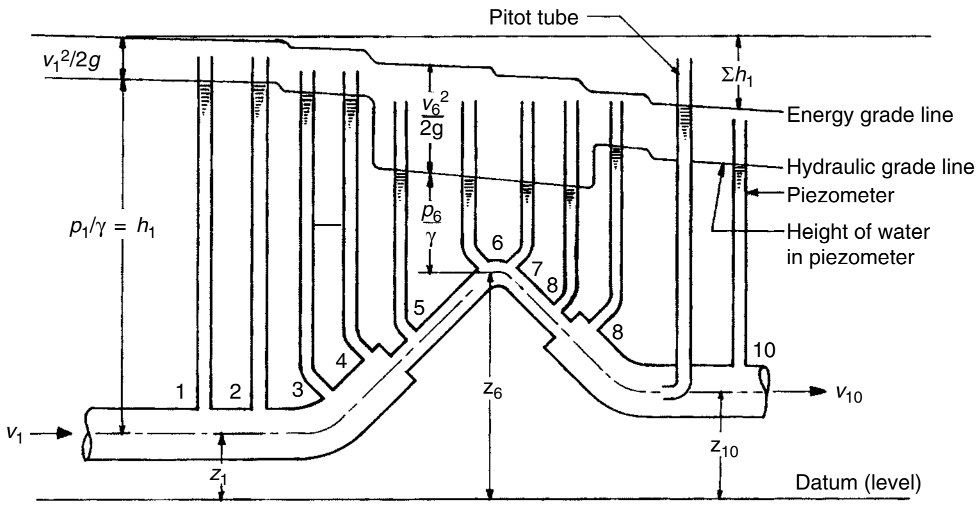


Figure 3-1. Pipe, piezometers, and energy and hydraulic grade lines.

head, the height to which water would rise in a piezometer (an open vertical tube), decreases as the velocity increases (as it must) in the smaller pipe.

Energy and Hydraulic Grade Lines

The energy grade line is a plot of the energy head, E (as defined by Equation 3-3), versus horizontal distance. In Figure 3-1, the energy grade line drops both gradually (due to friction head loss) and in steps (due to the turbulence in bends and discontinuities). The pressure at point 1 forces the water to rise in the piezometer. A pressure gauge at point 1 would read p_1, which, divided by γ, is the pressure head h_1. The locus of all pressure head values is the hydraulic grade line—always below the energy grade line by the amount of velocity head, $v^2/2g$. At each bend or constriction there is a small energy loss due to turbulence. At expansions (point 8), the energy loss is substantially greater. Even if the pipe were straight and prismatic from point 1 to point 10, there would be an energy loss due to friction caused by (1) shear at the wall and in the fluid, and (2) turbulence due to eddy formation.

The pitot tube in Figure 3-1 penetrates the pipe wall and the probe is bent with its open end facing upstream. In this open end of the probe of the pitot tube, the velocity is zero because there is no flow in the tube. But a short distance upstream, the velocity in the center of the pipe is v_c. Between that upstream point and the end of the probe, the kinetic energy head, $v_c^2/2g$, is converted to potential energy head and the water in the tube rises to the energy grade line. Actually, the water level in the pitot tube rises slightly above the energy line because v_c is the maximum, not the average, velocity. An average of all possible pitot tube readings from wall to center would match the energy grade line.

If the velocity in a medium-sized pipe (300 mm or 12 in.) is less than about 0.01 m/s (0.03 ft/s), there is no turbulence, friction depends only on shear, and the flow is called *laminar* or *tranquil*. The velocity distribution for laminar flow is pictured in Figure 3-2a. At the velocities usual in pipes, there are secondary, crosswise currents that cause mixing, the flow is turbulent, and the velocity distribution across a pipe appears as in Figure 3-2b. Between laminar and turbulent flow is a transitional zone where the flow might shift from laminar to turbulent and back again (or it might be a combination of both).

Momentum Equation

The force acting on a mass accelerates it according to Newton's third law of motion:

$$\mathrm{F} = \frac{d(mv)}{dt} \tag{3-6}$$

where F is force, m is mass, v is velocity, and t is time. F and v are vectors, and F can be considered the resultant (the vectorial sum) of any number of forces. If the mass remains constant, then

$$\mathrm{F} = m\,\frac{dv}{dt} = ma \tag{3-7}$$

where dv/dt is acceleration in the direction of F. In flowing water, it is convenient to rearrange Equation 3-7 to

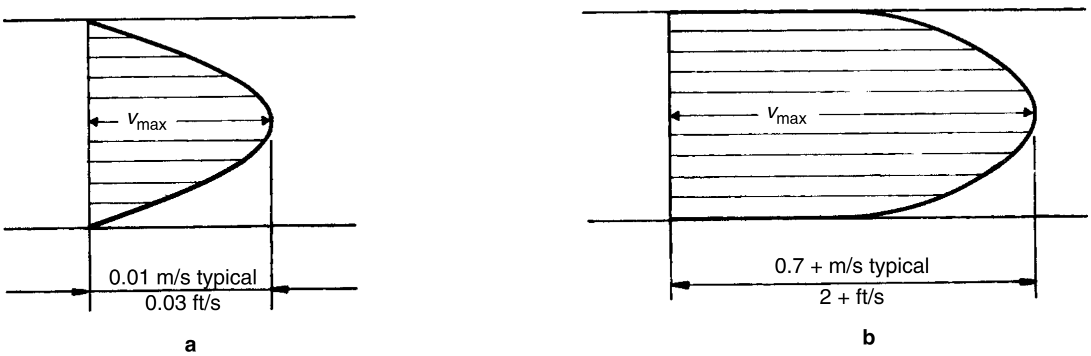


Figure 3-2. Velocity profile in a 300-mm (12-in.) pipe. (a) Laminar flow; (b) turbulent flow.

$$F = m\,\frac{dv}{dt} = \rho Q \Delta v = \rho Q(v_2 \rightarrow v_1) = (\gamma/g)Q(v_2 \rightarrow v_1) \tag{3-8}$$

where F is the sum (resultant) of all forces acting, ρ is mass density, Q is flow rate or discharge, the arrow sign indicates vectorial subtraction (change of velocity), γ is specific weight, and g is acceleration due to gravity.

A force acting on a free jet to deflect it is applied by the curved vane in Figure 3-3. Vectorial subtraction is made by changing the sign of vector $\rho Q v_1$ then adding it vectorially (as depicted in Figure 3-3b or 3-3c) to obtain F. The reaction of the jet against the vane is equal and opposite.

The momentum equation is needed to find the required strength of tie-downs, anchors, and thrust blocks to restrain piping at elbows, tees, etc. An example of the use of Equation 3-8 is given in Section 3-8.

3-2. Friction Losses in Piping

The first well-known formula for flow in pipes was proposed by deChezy. The deChezy friction coefficient was given by a complicated equation developed by Kutter. These formulas are no longer in common use. The Hazen–Williams (H–W) formula has been used in the United States for 90 yr. It is simple and easy to use, it has been verified by many field observations for common sizes of pipes at conventional flow rates, and its use is even mandated in the Ten-State Standards [1]. It has, however, some serious limitations. (See the next subsection, Hazen–Williams Equation.)

The Manning equation (Section 3-5) is somewhat similar to the H–W formula and is subject to the same limitations. It is widely used in the United States for open channel flow, such as pipes that are partly full. Sometimes it is used for full pipes, but for that application it has no advantage over the H–W formula.

The Colebrook–White equation is more accurate than the H–W formula and is applicable to a wider range of flow, pipe size, and temperature. It is widely used in the United Kingdom and elsewhere in Europe.

The Darcy–Weisbach equation is the only rational formula, and it is applicable to turbulent, laminar, or transitional flow, all sizes of pipe, and any incompressible Newtonian fluid at any temperature. It has not been popular because, being an implicit equation, it must be solved by successive trials. It was therefore inconvenient to use, but now, modern computers (even programmable pocket calculators) can be used to solve the equation in seconds.

In this text, only the H–W, Manning, and Darcy–Weisbach equations are discussed. Benedict [2] has discussed formulas for pipe flow extensively.

Hazen–Williams Equation and a Warning

The Hazen–Williams (H–W) equation, developed from extensive reviews of data on pipes installed all over the world, was made public in 1905 [3]. The appeal of the equation is due partly to its simplicity and ease of use, partly to the source of the data (real pipes in the field—not just laboratory pipes), and, by now, to a tradition of nearly a century of use and a blind faith in the results. Unfortunately, the formula is irrational; it is valid only for water at or near room temperature and flowing at conventional velocities; the flow regime must be in the transition zone (see Figure B-1); the C factor varies with pipe size; and 92% of the pipes studied were smaller than 1500 mm (90 in.) in diameter. These disadvantages seem to be generally ignored, but errors can be appreciable (up to 40%) for pipes less than 200 mm (8 in.) and larger than, say, 1500 mm (60 in.), for very cold or hot water, and for unusually high or low velocities. In field tests of a municipal water system only five years

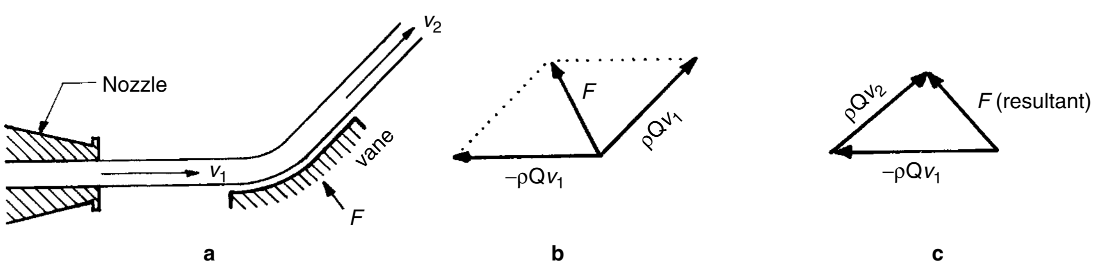


Figure 3-3. Impulse momentum. (a) Schematic diagram; (b) vector diagram; (c) equivalent vector diagram.

old, Bombardelli and Garcia [4] found the C value was only 91 for a 2250-mm (90-in.) pipe and 103 for a 1800-mm (72-in.) pipe, in contrast to the value of 120 used for design and thought to be conservative for good masonry. The low values were caused by deposits in the form of small scales several mm thick covering the inside surface. One long transmission main develops slimes that gradually reduce the C value from about 135 to 110. After a concentrated slug of chlorine is introduced, the C reverts to about 135 again. The errors in applying *any* formula for friction loss can be substantial and the consequences serious even for fairly new pipe. Consequently, prudent engineers use a *range* of coefficients to define a probable region of losses that may be encountered during the expected service life of the system.

The original form of the Hazen–Williams equation was developed in U.S. units. In SI units, the equation is

$$v = 0.849CR^{0.63}S^{0.54} \tag{3-9a}$$

where v is velocity in meters per second, C is a coefficient ranging from about 80 for very rough pipes to 150 for smooth pipes, R is the hydraulic radius in meters, and S is the friction headloss per unit length or slope of the energy grade line in meters per meter. The hydraulic radius is defined as the water cross-sectional area divided by the wetted perimeter. For full pipes, R reduces to $D/4$ where D is the ID.

In U.S. customary units, the equation is

$$v = 1.318CR^{0.63}S^{0.54} \tag{3-9b}$$

where v is in feet per second, R is in feet, and S is in feet per foot.

Friction headloss is expressed more conveniently as the gradient, h_f, in meters per 1000 m (or feet per 1000 ft) instead of S. Velocity can also be expressed as flow rate, Q, divided by water cross-sectional area, A. Substituting and rearranging Equation 3-9 yields another, somewhat more convenient form. In SI units,

$$h_f = \left[10{,}700\left(\frac{Q}{C}\right)^{1.85}\right]D^{-4.87} = \left(\frac{151Q}{CD^{2.63}}\right)^{1.85} \tag{3-10a}$$

where h_f is meters per 1000 m, Q is cubic meters per second, and D is pipe diameter in meters. Values of C are given in Table B-5 (Appendix B).

Equation 3-10b, expressed in U.S. customary units, is

$$h_f = \left[10{,}500\left(\frac{Q}{C}\right)^{1.85}\right]D^{-4.87} = \left(\frac{149Q}{CD^{2.63}}\right)^{1.85} \tag{3-10b}$$

where h_f is feet per 1000 ft, Q is gallons per minute, D is pipe diameter in inches, and C, again, is given in Table B-5 (Appendix B).

See Subsection "Friction Coefficients" for a discussion of the H–W C factor and Subsection "Warning" for reliance on answers that may be in error.

Darcy–Weisbach Equation

The equation for circular pipes is

$$h = f\frac{L}{D}\frac{v^2}{2g} \tag{3-11}$$

where h is the friction headloss in meters (feet), f is a coefficient of friction (dimensionless), L is the length of pipe in meters (feet), D is the inside pipe diameter in meters (feet), v is the velocity in meters per second (feet per second), and g is the acceleration of gravity, $9.81\,\mathrm{m/s^2}$ ($32.2\,\mathrm{ft/s^2}$). The advantages of the Darcy–Weisbach equation are as follows:

- It is based on fundamentals.
- It is dimensionally consistent.
- It is useful for any fluid (oil, gas, brine, and sludges).
- It can be derived analytically in the laminar flow region.
- It is useful in the transition region between laminar flow and fully developed turbulent flow.
- The friction factor variation is well documented.

The disadvantage of the equation is that the coefficient f depends not only on roughness but also on Reynolds number, a variable that is expressed as

$$\boldsymbol{R} = \frac{vD}{\boldsymbol{\nu}} \tag{3-12}$$

where $\boldsymbol{R}$ is Reynolds number (dimensionless), v is velocity in meters per second (feet per second), D is the pipe ID in meters (feet), and $\boldsymbol{\nu}$ is kinematic viscosity in square meters per second (square feet per second) as given in Appendix A, Tables A-8 and A-9.

Determination of f

In the laminar flow region where $\boldsymbol{R}$ is less than 2000, f equals $64/\boldsymbol{R}$ and is independent of roughness. Between

Reynolds numbers of 2000 and about 4000, flow is unstable and may fluctuate between laminar and turbulent flow, so f is somewhat indeterminate. When R is very large (greater than about 10^5), the flow is fully turbulent and f depends only on roughness. In the transition zone between turbulent and laminar flow, both roughness and R affect f, which can be calculated from a semi-analytical expression developed by Colebrook [5]:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{R\sqrt{f}}\right) \tag{3-13}$$

where ε is the absolute roughness in millimeters (or inches or feet) and D is the inside diameter in millimeters (or inches or feet), so that ε/D is dimensionless. The Moody diagram, Figure B-1 (Appendix B), was developed from Equation 3-13 [6]. Note that the curves are asymptotic to the smooth-pipe curve (at the left). To the right, curves calculated from the Colebrook (also called Colebrook–White) equation are indistinguishable from the horizontal lines for fully developed turbulent flow given in Prandtl [7]. The probable variation of f for commercial pipe is about $\pm 10\%$, but this variation is masked by the uncertainty of quantifying the surface roughness.

An explicit, empirical equation for f was developed by Swamee and Jain [8]:

$$f = \frac{0.25}{\left[\log_{10}\left(\frac{\varepsilon/D}{3.7} + \frac{5.74}{R^{0.9}}\right)\right]^2} \tag{3-14}$$

The value of f calculated from Equation 3-14 differs from f calculated from the Colebrook equation by less than 1%.

Friction headloss can be determined from the Darcy–Weisbach equation in a number of ways:

- Use one of the Appendix tables (B-1 to B-4) to find the appropriate pipe size. Compute R, find f from the Moody diagram, and compute an accurate value of h from the Darcy–Weisbach equation. Because f changes only a little for large changes of R, no second trial is needed. Compare the h so obtained with the value in Tables B-1 to B-4 for an independent check.
- Program Colebrook's Equation 3-13 to find f as an iterative subroutine for solving Equation 3-11 with a computer. Once programmed (a simple task even for a hand-held, card-programmable calculator), any pipe problem can be solved in a few seconds.
- Use the Swamee–Jain expression for f in the Darcy–Weisbach equation. Equation 3-14 could even be used as a first approximation for iteration of Equation 3-13.
- Refer to the extensive tables of flow by Ackers [9].
- Program the Moody diagram on a computer by assuming it to consist of a family of short, straight lines [10].
- Guess the pipe size and thus estimate v, calculate R, find f from the Moody diagram, and compute h from the Darcy–Weisbach formula. Revise v and D, if necessary, and recompute.

Other Pipe Formulas

There are many formulas for flow in pipes, but none is easier to use than the Hazen–Williams, and none is more accurate or universally applicable than the Darcy–Weisbach supplemented by the Moody diagram or the Colebrook equation. The limitation of the accuracy of all pipe formulas lies in the estimation of the proper coefficient of friction, a value that cannot be physically measured and, hence, is subject to large error (see Subsection "Friction Coefficients").

Comparison of f *and* C

The Darcy–Weisbach friction factor can be compared to the Hazen–Williams C factor by solving both equations for the slope of the hydraulic grade line and equating the two slopes. Rearranging the terms gives, in SI units,

$$f = \left(\frac{1}{C^{1.85}}\right)\left(\frac{134}{v^{0.15}D^{0.167}}\right) \tag{3-15a}$$

where v is in meters per second and D is in meters. In U.S. customary units, the relationship is

$$f = \left(\frac{1}{C^{1.85}}\right)\left(\frac{194}{v^{0.15}D^{0.167}}\right) \tag{3-15b}$$

where v is in feet per second and D is in feet. For any given pipe and velocity, the relation between ε/D and C can be found by calculating f from Equation 3-15 and entering the Moody diagram with R and f to find ε/D.

Friction Coefficients—Warning

The major weakness of any headloss formula is the uncertainty in selecting the correct friction coefficient. The proper friction factor for new pipe is

uncertain because of the variation in roughness of the pipe walls, quality of installation, effect of slight angular offsets in laying the pipe, and water quality. For example, the H–W C factor should be reduced by 5 units for pipe laid in hilly regions due to the angular deflection at joints.

Anticipating the friction factor after years of service is doubly difficult due to changes caused by corrosion, deposition of minerals or grease, or attachment of bacterial slimes. Estimation of the friction coefficient merits judicious attention.

A century ago, unlined cast-iron water pipe (or cast iron with the then-common but short-lived bituminous linings) did indeed become coated with tubercles and, thus, became very rough in a few years. But the modern use of cement mortar or plastic linings for ductile iron and steel pipe has eliminated the devastating effect of rust and tuberculation on friction. Plastic pipe remains very smooth unless foreign matter collects on the walls. Chemical precipitates (from water treatment) and bacterial slime in water pipes and grease or debris deposits in sewers can greatly increase the interior pipe roughness independently of the pipe material. This subject is addressed in more detail below.

Hazen–Williams C *Factor*

The basis of the Hazen–Williams C factor in Equation 3-9 has resulted in some confusion. The factor is a function not only of the smoothness of the pipe wall, but also of the difference between the actual ID of the pipe and the nominal pipe size. The calculation of C from field data is, by custom, based on the nominal diameter of the pipe. One could scarcely do otherwise, because finding the ID of a buried pipe is somewhat difficult and costly. Physical measurement at one point does not guarantee the ID at other points, so the best method is the use of tracers to measure the true average diameter. (See Section 3-9.) The use of nominal (not true average) diameters leads to strange conclusions. For example, the C value of an uncoated, new Class 50 ductile iron pipe 300 mm (12 in.) in diameter is typically given as 130. A Class 56 pipe carries 94% as much water, so its C value should be 122, but such a listing is unlikely to be found. The ratio of actual to nominal diameter accounts for the difference between the published C values for ductile iron pipe (DIP) and steel pipe with its smaller bore when both are lined with cement mortar. The confusion over the proper C value to use is worsened because the nominal diameter of steel pipe 300 mm (12 in.) and smaller is the ID, whereas for larger pipe, the nominal diameter is the OD. To permit reasonably accurate estimation of friction losses, the C value ought to be selected for the type of lining, thus allowing the true ID to be applied to the Hazen–Williams equation.

Matters are not improved by the apparent increase of C with diameter. According to AWWA Manual M11 [11], the average value of C for pipe with smooth interior linings can be approximated as $C = 140 + 0.17d$, where d is inside pipe diameter in inches. After a long term of lining deterioration, slime buildup, etc., $C = 130 + 0.16d$. However, above a diameter of about 900 or 1200 mm (36 or 48 in.), there is little increase in C values according to Gros [12], who has had many years of experience in measuring C values in the field. The values of C listed in the first part of Table B-5 reflect this experience.

In addition to the discussion above, there are other limitations on the value of C. Values of C less than 100 are only applicable for velocities reasonably close to 1 m/s (3 ft/s). At other velocities, the coefficients are somewhat in error. For water pipes, Lamont [13] advises the following:

- C values of 140 to 150 are suitable for smooth (or lined) pipes larger than 300 mm (12 in.).
- For smaller smooth pipes, C varies from 130 to 140 depending on diameter.
- C values from 100 to 150 are applicable in the transitional zone (between laminar and turbulent flow), but the scale effect for different diameters is not included in the formula.
- The formula is unsuitable and, hence, not recommended for old, rough, or tuberculated pipes with C values below 100.
- Force mains for wastewater can become coated with grease and C values may vary down to 120 or less for severe grease deposition.

Linings

Before 1950, it was common to line steel and cast-iron pipe with hot coal-tar dip, which provided poor protection and allowed C values to drop from 130 for new pipe to 100 or less for pipe in service for 20 yr or more [14]. The modern use of cement mortar or plastic linings makes pipe very smooth, prevents corrosion and tuberculation, and maintains its smoothness indefinitely. In field measurements [15] made all over the United States on new water pipe with diameters of 100 to 750 mm (4 to 30 in.) lined with cement mortar, the values of C varied from 134 to 151 (median = 149, average = 144). For 150- to 900-mm (6- to 36-in.) pipe in service for 12 to 39 yr, C varied

from 125 to 151 (median = 139, average = 140)—a decrease of only about 5 units.

Shop lining applied to DIP is usually one layer of cement mortar centrifugally cast with minimum thicknesses varying from 1.6 mm ($^1\!/_{16}$ in.) for small pipe to 3.2 mm ($^1\!/_8$ in.) for large pipe and given a thin asphaltic seal coat to control curing per ANSI/AWWA C104/A21.4. Optionally, a double thickness can be specified. The thicknesses of actual linings may vary considerably from those specified. These linings are so thin that many engineers specify double thickness to ensure adequate coverage, eliminate the danger of pinholes, and provide greater integrity. Shop linings applied to steel pipe can be coal-tar, enamel, thin plastic, or thick cement mortar varying from 6 to 13 mm ($^1\!/_4$ to $^1\!/_2$ in.) per AWWA C205. Cement mortar lining for steel pipe is customarily three to four times as thick as for DIP. See Table 4-6.

Field-applied cement-mortar linings, according to Table 4-7, can vary from 3 to 13 mm ($^1\!/_8$ to $^1\!/_2$ in.). Considering all the possibilities for pipe thickness and for shop and field linings, inside diameters can vary substantially. Designers should determine the metal IDs from manufacturers' catalogs and industry standards such as AWWA and ASME. Determine the probable net ID and the probable range of friction coefficients carefully. For final design, trust no table, but calculate the flow by formula. If the pipe is larger than about 600 mm (24 in.) or smaller than about 75 mm (3 in.), or if the temperature is less than about 10°C (50°F) or more than about 30°C (86°F), do not trust the Hazen–Williams formula. Use the Darcy–Weisbach formula instead.

Deposition in Pipes

Water treatment often creates deposits that greatly increase friction in pipes. In one pipeline, lime incrustation reduced the measured value of *C* to only 80 downstream from the treatment plant. Pipe can, however, be cleaned and relined with cement mortar in situ and restored to nearly its original smoothness. Under some circumstances, deposits of bacterial slime in water pipes can change the smoothest pipe (whatever the material) into very rough pipe. Fortunately, chlorination destroys the slime and restores the former smoothness. In New York, for example, the *C* factor for a 1800-mm (72-in.) water main 7.7 km (4.8 mi) long drops from 140 to 120 about twice per year and is chlorinated to restore the *C* value to 140. Another example is a 1050-mm (42-in.) cement-lined steel cylinder prestressed concrete transmission main 48 km (30 mi) long. It develops a slime layer only about 3 mm ($^1\!/_8$ in.) thick every five years, but the thin slime is sufficient to decrease the *C* value from 140 to 100. A massive dose of chlorine restores its former smoothness. Instead of massive doses of chlorine at long intervals, however, the maintenance of a free chlorine residual of about 0.5 mg/L or a stronger (2 to 3 mg/L) dose for two hours twice per week in a southern California pipeline has been reported to maintain the original capacity. As bacteria do not develop immunity to chlorine, experimentation with doses, contact time, and time intervals between doses offers an opportunity to achieve overall economy [16]. Biofouling is far more prevalent than most people realize, so chlorination facilities must be added for pipelines subject to slime buildup. The coefficient of friction should be determined when a new pipeline is first put into service to establish an irrefutable reference point for future cleaning needs and for evaluating cleaning procedures.

Sewers often become fouled with grease, and grease from industries (such as commercial laundries, slaughterhouses, or locomotive repair shops) can reduce the diameter of wastewater pipes by one-third or more. Their original size and smoothness can be restored, however, by cleaning the pipe in place. To prevent excessive buildup of grease, include pig launching and recovery stations (see Section 4-9).

The headloss in pumping station piping is usually small (about 2 m or 5 ft) and is largely related to valve and fitting losses (see Tables B-6 and B-7), so the selection of a *C* value for piping within the pumping station is of minor importance. If the static lift is the major part of the TDH and the transmission or force main is short, say 150 m (500 ft) or less, the *C* value is of minor importance.

Long Force Mains

Friction coefficients for long force or transmission mains must be established with great care. Using the Hazen–Williams formula can lead to serious errors, particularly for (1) large pipes, (2) high velocities, or (3) water temperature that differs from 15°C (60°F) by more than about 11°C (20°F). For such situations, use the Darcy–Weisbach equation. If the energy loss is a vital design consideration, search the literature for tests on similar conduits instead of attempting to use the Moody diagram for the Darcy–Weisbach equation.

Pump and Impeller Selection

To base pump operating points on station curves drawn for an unrealistic roughness is a serious blunder. If some jurisdiction requires the use of some specific

value of roughness deemed by the designer to be too great (for example, a C value of 120, as specified in the Ten-State Standards), use that value only to find the size of the pipe. Select the pump and its impeller for a rational envelope of curves that include the maximum and minimum limit of possible roughness, for example, $150 > C > 120$. By choosing a pump that can accommodate impellers of a substantial range of diameters, the pump can be modified to operate at or near its best efficiency point (BEP) for any curve within the envelope. However, assuming an excessively rough pipe can be disastrous. One pumping station featured several sets of two pumps in series to develop the head calculated for a single C value of 100. To keep the pumps from vibrating, the operators partly closed a downstream valve. A better solution would have been to bypass the tandem pump and achieve a savings of $100,000 per year in electric power.

It is wise to use a calibrated pressure gauge in measuring the total dynamic head (TDH) during the start-up procedure so that the impeller trim can, if necessary, be refined with confidence.

3-3. Pipe Tables

So many materials, pipe diameters, wall thicknesses, and liner thicknesses can be used for pumping stations, yard piping, and transmission or force mains that complete tables of flow and headloss would have to be extensive indeed. Because interpolation between tabular values is onerous, calculating the flow and headloss with the Hazen–Williams formula is much quicker. Tables, however, can be used advantageously to find the proper size of pipe quickly, to approximate the friction headloss, and to provide an independent check on a solution by formula.

The purposes of the pipe tables in Appendix B (Tables B-l to B-4) are the following:

- For a quick, preliminary determination of pipe size, flow rate, and headloss for a moderate friction coefficient (C = 120) and velocity (2 m/s in Tables B-l and B-3 and 5 ft/s in Tables B-2 and B-4);
- For finding the available sizes and weights of the thinnest (and most common) pipes used within pumping stations;
- For a quick, rough check of flow or headloss found by other means; and
- For providing useful data for both ductile iron pipe (DIP) and steel in both SI and U.S. customary units.

For final design, for different conditions, and for piping outside of the pumping station, calculate flow and headloss with the Darcy–Weisbach formula and consult the tables to check for blunders.

Example 3-1
Designing Pipe with the Pipe Tables

Problem: Select the pipe for a water pumping station with a 15-km- (9.3-mi)-long transmission main. Maximum flow is $0.4\,\mathrm{m^3/s}$ (6360 gal/min or $14.1\,\mathrm{ft^3/s}$).
Solution: One choice for the pumping station is DIP lined with cement mortar and sized for a velocity of about 2.5 m/s (8.2 ft/s), which is high enough to minimize the size and cost of valves and other fittings and low enough to prevent cavitation and excessive headloss.

Using Table B-l for SI units (or Table B-2 for U.S. customary units),

SI Units

Pipe size for v = 2 m/s: 500 mm

$$\frac{v_{\text{desired}}}{v_{\text{Table B-1}}} = \frac{2.5\ \mathrm{m/s}}{2\,\mathrm{m/s}} = 1.25$$

$$\text{Area required} = \frac{0.213\,\mathrm{m^2}}{1.25} = 0.17\,\mathrm{m^2}$$

$$v = Q/A = 0.4/0.172 = 2.33\,\mathrm{m/s}$$

U.S. Customary Units

Pipe size for v = 5 ft/s: 24 in.

$$\frac{v_{\text{desired}}}{v_{\text{Table B-2}}} = \frac{8.2}{5} = 1.64$$

$$\text{Area required} = \frac{3.32\,\mathrm{ft^2}}{1.64} = 2.02\,\mathrm{ft^2}$$

$$v = Q/A = 14.1/1.85 = 7.78\,\mathrm{ft/s}$$

SI Units

Choose 450-mm pipe: $A = 0.172\,\mathrm{m}^2$

Friction headloss: use Equation 3-10a and $C = 120$

$$h_f = 10{,}700\left(\frac{0.4}{120}\right)^{1.85}(0.468)^{-4.87}$$

$$h_f = 11.3\,m/1000\,m$$

U.S. Customary Units

Choose 18-in. pipe: $A = 1.85\,\mathrm{ft}^2$

Friction headloss: use Equation 3-10b and $C = 120$

$$h_f = 10{,}500\left(\frac{6360}{120}\right)^{1.85}(18.4)^{-4.87}$$

$$h_f = 11.3\,ft/1000\,ft$$

Always check such calculations with the pipe table. Note that headloss is a function of Q or v to the 1.85 power. Hence,

$$\begin{aligned} h_{f_{\text{actual}}} &= h_{f_{\text{table}}}\left(\frac{Q_{\text{actual}}}{Q_{\text{table}}}\right)^{1.85} \\ &= 8.5\left(\frac{0.4}{0.344}\right)^{1.85} \\ &= 11.2\,\mathrm{m}/1000\,\mathrm{m} \end{aligned}$$

Good check.

$$\begin{aligned} h_{f_{\text{actual}}} &= h_{f_{\text{table}}}\left(\frac{Q_{\text{actual}}}{Q_{\text{table}}}\right)^{1.85} \\ &= 5.1\left(\frac{6360}{4160}\right)^{1.85} \\ &= 11.2\,\mathrm{ft}/1000\,\mathrm{ft} \end{aligned}$$

Good check.

Entrance, fitting, and valve losses must be added (see Section 3-4).

At $C = 145$, the friction headloss is 8.4 m/1000 m, which, in the short length of pipe in a pumping station, would only be about 60 mm (0.2 ft). At 11.3 m/1000 m, the loss in head would be only about 40 mm (0.1 ft) more, which is insignificant.

For the transmission main, a velocity of about 2 m/s (6 ft/s) seems likely to be economical when the cost of pipe, valves and fittings, water hammer control methods and devices, installation, and energy are analyzed for, say, a 20-yr period. Using Table B-1 or B-2, a 500-mm (20-in.) pipe would fit the conditions. But flanged joints are not needed for the transmission main, where mechanical or push-on joints allow Class 50 DIP to be used. By referring to the DIPRA handbook [17], the wall thickness is 9.1 mm (0.36 in.). Let us assume the cement mortar is to be double thickness with negligible tolerance. The OD values in Tables B-1 and B-2 are correct for all classes of DIP, so

$$\begin{aligned} \mathrm{ID} &= 549 - 2(9.1 + 2 \times 2.38) \\ &= 521\,\mathrm{mm} \end{aligned}$$

$$\begin{aligned} h_f &= 10{,}700\left(\frac{0.4}{120}\right)^{1.85}(0.521)^{-4.87} \\ &= 6.69\,\mathrm{m}/1000\,\mathrm{m} \end{aligned}$$

Or, use Table B-1,

$$\text{True velocity} = \frac{0.4\,\mathrm{m}^3/\mathrm{s}}{0.213\,\mathrm{m}^2} = 1.88\,\mathrm{m/s}$$

$$h_f = 7.5\left(\frac{1.88}{2}\right)^{1.85} = 6.69\,\mathrm{m}/1000\,\mathrm{m}$$

Good check.

$$\begin{aligned} \mathrm{ID} &= 21.60 - 2[0.36 + 2(3/32)] \\ &= 20.5\,\mathrm{in.} \end{aligned}$$

$$\begin{aligned} h_f &= 10{,}500\left(\frac{6360}{120}\right)^{1.85}(20.5)^{-4.87} \\ &= 6.65\,\mathrm{ft}/1000\,\mathrm{ft} \end{aligned}$$

Or, use Table B-2,

$$\text{True velocity} = \frac{14.1\,\mathrm{ft}^3/\mathrm{s}}{2.29\,\mathrm{ft}^2} = 6.16\,\mathrm{ft}$$

$$h_f = 4.5\left(\frac{6.16}{5}\right)^{1.85} = 6.62\,\mathrm{ff}/1000\,\mathrm{ft}$$

Good check.

For 15 km (9.3 miles) of pipe, the total headloss at $C = 120$ is $6.69 \times 15 = 100\,\mathrm{m}$ (330 ft) versus 70.5 m (231 ft) at $C = 145$. The difference in headloss and energy use is important and worthy of careful study. Check by using the Darcy–Weisbach formula; also consider the maximum and minimum water temperatures.

Air in Pipelines

Dissolved air (or other gas) is a serious problem in pipelines that have intermediate high points or are nearly flat. If air comes out of solution, it forms bubbles that accumulate, reduce the water cross-sectional area, and increase resistance to flow—sometimes greatly—and the air-moisture environment is conducive to corrosion. In sewer force mains, air and consequent corrosion is disastrous. Various ways to deal with air in pipelines include (1) designing the pipeline profile to rise all the way to the exit; (2) installing air release valves at high points in the pipeline (or at frequent intervals for flat profiles); or (3) designing for velocities high enough to scour air bubbles to the exit. Obviously, the first is preferred if possible. Air release valves are risky because of uncertain maintenance. They should not be used at all on wastewater force mains because maintenance must be done so frequently (for example, monthly) and without fail. (See Sections 5-7 and 7-1 for an exception.) If the valves are not maintained properly, they are worse than useless because they then engender a false sense of security. Designing for scouring velocities in large pipes may result in excessive headlosses and energy needs. The required scouring velocities are given in Table B-9.

Some consultants customarily install manways at 450-m (1500-ft) intervals in water pipelines equal to or larger than 900 mm (36 in.) in diameter to permit worker entry and inspection of, and repairs to, the lining and to fix leaks. Air release valves are required in the manway covers to prevent the accumulation of air under them.

3-4. Headlosses in Pipe Fittings

Pumping stations contain so many pipe transitions (bends, contractions) and appurtenances (valves, meters) that headlosses due to form resistance (turbulence at discontinuities) are usually greater than the frictional resistance of the pipe. The simplest approach to design is to express the headlosses in terms of the velocity head, $v^2/2g$, usually immediately upstream of the transition or appurtenance. The equation for these losses is

$$h = K\frac{v^2}{2g} \tag{3-16}$$

in which K is a headloss coefficient (see Appendix B, Tables B-6 and B-7). The few exceptions to Equation 3-16 are noted in the tables.

The headloss coefficient, K, is only an approximation, and various publications are not always in agreement and may differ by 25% or more. The values in Tables B-6 and B-7 have been carefully selected from many sources and are deemed to be reliable.

In Equation 3-16, K varies with pipe size as noted in Table B-6. Furthermore, published values are for isolated fittings with a long run (for example, 20 pipe diameters) of straight pipe both upstream and downstream from the fitting. The headloss is measured between one point a short distance upstream from the fitting and another point at the downstream end of the piping system. This piping ensures symmetrical flow patterns. The difference in headloss with and without the fitting is used to compute K. Headlosses for a series of widely separated fittings are therefore directly additive.

Part of the headloss is due to the turbulence within the fitting, but probably about 30% (less for partially closed valves) is due to eddying and turbulence in the downstream pipe. So if one fitting closely follows another (as in a pumping station), the apparent K value for the first fitting is, probably, reduced to about 70%. For example, because K for a 90-degree bend is 0.25 (see Table B-6), K for two 90-degree bends would be 0.50 if the bends were separated by, say, a dozen pipe diameters. But if the bends were bolted together to make a 180-degree bend, K for the entire bend could be figured as $0.70 \times 0.25 + 0.25 = 0.43$, which is within 8% of the K value for a 180-degree bend in Table B-6. As another example, K for a 90-degree bend consisting of three 30-degree miters can be determined directly from Table B-6 as 0.30 or indirectly by adding reduced K values for each miter except the last. Thus, $0.70\ (0.10 + 0.10) + 0.10 = 0.24$—an error of 20% (one publication lists the K for the mitered fitting as 0.20).

Pumps, especially when operating on either side of their best efficiency point, usually cause swirling (rotation) in the discharge pipe. Swirling sometimes also occurs in inlets and suction pipes. The effect of such swirling is to increase eddy formation and turbulence; consequently, the headloss in fittings can be doubled or even tripled. If swirling is likely to occur and if headloss within the pumping station is critical (which is often true in suction piping), the safe and conservative practice would be to design for headloss without swirling and again for headloss using, say, 200% of the fitting losses. Because there is no definitive body of literature about this complex subject, designers must either rely on experience or guess at headlosses.

Another method for computing headlosses is to use an "equivalent length" of straight pipe. This

method is less accurate partly because, for example, 50 pipe diameters of a smooth pipe length would have less headloss than the same length of rough pipe. But, of course, corrections can be made by multiplying a tabular value of equivalent length by the ratio of $C_{\text{actual}}/C_{\text{tabular}}$. This method is used in pipe network analysis for simplification, but there is no reason for using it in pumping station calculations.

3-5. Friction Losses in Open Channel Flow

The most common equation used in the United States for open channel flow is the Manning equation. In SI units,

$$v = \frac{1}{n} R^{2/3} S^{1/2} \tag{3-17a}$$

where v is velocity in meters per second, n is Manning's friction coefficient (given in Appendix Table B-5), R is the hydraulic radius in meters, and S is the slope in meters per meter. In U.S. units,

$$v = \frac{1.486}{n} R^{2/3} S^{1/2} \tag{3-17b}$$

where v is velocity in feet per second, R is hydraulic radius in feet, S is slope in feet per foot, and n is Manning's friction coefficient (in Table B-5); the constant, 1.486, converts SI to U.S. customary units. The Manning equation can be used for pipes flowing full, but has no advantage over the Hazen–Williams equation. For pipes flowing full and under pressure, the relationship between C and n is

$$n = 1.12 \frac{D^{0.037}}{CS^{0.04}} \tag{3-18a}$$

in SI units where D is the ID in meters. In U.S. customary units, the equation is

$$n = 1.07 \frac{D^{0.037}}{CS^{0.04}} \tag{3-18b}$$

where D is the ID in feet. (See Brater and King [18] for a more extensive discussion.)

Error in the Manning Equation

In spite of its common use for circular pipes flowing partly full, the Manning equation is valid only for full pipes. In extensive studies, Yarnell and Woodward [19] confirmed the exponents of $\frac{2}{3}$ and $\frac{1}{2}$, respectively, for R and S in Equation 3-17 for full pipes and also proved that the apparent values of n vary considerably with depth of flow. At depths from 15 to 55% of the pipe diameter, the observed apparent n is about 25% greater than n for a full pipe. Camp [20] added the work of Wilcox [21] to that of Yarnell and Woodward to obtain the curves shown in Figure B-5. (Camp also included a graph of slopes required for keeping sewers clean at minimum flow rate.)

Escritt [22] stated that the Manning equation is accurate within a few percent if half of the width of the free water surface is added to the wetted perimeter, P, of the pipe when computing a modified hydraulic radius, R_m. He also proposed an equation that Saatci [23] transformed into the expression

$$R_m = R_f \left[\frac{\theta - \sin\theta}{\theta + \sin\dfrac{\theta}{2}} \right] \tag{3-19}$$

where R_m is the modified hydraulic radius for a given depth, R_f is hydraulic radius ($D/4$) for a full pipe, and θ is the central angle in radians. See Figure B-4 for relations between θ, area, perimeter, and depth of flow. Equation 3-19 fits the curves of observed data in Figure B-5, usually within an error of about 3%. Wheeler [24] reduced the error to less than 2% by introducing empirical coefficients into Equation 3-19 as follows:

$$R_m = R_f \left[\frac{\theta - \sin\theta}{\theta + 1.23 \sin\dfrac{\theta}{1.924}} \right] \tag{3-20}$$

Quick Solutions for Manning's Equation

Velocities, depths, and water cross-sectional areas corresponding to the Manning equation are compared in Table B-8, in which observed values are closely (within 2%) represented by Equation 3-20. The table makes it particularly easy to find accurate values of the hydraulic elements for a pipe flowing partly full. Wheeler [25] also developed a computer program, PARTFULL®, that can be used to find hydraulic elements of open channel flow including the conjugate and sequent (before and after a hydraulic jump) depths for the flow rates given in Tables B-10 and B-11.

The most universal (and one of the quickest) means for solving open channel flow with or without the Escritt modification are found in a user-friendly

program, **UnifCrit2.2** developed by Cahoon [26]. **UnifCrit2.2** solves the Manning equation either unmodified (Escritt=0 in the program) or modified by the use of R_m (Escritt=1) in round pipes. This versatile program also solves for critical velocity, depth, and flow in both round and trapezoidal conduits. It can be downloaded freely from the internet at http://www.coe.montana.edu/ce/joelc/wetwell/.

Sewers

Sewers constitute one kind of open channel in which stable, uniform flow rarely occurs because of discontinuities such as changes of gradient and because of constantly changing flow, all of which cause long backwater curves. For practical purposes, however, these effects are not great and are easily absorbed in the assumption that small sewers flow "full" with the water surface at mid-depth. Sewers 375 mm (15 in.) in diameter and larger are assumed to flow full with a water depth of about 70% of pipe diameter.

Uncertainty in the "n" value and the amount of flow make hydraulic calculations of sewer flow quite uncertain. See subsection *Friction Coefficients—Warning* in Section 3-2.

Example 3-2
Design of a Sewer Pipe

Problem: Design a sewer pipe to carry a maximum design flow of $123\,m^3/hr$ (541 gal/min). Select the pipe material and find the required diameter and slope.
Solution: Sewer pipe is usually clay, concrete, or plastic. Plastic is popular because of its light weight, ease of handling, tight joints, durability, and economy, but it is flexible and tends to flatten, so care must be taken to backfill it properly.

Size: Plastic sewer pipe is available in the following nominal diameters: 200, 250, 300, 375, 450, 525, 600, etc. mm (8, 10, 12, 15, 18, 21, 24, etc. in.). To pick up grit deposited at low flows and to scour the pipe clean, the velocity at maximum flow should equal or exceed 1.1 m/s (3.5 ft/s). In large (> 600-mm or 24-in.) pipe, keep the velocity greater than 1.5 m/s (5 ft/s) to inhibit septicity and odors.

SI Units	**U.S. Customary Units**
From Table A-11 (Appendix A),	
$(123\,m^3/hr)(2.78 \times 10^{-4}) = 0.0342\ m^3/s$	$(541\,gal/min)(2.23 \times 10^{-3}) = 1.21\,ft^3/s$
The area required is, from Equation 3-2,	
$0.0342\ m^3/s = A \times 1.1\,m/s$ $A = 0.0311\ m^2$	$1.21 ft^3/s = A \times 3.5\ ft/s$ $A = 0.346 ft^2$
The area of a half-circle is $\pi D^2/8$, so	
$\pi D^2/8 = 0.0311\,m^2$ $D = 0.281\,m = 281\,mm$	$\pi D^2/8 = 0.346\,ft^2$ $D = 0.939\,ft = 11.3\,in.$
Choose a 300-mm pipe.	Choose a 12-in. pipe.

Slope: Plastic pipe is very smooth but it may become coated with grease, and Ten-State Standards [1] require n to be 0.013, which is rather rough (see Table B-5).

Because the pipe is larger than required, the water surface is below the center, so the perimeter is less than half the circumference. Using Equation 3-21 to find the wetted area,

$$A = \frac{1}{2} r^2(\theta - \sin\theta) \tag{3-21}$$

where A is the area of the segment, r is the pipe radius, and θ is the central angle (in radians) of the sector enclosing the segment. Substituting,

$$0.0311 = \frac{1}{2}\left(\frac{0.300}{2}\right)^2 (\theta - \sin\theta)$$
$$\theta = 2.95 \text{ radians by trial}$$

$$0.346 = \frac{1}{2}\left(\frac{1}{2}\right)^2 (\theta - \sin\theta)$$
$$\theta = 2.95 \text{ radians by trial}$$

The wetted perimeter is

$$P = r\theta = \frac{0.300}{2} \times 2.95 = 0.443\,\text{m}$$
$$R = A/P = 0.0311\,\text{m}^2/0.443\,\text{m} = 0.0702\,\text{m}$$

$$P = r\theta = \frac{1}{2}(2.95) = 1.48\,\text{ft}$$
$$R = A/P = 0.346\,\text{ft}^2/1.48\,\text{ft} = 0.234\ \text{ft}$$

Substituting in Equation 3-17a,

$$1.1\ \text{m/s} = \frac{1}{0.013}(0.0702)^{2/3}S^{1/2}$$

$$S = 0.0071\ \text{m/m}$$

Substituting in Equation 3-17b,

$$3.5\ \text{ft/s} = \frac{1.486}{0.013}(0.234)^{2/3}S^{1/2}$$

$$S = 0.0065\ \text{ft/ft}$$

The discrepancy is caused by inexact conversions of SI to U.S. units. The minimum slope from Ten-State Standards [1] is 0.0022 m/m.

Short Cuts

Most of the mathematics above can be avoided by means of specialized slide rules or computer programs such as **UnifCrit2.2** [26] or PARTFULL® [25]. Both programs are free. The former is available in U.S. Customary Units on the Internet. Using it to solve the problem, go to the bottom of the **UnifCrit2.2** worksheet and enter:

d = 1.00 ft. Press Enter
V = 3.50 ft/sec. Press Enter
Q = 1.21 cfs. Press Enter. In the group of cells at the right, note y = 0.45 ft

Go to the top of the worksheet, enter Escritt=0 (because Escritt's assumption is not used in this example). Press Enter.

Q = 1.21 cfs. Press Enter.
n = 0.013. Press Enter.
S = unknown as yet. Ignore value shown.
d = 1.00. Press Enter.

Under "Uniform Flow Depth," see that y_n is an erroneous value, so change unknown S until $y_n = 0.45$ ft.

S = 0.0065 ft/ft. Note Critical Flow Depth, $y_c = 0.46$ ft, meaning flow is slightly supercritical.

3-6. Energy in Pressurized Pipe Flow

Many problems in hydraulics can be solved by equating the energy at two points along a conduit. Within a closed system (one inlet and one outlet) the total energy at point 1 equals the total energy at point 2 plus any losses due to friction, as shown by Equation 3-5. A typical example is the Venturi meter in Figure 3-4.

From the principles presented in Section 3-1, the pressure at point 1, as measured by a gauge or a piezometer, is greater than at point 2 by the amount $\Delta h_a = \Delta(p/\gamma)$ if the Venturi meter is horizontal. The differential pressure can be measured by a differential pressure gauge (not shown), by a mercury manometer, or by an air manometer. Differential pressure gauges or transducers are costly and delicate but

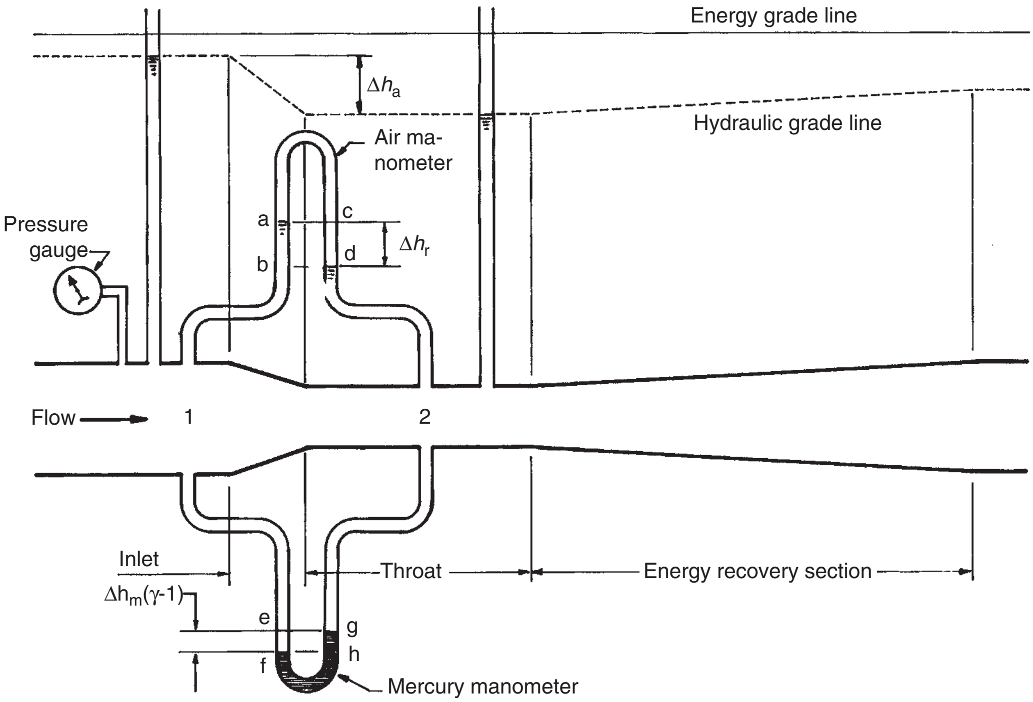


Figure 3-4. Venturi meter.

necessary if a remote reading is needed. Mercury manometers are still made, but mercury is poisonous and an accidental spill, sure to occur eventually, is nearly impossible to clean up. The air manometer is sensitive, accurate (if purged properly), cheap, and excellent even when homemade. The column of air, even if highly compressed, is insignificant in mass, so the manometer reading, Δh_r, equals the true value, Δh_a, with negligible error.

The connecting tubing to any pressure gauge or manometer must be purged of air to ensure accuracy. In the air manometer of Figure 3-4, for example, there must be no air except between points a and d, and this requires purging petcocks or bleeders at strategic locations. A means of introducing the required bubble of air at the top of the manometer is also required.

Example 3-3
Venturi Meter in a Pipe

Problem: Assume the inlet diameter in Figure 3-4 to be 254 mm (10 in.) and the throat diameter to be 152 mm (6 in.). The air manometer reads 305 mm (12 in.). Find the flow, the mercury manometer reading, and the differential gauge pressure.

Solution: Assume the friction loss between points 1 and 2 to be zero (nearly true). If the datum plane is the pipe centerline, Equation 3-5 becomes

$$h_1 + \frac{v_1^2}{2g} = h_2 + \frac{v_2^2}{2g}$$

SI Units | **U.S. Customary Units**

From Equation 3-2, velocity v_2 must be

$$v_2 = \left(\frac{254}{152}\right)^2 v_1 = 2.79 v_1$$

$$v_2 = \left(\frac{10}{6}\right)^2 = 2.78 v_1$$

SI Units

Substituting,

$$h_1 - h_2 = 0.305 = \frac{(2.79v_1)^2 - v_1^2}{2 \times 9.81}$$

$v_1 = 0.939$ m/s

$$Q = Av = \frac{(0.254)^2\pi}{4} \times 0.939$$
$$= 0.0476\ \text{m}^3/\text{s}$$

U.S. Customary Units

$$h_1 - h_2 = 1.0 = \frac{(2.78v_1)^2 - v_1^2}{2 \times 32.2}$$

$v_1 = 3.09$ ft/s

$$Q = Av = \frac{10^2\pi}{4} \times 3.09$$
$$= 1.69\ \text{ft}^3/\text{s}$$

In the mercury manometer of Figure 3-4, the pressure at f and h are equal (because the elevations are the same in a static continuum). The differential pressure between g and h equals the weight of mercury column gh minus the weight of water column ef. From the terms of the problem, the differential pressure between points 1 and 2 (and, hence, between points e and g) is 305 mm (12 in.) WC. The specific gravity of mercury (Hg) is 13.6 and of water is 1.0, so

SI Units:

$$305 = p_f - \overline{ef}(1.0) - [p_h - \overline{gh}(13.6)]$$

$p_f = p_h$ and $\overline{ef} = \overline{gh}$

$$\overline{gh} = 305/(13.6 - 1) = 24.2\ \text{mm Hg}$$

U.S. Customary Units:

$$12 = p_f - \overline{ef}(1.0) - [p_h - \overline{gh}(13.6)]$$

$p_f = p_h$ and $\overline{ef} = \overline{gh}$

$$\overline{gh} = 12/(13.6 - 1) = 0.952\ \text{in. Hg}$$

The differential gauge pressure can be calculated from the water column ab (Figure 3–4) in the air manometer by using the data for pressure in Table A-11.

$$p = 0.305 \times 9.79 = 2.99\ \text{kPa}$$

$$p = 1 \times 4.33 \times 10^{-1} = 0.433\ \text{lb/in.}^2$$

Using the mercury manometer and subtracting the water column ef from the mercury column gh,

$$p = 24.2 \times 1.33 \times 10^{-1} - 0.0242 \times 9.79$$
$$= 2.98\ \text{kPa}$$

$$p = 0.952 \times 4.91 \times 10^{-1} - (0.952/12) \times 4.33 \times 10^{-1} = 0.433\ \text{lb/in.}^2$$

3-7. Energy in Open Channel Flow

Water in open channels can flow in three regimes:

- Sub-critical or tranquil flow (waves can move upstream; example: sluggish stream);
- Critical flow with standing waves (hydraulically unique; example: flow over a broad crested weir);
- Super-critical or shooting flow (waves move only downstream; example: water flowing down a dam spillway).

Theory

To establish which of the three possible flow regimes occurs, consider Equation 3-4 and Figure 3-5 (and review Section 3-1, if necessary).

$$E_s = y + \frac{v^2}{2g} \qquad (3\text{-}4)$$

If the velocity is zero and the water is standing, the specific energy equals the depth, y, and plots as the straight line, oa, in Figure 3-5a. At a constant flow rate, however, E_s is the curved line represented by $y + v^2/2g$. In tranquil flow, the velocity head is small, but as y decreases, the velocity (to produce the same discharge) increases—and very rapidly below the tranquil zone. Hence, there is a critical depth, y_c, at which specific energy is minimum. To find it, replace v^2 with $(Q/A)^2$, differentiate Equation 3-4 with respect to y, and set the resulting expression equal to zero; thus,

$$\frac{dE_s}{dy} = 1 + \frac{Q^2}{2g}\left(-2A^{-3}\frac{dA}{dy}\right) = 0$$

For a prismatic channel of any shape, as in Figure 3-5b, $dA = bdy$. Substituting bdy for dA, canceling dy/dy, and collecting terms yields

$$\frac{Q^2}{g} = \frac{A^3}{b} \qquad (3\text{-}22)$$

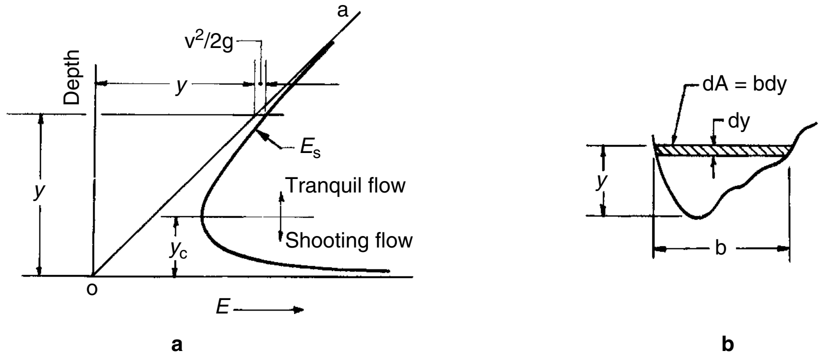


Figure 3-5. Energy in open channel flow. (a) Specific energy for constant flow rate; (b) channel cross-section.

at critical depth. Substituting $v_c = Q/A$ gives

$$\frac{v_c^2}{2g} = \frac{A}{2b} \tag{3-23}$$

at critical depth. Note that b is the width of the water surface—not the average width.

One illustration of the usefulness of Equations 3-22 and 3-23 is the application to open-channel Venturi flumes for measuring flow rate. Any obstruction in the channel that produces a short length of uniform flow with straight, level streamlines at critical depth can be a Venturi flume. One example is the Palmer–Bowlus meter (see Chapter 20), which usually has a trapezoidal throat shape as shown in Figure 20-18, but may be just a simple step as shown in Figure 3-6. (The Parshall flume is another popular measuring device, but it does not have straight, level streamlines at the critical section and is not really a Venturi flume.) The Palmer–Bowlus meter acts as a choke to increase the water depth upstream. Strange though it may seem, the water surface drops between sections b–b and c–c, then rises again from sections c–c to e–e so that y_e nearly equals y_a. If either (1) the step, (2) a reduced throat width, or (3) a combination of (1) and (2) produces critical depth for a suitably wide range of flow, the flume can be used to meter the flow rate with a basic uncertainty of only about plus or minus 3% [27]. Metering is easily accomplished because a wide range of dimensional changes between channel and throat—even a change in shape—can be used. For example, Palmer–Bowlus meters with trapezoidal throats are commonly installed in circular sewers.

The downstream depth, y_e, equals the depth if there were no choke and is found by means of an equation of uniform flow such as Manning's (Equation 3-17). If the total energy (Equation 3-3) in the throat is greater (say, 5% greater to allow for friction losses) than the energy in the channel without the choke or meter, critical depth theoretically occurs in the throat. Discharge can be calculated by measuring y_c, using it to find A and b, and substituting into Equation 3-22. It is customary, however, to measure y_a (upstream) because it is greater than y_c and, hence, the accuracy is better. Methods for obtaining rating curves for y_a are given in Example 3-4. Calculated rating curves give results within about 5% of the observed flow, and therefore Palmer–Bowlus meters, if correctly installed, are sufficiently accurate for measuring wastewater flow rates without the need for field calibration.

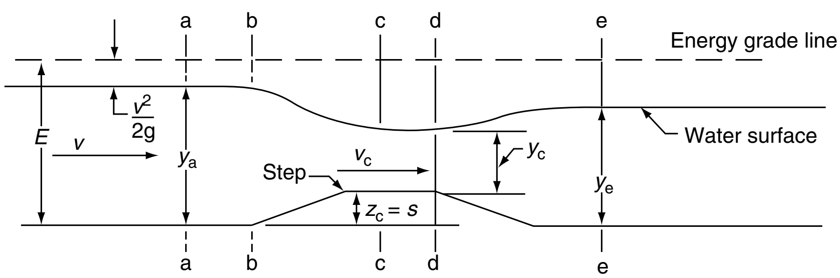


Figure 3-6. Longitudinal section through a Venturi flume in an open channel.

Example 3-4
Venturi Flume in a Channel

Problem: A smooth, rectangular concrete channel is 610 mm (2 ft) wide and carries a flow of $0.170\,\mathrm{m^3/s}$ ($6\,\mathrm{ft^3/s}$) at a slope of 0.00124 m/m. A rectangular Venturi flume with a throat width of 457 mm (1.5 ft) is installed with its step 76 mm (3 in.) above the channel invert. Calculate (1) the depth of flow in the throat and (2) the depth in the channel in front of (upstream from) the flume; (3) compare that depth with the depth far upstream; and (4) find whether critical depth truly occurs in the measuring flume.

Solution: (1) Assume critical depth occurs in the throat and find the critical depth from Equation 3-22.

SI Units	**U.S. Customary Units**
$\frac{0.170^2}{9.81} = \frac{A^3}{0.457},\ A = 0.110\,\mathrm{m}^2$	$\frac{6^2}{32.2} = \frac{A^3}{1.5},\ A = 1.19\,\mathrm{ft}^2$
$y_c = \frac{A}{b} = \frac{0.110}{0.457} = 0.241\,\mathrm{m}$	$y_c = \frac{A}{b} = \frac{1.119}{1.5} = 0.793\,\mathrm{ft}$

From Equation 3-2, the throat velocity is

$v_c = Q/A = 0.170/0.110 = 1.55\,\mathrm{m/s}$	$v_c = Q/A = 6/1.19 = 5.04\,\mathrm{ft/s}$

(2) To find conditions in front of the measuring flume, use Bernoulli's equation (Equation 3-5) and assume h_f is zero (which is practically true).

$$z_a + y_a + \frac{v_c^2}{2g} = z_c + y_c + \frac{v_c^2}{2g} + 0 \tag{3-5}$$

Note that the velocity in the channel is

$Q/A = Q/by = (0.170/0.61)y_a = 0.279/y_a$	$Q/A = Q/by = (6/2)y_a = 3/y_a$

so Bernoulli's equation becomes

$0 + y_a + \frac{(0.279/y_a)^2}{2 \times 9.81} = 0.076 + 0.241 + \frac{1.55^2}{2 \times 9.81}$	$0 + y_a + \frac{(3/y_a)^2}{2 \times 32.2} = \frac{3}{12} + 0.792 + \frac{5.04^2}{2 \times 32.2}$

By trial,

$y_a = 0.416\,\mathrm{m}$	$y_a = 1.36\,\mathrm{ft}$

So the water surface falls

$y_a - (y_c + \mathrm{step}) =$	$y_a - (y_c + \mathrm{step}) =$
$0.416 - (0.241 + 0.76) = 0.099\,\mathrm{m}$	$1.36 - (0.793 + 0.25) = 0.317\,\mathrm{ft}$

(3) Use Equation 3-17 (Manning's) to find the depth far upstream (or downstream). Choose $n = 0.011$ for smooth concrete.

$$v = \frac{1}{n} R^{2/3} S^{1/2}$$
$$v = Q/A = (0.170/0.610)y = 0.279/y$$
$$R = A/P = 0.610y/(2y + 0.610)$$
$$\frac{0.279}{y} = \frac{1}{0.011}\left(\frac{0.610y}{2y + 0.610}\right)^{2/3} \times (0.00124)^{1/2}$$
$$y = 0.305\,\text{m by trial}$$

$$v = \frac{1.486}{n} R^{2/3} S^{1/2}$$
$$v = Q/A = 6/2y = 3y$$
$$R = A/P = 2y/(2y + 2)$$
$$\frac{3}{y} = \frac{1.486}{0.011}\left(\frac{2y}{2y + 2}\right)^{2/3} \times (0.00124)^{1/2}$$
$$y = 1.00\ \text{ft by trial}$$

So the measuring flume acts as a choke to raise the water surface from 0.305 m to 0.416 m (1.00 ft to 1.36 ft). The ‘‘backwater’’ created by the choke extends far upstream.

(4) If the total energy in the throat is greater (say, 5% greater to allow for friction losses) than the energy in the channel without the choke, critical depth must theoretically occur in the throat. Without the choke, the total energy (referenced to the channel bottom) would be, from Equation 3-3,

$$E = z + y + \frac{v^2}{2g}$$
$$E = 0 + 0.305 + \frac{[0.170/(0.610 \times 0.305)]^2}{2 \times 9.81}$$
$$E = 0.348\,\text{m}$$

$$E = z + y + \frac{v^2}{2g}$$
$$E = 0 + 1 + \frac{(6/2)^2}{2 \times 32.2}$$
$$E = 1.140\,\text{ft}$$

With the choke, the total energy in the throat (again referenced to the invert of the channel) is

$$E = 0.076 + 0.241 + \frac{(1.55)^2}{2 \times 9.81}$$
$$E = 0.438\,\text{m}$$

$$E = 0.25 + 0.793 + \frac{(5.04)^2}{2 \times 32.2}$$
$$E = 1.44\,\text{ft}$$

which is 26% greater than the energy without the choke, so critical depth does occur.

Field Determination of Critical Flow

The very uniqueness of critical flow makes it easy to determine if the metering flume is registering properly. A drowned meter usually has a nearly level surface from inlet to outlet. If a hydraulic jump occurs at the downstream end, the flow in the throat is certain to be critical. But usually a jump does not occur, so another criterion is to compare the depths of flow upstream (at Section a–a in Figure 3-6) and downstream (at Section e–e). If the downstream depth is no more than 85% of the upstream depth, the flow in the throat is critical [28]. In Example 3-4, the upstream depth is 0.416 m and the downstream depth is 0.305 m or 73% of the upstream depth—another demonstration that critical flow does occur.

Arredi Diagram

Trial solutions are unsatisfactory for more than one or two points (unless programmed on a computer). To make a rating curve of head versus flow rate, an Arredi diagram [29] is preferable. Select several values of discharge (Q_1, Q_2, etc.) covering an appropriate range from small to large values. For each Q, plot a curve of throat area versus $v^2/2g$ as shown in the upper half of Figure 3-7. These curves are the same for all Venturi flumes regardless of size or shape. Now plot a curve of $v_c^2/2g$ versus throat area (curve OA) from Equation 3-22 in the upper half of Figure 3-7. This curve is specific for a particular throat. The points of intersections with the Q curves give velocity head energy for each value of Q.

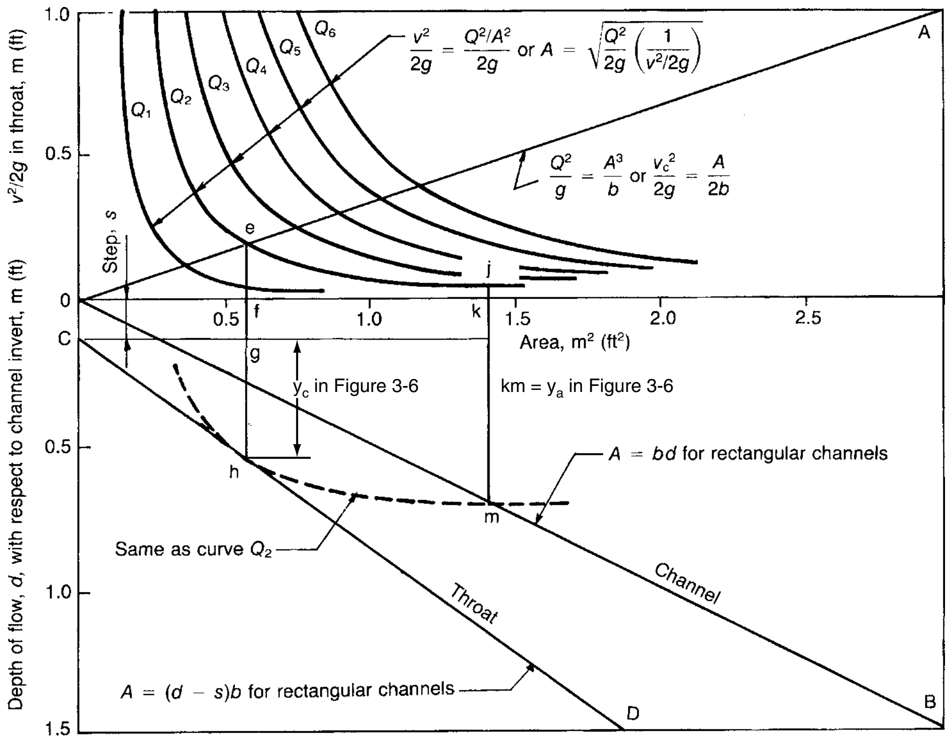


Figure 3-7. Arredi diagram.

In the lower half of the figure, plot (to the same scale) depth, d, versus area for the channel (curve OB) and plot depth versus throat area referred to the channel invert (curve CD). Note that OC is the height of the step, s, so throat depth is $d - s = y_c$.

Consider point e in Figure 3-7. The total energy in the throat is eh, because ef is $v^2/2g$, fg is s, and gh is y_c. Set a pair of dividers to length eh and follow the curve of Q_2 to the right until the dividers intersect curve OB so that jm equals eh. The depth indicated by km is the upstream channel depth, y_a, for Q_2.

To develop a rating curve, repeat this procedure for several values of Q and construct a graph of y_a (as the ordinate) versus Q (as the abscissa). For shapes such as trapezoidal throats in semicircular flumes, lines OA, OB, and CD are curved.

The difference between these theoretical rating curves and discharges measured in the laboratory is less than about 4% of the maximum flow rate. In the field, the uncertainty of measurement would not exceed 4 or 5% if the flume is installed with reasonable care. The great advantage of the Palmer–Bowlus meter is that it does not require physical calibration if the aforementioned errors are acceptable.

3-8. Unbalanced Hydraulic Forces

Adequate pipe support is important to protect flanges, bolts, and the pipe itself from destructive forces due to the weight of pipe and water or due to a change in fluid velocity. (Earthquake, vibration, and water hammer may also be destructive.) Valve bodies and pump casings are especially vulnerable, and undue strain can cause binding long before breakage occurs. Straight pipe needs support in the vertical plane (to resist gravity) and in the horizontal plane (to resist earthquake). Hangers or piers should be spaced to minimize vibration (see Chapter 22). Bends, tees, and wyes produce unbalanced thrusts in the plane of the fitting and, hence, require substantial anchors that must often resist large horizontal forces without appreciable deformation.

The purpose of the following example is to show how to compute the unbalanced forces. Review the fundamentals of momentum in Section 3-1, if necessary.

Example 3-5
Unbalanced Forces on a Wye

Problem: Compute the forces due to water velocity, static pressure, and water hammer on the horizontal steel wye shown in Figure 3-8 and determine whether an anchor is required.
Solution: Force due to velocity and pressure. In the force system of Figure 3-8, positive vectors act right or up. The net forces ($\mathbf{R}_x$ or $\mathbf{R}_y$) to produce the accelerations are assumed to

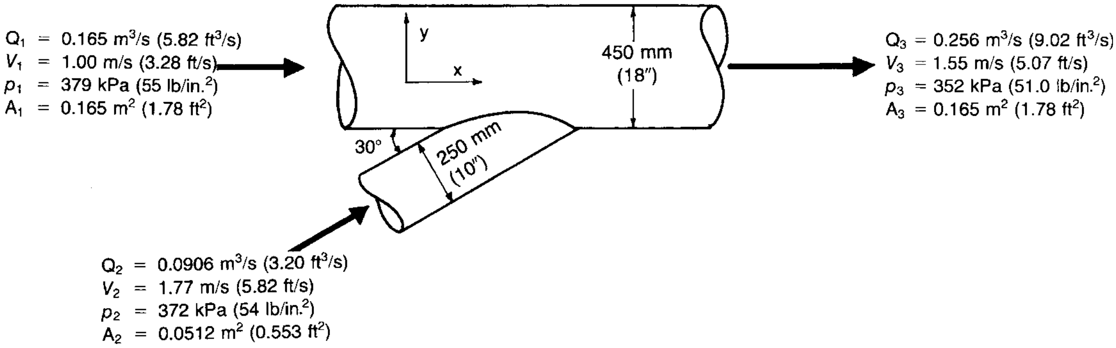


Figure 3-8. Flow in a wye.

be positive (and, therefore, act up or right), so a negative answer indicates a force acting opposite (down or left). The x component of Equation 3-8 is

$$P_1A_1 + P_2A_2 \cos 30° - P_3A_3 + \mathbf{R}_x = \rho Q_3 v_3 - \rho(Q_1 v_1 - Q_2 v_2 \cos 30°)$$

The y component is

$$0 + P_2A_2 \sin 30° + 0 + \mathbf{R}_y = 0 - \rho Q_2 v_2 \sin 30° + 0$$

Calculations for these terms are as follows.

SI Units	**U.S. Customary Units**
p_1A_1: $379 \times 0.165 = 62.5\,\text{kN}$	$55.0 \times 1.78 \times 144 = 14{,}100\,\text{lb}$
p_2A_2: $372 \times 0.0512 = 19.0\,\text{kN}$	$53.9 \times 0.553 \times 144 = 4290\,\text{lb}$
p_3A_3: $352 \times 0.164 = 58.1\,\text{kN}$	$51.0 \times 1.78 \times 144 = 13{,}100\,\text{lb}$
$\rho Q_1 v_1$: $1000 \times 0.165 \times 1.00 = 0.165\,\text{kN}$	$1.94 \times 5.82 \times 3.28 = 37.0\,\text{lb}$
$\rho Q_2 v_2$: $1000 \times 0.0906 \times 1.77 = 0.160\,\text{kN}$	$1.94 \times 3.20 \times 5.81 = 36.1\,\text{lb}$
$\rho Q_3 v_3$: $1000 \times 0.256 \times 1.55 = 0.397\,\text{kN}$	$1.94 \times 9.02 \times 5.07 = 88.7\,\text{lb}$

The x component is

$62.5 + 19.0 \cos 30° - 58.1 + \mathbf{R}_x$	$14.100 + 4290 \cos 30° - 13{,}000 + \mathbf{R}_x$
$= 0.397 - (0.165 + 0.160 \cos 30°)$	$= 88.7 - (37.0 - 36.1 \cos 30°)$
$\mathbf{R}_x = -20.8\,\text{kN} = 20.8\,\text{kN}$ (acting left)	$\mathbf{R}_x = -4630\,\text{lb} = 4630\,\text{lb}$ (acting left)

SI Units	**U.S. Customary Units**

The y component is

SI Units	U.S. Customary Units
$0 + 19.0 \sin 30° + 0 + \mathbf{R}_y = 0 - 0.160 \sin 30°$	$0 + 4290 \sin 30° + 0 + \mathbf{R}_y = 0 - 36.1 \sin 30° + 0$
$\mathbf{R}_y = -9.58\,\text{kN} = 9.58\,\text{kN}$ (acting down)	$\mathbf{R}_y = -2160\,\text{lb} = 2160\,\text{lb}$ (acting down)

Force due to water hammer. As shown in Chapter 6, water hammer pressure for the instantaneous (worst-case) closure of a downstream valve is

$$\Delta H = \frac{a \Delta v}{g}$$

where ΔH is pressure rise in meters (feet), a is wave velocity in the pipeline in meters per second (feet per second), g is the acceleration due to gravity [9.81 m/s^2 (32.2 ft/s^2)], and v is the change in velocity in meters per second (feet per second), which is 1.55 m/s (5.07 ft/s) in this problem. Assuming some air is present in the steel pipe (see Equation 6-4 and Table 6-2), a is about 981 m/s (3220 ft/s), so

SI Units	U.S. Customary Units
$\Delta H = \frac{981 \times 1.55}{9.81} = 155\,\text{m}$	$\Delta H = \frac{3220 \times 5.07}{32.2} = 507\,\text{ft}$

and the pressure from Appendix Table A-11 is

SI Units	U.S. Customary Units
$\Delta p = 155 \times 9.79 = 1520\,\text{kPa}$	$\Delta p = 507 \times 0.433 = 220\,\text{lb}$

This pressure increment can be assumed uniform throughout the wye, so the forces are

SI Units	U.S. Customary Units
$p_1 A_1$: $(379 + 1520)0.165 = 313\,\text{kN}$	$(55.0 + 220)1.78 \times 144 = 70{,}500\,\text{lb}$
$p_2 A_2$: $(372 + 1520)0.0512 = 96.8\,\text{kN}$	$(53.9 + 220)0.553 \times 144 = 21{,}800\,\text{lb}$
$p_3 A_3$: $(352 + 1520)0.165 = 309\,\text{kN}$	$(51.0 + 220)1.78 \times 144 = 69{,}500\,\text{lb}$

Assume that the $\rho Q v$ terms remain essentially unchanged because the water may still be moving when the shock wave arrives. Reinforcement by reflected waves could possibly occur in a complex situation, but the assumptions here are reasonable for most applications.

SI Units	U.S. Customary Units
$313 + 96.8 \cos 30° - 309 + \mathbf{R}_x = 0.397 - (0.165 + 0.160 \cos 30°)$	$70{,}500 + 21{,}800 \cos 30° - 69{,}500 + \mathbf{R}_x = 88.7 - (37.0 + 36.1 \cos 30°)$
$\mathbf{R}_x = -87.7\,\text{kN} = 87.7\,\text{kN}$ (acting left)	$\mathbf{R}_x = -19{,}800\,\text{lb} = 19{,}800\,\text{lb}$ (acting left)
$0 + 96.8 \sin 30° + 0 + R_y = 0 - 0.160 \sin 30° + 0$	$0 + 21{,}800 \sin 30° + 0 + R_y = 0 - 36.1 \sin 30° + 0$
$\mathbf{R}_y = -48.5\,\text{kN} = 48.5\,\text{kN}$ (acting down)	$\mathbf{R}_y = -10{,}900\,\text{lb} = 10{,}900\,\text{lb}$ (acting down)

and the forces acting on the anchor bolts are equal and opposite (i.e., same as force on water and opposite to forces on pipe).

The wye need not be anchored if the piping is able to resist the final resultant, which is

SI Units

$$R = \sqrt{(87.7)^2 + (485)^2} = 100\,\text{kN}$$

U.S. Customary Units

$$R = \sqrt{(19{,}800)^2 + (10{,}900)^2} = 22{,}600\,\text{lb}$$

at a reasonable deflection (which should be calculated). Do not allow this resultant force to be transferred into valve bodies or pump casings, because they are not designed to resist such forces and are likely to bind or break. Anchorage for unbalanced forces is always required at flexible couplings.

3-9. Field Measurement of Pipe Diameter and Friction Coefficient

Obstructions such as trash, valves partially closed through carelessness, reduction in pipe diameter due to mineral deposits, and the increase of roughness due to tuberculation or unevenness of mineral deposits are all manifested by an apparent decrease in the Hazen–Williams coefficient, C (or by an increase in the Darcy–Weisbach coefficient, f). The flow decreases markedly and the discharge pressure increases somewhat. Pumps run longer and consume more power unnecessarily—often at significant cost.

Field measurements to determine pipe roughness are needed whenever

- a new pump is to be used with an existing pipeline;
- the pumping station is to be remodeled;
- any expensive change is contemplated that involves existing pipe; and
- the capacity of the pump has declined without excessive impeller wear.

Other reasons for determining roughness include

- timing for cleaning and relining;
- calibration of a network model; and
- plans to increase the flow rate.

Using old plans and specifications as data for a new design without a confirming field survey is dangerous. According to some experienced consultants, the majority of record (or ''as-built'') drawings contain serious errors, such as wrong dimensions, alterations never recorded, significant mistakes in the static head, and an inability to achieve the design flow. Without a survey, trusting the plans is thus likely to lead to more blunders, embarrassment to both engineer and client, and an added, unnecessary cost for rectification.

The most dramatic result of a field survey is to find excessive roughness (low C value), which usually indicates a force main partially plugged with debris or heavily coated with grease, mineral deposits, or tuberculation. Mineral deposits over 50 mm (2 in.) in thickness were developed in one transmission main within two months [30]. Valves have been found partially closed, and improper repairs or additions long forgotten have collected sticks and rags. Camera inspection or forcing ''pigs'' through the line can disclose gross problems—once the need for such methods has been established by a field survey of pipe roughness. (A self-destructing pig that need not be recovered can be a loaf of hard bread, a block of ice, or a bag of ice cubes.)

Roughness is determined by measuring the friction headloss between two points separated by a known length of pipe of known internal diameter at a known rate of flow. Several runs (at least three) should be made, preferably at different flow rates, to obtain an average roughness coefficient. It is wise to demand two independent determinations of the roughness coefficient because either blunders or excessive errors can be so costly.

Pressure Gauging

Headloss can be measured with (usually) sufficient accuracy by installing a suitable pressure gauge downstream from the pump to determine the pressure drop to a free surface. The static head can be determined from a field survey or by halting the flow and reading the pressure gauge. To measure the friction headloss with enough accuracy, a ''suitable'' pressure gauge should have at least a 150-mm (6-in.) face, be accurate to 0.25%, and have a maximum pressure reading of not much more than 150% of the expected pressure; it should also have been calibrated recently.

Pressure gauges can be installed (without interrupting flow) by attaching them to corporation cocks, which in turn can be installed under pressure by the water utility for a nominal cost if the pipe is exposed. A short piece of flared copper tubing leading to pipe threads to match the gauge completes the installation. Some corporation cocks can be obtained with standard pipe threads or with standard pipe thread adapters (see Figure 20-6 for a good gauge assembly).

To obtain a static pressure reading, stop the flow in the force main and read the static gauge pressure, p_s. (It can also be calculated if the difference in

elevation between the gauge and the free water surface is known.) After the flow is resumed, the gauge pressure is p_t and the pressure loss due only to friction head loss is

$$\Delta p = p_t - p_s \tag{3-24}$$

Convert differential pressure, Δp, to differential head, Δh, by using Table A-11, then calculate friction loss, h_f, in the usual units, meters per 1000 meters (or feet per 1000 feet), as

$$h_f = 1000\,\Delta h/L \tag{3-25}$$

where L is the length of pipe between gauge and free water surface in meters (feet).

If the transmission main branches or the diameter changes or if there is no free water surface, pressure must be obtained at a second point. If two pressure gauges are used, one at point A and another at point B, Equation 3-24 is modified to

$$\Delta p_{A-B} = (p_t - p_s)_A - (p_t - p_s)_B \tag{3-26}$$

Exposing buried pipe for the attachment of pressure gauges is very expensive, so connect the second gauge to a fire hydrant or a service line, if possible.

Unless the two pressure gauges are far apart, the error (even with gauges accurate to 0.25%) can be unacceptable. An alternative is to connect hoses or tubing at points A and B and lead them to a differential gauge or an air manometer such as that shown in Figure 3-4. The hoses must be purged, and a snubber may be needed to prevent rapid fluctuations.

Differential Pressure

Differential pressures must be measured when pitot tubes, Venturi meters, or orifice meters are used to determine flow. Usually, differential pressures range from 2 to 500 mm (0.1 to 20 in.) of water column. Pressure in the pipe may be 200 to 10,000 times greater, so it is futile to measure the differential pressure by trying to subtract gauge readings at the two pressure taps. Differential gauges are both expensive and delicate. Manometers that use mercury, carbon tetrachloride, or oil are, according to Murphy's Law, almost certain to blow out sooner or later and contaminate the pipeline—a serious disaster in a public water supply.

An inexpensive substitute is the air-filled, upside-down manometer in Figure 3-4. If the air blows out it can easily be replaced by a tire pump or a pressure tank, so a tire valve should be installed at the top. Petcocks to bleed air from the tubing leading to the manometer are also needed. Glass tubing is fragile, but plastic tubing is rugged. A satisfactory homemade device equipped with (1) 6-mm ($\frac{1}{4}$-in.) valves or petcocks, (2) a method for attaching it to the copper tubing from the meter [such as 6-mm ($\frac{1}{4}$-in.) heavy rubber tubing with hose clamps], and (3) a meter stick can be made for a few dollars from materials available at hardware stores or laboratories. Commercial models are available. Differential pressure measurement is discussed thoroughly by Walski [31, 32].

Pipe Diameter

The internal pipe diameter (ID) must be known with any of these methods, and the true ID must be known with reasonable accuracy to use a pitot tube, strap-on meter, or any device that measures only velocity in the pipe. Wall thicknesses of pipe are increased by decreasing the ID, so OD is no indicator of ID. Furthermore, coatings (with unreliable thicknesses) and mineral deposits reduce the ID. Hence, do not trust the plans or specifications. One reliable means is to remove a section of pipe and measure the ID, but that can be exorbitantly costly, and it gives only local data. Note that mineral deposits may not be uniform. Other nondestructive ways of measuring pipe thickness may exist or be developed, but thickness transducers (at the time of this writing) are inaccurate for pipe with a mortar lining or mineral deposits.

Calculation of Average Pipe Diameter

Equation 3-2 can be used to find the average pipe diameter if both average velocity and flow rate are known. A tracer can be used to find the average velocity as follows:

- Isolate a section of the pipe and use a fire hydrant to establish a given flow rate.
- Pump a slug of tracer at one point through a service connection.
- Measure the average time for the tracer to pass a downstream point (actually, the time for half of the tracer to pass the point). Obtain samples through a second service connection.
- Measure the length of pipe between the two points and calculate the velocity with due allowance for the residence time in the service connections.

- Obtain the flow rate with a calibrated flowmeter at the downstream fire hydrant corrected for the flow rate from the second service connection.
- Use Equation 3-2 to calculate the average cross-sectional area of the pipe.
- Use a differential pressure gauge to find the head-loss and compute H−W *C* from Equation 3-9.

Tracers can also be used to determine the flow rate independently of the velocity, as discussed at the end of this section. (Other methods discussed in the following subsection could also be used.) With very careful work, the average ID of a pipe can be found to about the nearest 2% of the pipe diameter.

Flow Rate Measurement

Flow rate, in conjunction with friction headloss, is needed for constructing the existing system curve. Flow rate can be measured in a variety of ways, generally dictated by the specific situations encountered.

- Obtain volumetric measurements using a stop-watch and the wet well, the clear well, or the reservoir. Always try to measure flow volumetrically by using any practical means available.
- Use the existing flowmeter in the station, but be suspicious if the calibration is not recent. Check the read-out system by manual readings, if possible, or by comparison with volumetric measurements. As a rule, errors in most flowmeter readings exceed 10%, and 50 to 200% errors are not uncommon [33].
- Temporarily install a Palmer–Bowlus flume in a manhole (preferably downstream) where there is open channel flow.
- Use the manufacturer's pump curves as a convenient check to confirm any of the other methods. When carefully done, the uncertainty of flow rate measurement can be kept to ±7%, if there is reasonable surety that (1) no wear has occurred, (2) the impeller has not been replaced with a significantly different one, and (3) the pump curves were developed from good data under conditions that can be reproduced in the field. Some consultants do not trust pump curves, but others have found them reliable.
- Install a temporary flowmeter (preferably in a long, straight section of pipeline or on a fire hydrant), such as a multiport pitot tube or magnetic flow rod, or use a strap-on Doppler or ultrasonic meter. A traverse with a single port pitot tube is a useful substitute, but traversing is tedious, and within, say, 15 pipe diameters downstream from a discontinuity (e.g., an elbow), accuracy suffers because of unsteady flow lines.
- Use tracers. The use of tracers provides the highest accuracy and reliability, but it requires either expensive equipment or a consulting specialist. Errors in repeatability can, with care, be kept within less than 2% of the particular flow rate being tested.

All of these techniques require resourceful, knowledgeable personnel who are either experienced or aggressive enough to learn how to make the selected tests with accuracy. If needed, pump consultants, some consulting engineers, or university staff members in environmental, civil, mechanical, or chemical engineering may be able to assist.

Volumetric Measurement

Volumetric measurement, the method of choice, is often the most convenient and reliable because many (perhaps most) pumping stations have associated wet wells, clear wells, or storage tanks. Volume computed from careful measurements can be used in conjunction with timed drawdown to determine flow rate with precision. Again, beware of trusting the construction drawings for calculating volumes; always make at least two confirming field measurements of the basin size. Under field conditions, it is difficult to time an event to the nearest second, so the running time should be at least 2 min; 5 min is better.

If the influent flow into a wet well or clear well cannot be halted for the duration of a run, a reasonable approximation of volume pumped can be made by first measuring inflow with the pumps off and then accounting for inflow while the pumps run. Because wastewater inflow varies, several runs are necessary to get a reasonable, average result. Because the volume of a sewer pipe is not only very large but also indeterminate, be sure to keep the upper wet well level below the influent sewer pipe.

One method for timing wet well drawdown is to tie two short objects to a string at a convenient, measured distance apart. Attach a weight (a rock or brick will do) to the end of the string and suspend it in the wet well. As the water rises (or falls) the meniscus at the objects can usually be clearly seen by using a flashlight [34]. If visibility is poor, a pair of electrical contacts and a battery to operate low-voltage lights can be used.

Permanent Flowmeters

When present, a flowmeter may be the most convenient way to measure flow rate, but take care to ensure

that the meter itself is not a source of error. Manufacturers' claims of accuracy are rarely realized in the field, and a surprising number of installations have shown excessive errors—usually more than 10%, often more than 20%, and occasionally as much as 200% [33]. (For example, the influent flowmeter at a water treatment plant in Montana was about 100% in error for nearly a year [35].) Reasons for poor accuracy include improper installation, clogged pressure taps, improper adjustment of the read-out system (mechanical or electrical), non-ideal hydraulic conditions, incorrect flow coefficients, incorrect flowmeter gearing, and corrosion or moisture in electrical circuits.

The electronic read-out system of any differential pressure device, such as a Venturi or orifice, can sometimes be checked by temporarily installing a differential head device (differential pressure gauge or air manometer) that can be read manually, but it checks only the differential pressure and not the meter itself.

Temporary Meters in Open Channels

Wastewater force mains and some water pumping stations take fluid from (or discharge into) open channels. Palmer–Bowlus flumes can be placed in sewer pipes at manholes. Weirs and Parshall or Venturi flumes can be installed in ditches. The water level can be read with a point gauge (equipped for convenience with a battery-operated electric contactor) in the upstream channel. Floats, battery-powered "dipper" needles, and sonic meters are also available.

Pump Curves

The use of pump curves is a convenient check on other methods. Pressure gauges are needed on both the suction and the discharge spools (but not in the pump casing). Worn impellers must not be used. (Of course, it is even better if the pump has been recently tested and certified test curves are available.) The accuracy of flow rate determination depends on knowing the conditions used by the manufacturer to derive the pump curves and avoiding operation where the pump curve is too flat.

Begin by operating the pump against a closed discharge valve and adjust (by calculation) the manufacturer's impeller head-flow rate (H-Q) curve up or down to correspond to the differential pressure gauge reading across the pump at shutoff. If possible, operate the system at a point where the pump H-Q curve is reasonably steep by running only one of a bank of pumps. Calculate the flow rate from the adjusted pump curve at the proper differential pressure across the pump.

This technique and the probable errors associated with each component of the measurement are thoroughly discussed in DIN [36]. With care, the uncertainty of flow rate measurement can be kept within $\pm 7\%$ and, frequently, to within 5% [37].

Temporary Flowmeters in Pipelines

The recommendations for installing temporary flowmeters are the same as those for permanent ones. The manufacturer's advice must be followed, especially with regard to the required lengths of straight pipe before and after the meter. If there is insufficient length of straight pipe (which is often true in the cramped quarters of pumping stations), consider a meter installation in the yard piping.

Assuming a satisfactory location for a meter, several available types are compared in Table 3-1 (see also Chapter 20).

Tracers

The purpose of this subsection is to introduce a little-known, little-understood, but versatile and accurate method for measuring flow rate and for calibrating flowmeters in place. Errors of less than 5% of the measured flow rate are easily achievable, and with care, the errors can be reduced to less than 2%. There are few places (e.g., a closed or recirculating system) where the use of tracers would be inappropriate. The method is useful for measuring flows in manufacturing plants, open channels, and pipelines of any size or of varying size. Engineers who have some background in wet chemistry can learn to use tracers in a couple of days.

Tracers can be used in three ways:

- Tracer velocity or time-of-travel
- Slug addition
- Constant rate injection.

In the first method, a concentrated dose of tracer is dumped into an upstream point, and the time required for the peak concentration to pass a downstream point gives the average velocity. Salt is frequently used for the tracer and its concentration is measured with a portable conductivity meter. If the flow is steady and the channel is prismatic, an approximate flow rate can be determined by measuring

Table 3-1. Temporary Flowmeters

Type	Advantages	Disadvantages
Flow nozzle	Locate anywhere along pipe; only one pressure gauge required if at end of pipe.	Expensive to dismantle pipe.
Pitot tube, single port	Insert through corporation cock. Can install under pressure; inexpensive in exposed pipe; one pitot tube can serve all stations.	Unsuitable for raw wastewater. Must take many readings to find average flow [38]; flow must be steady while survey is in progress; expensive for buried pipe.
Pitot tube, multiport (integrating)	Single reading gives average flow; can follow changing flow; good as a permanent meter.	Suitable for only a single ID. Unsuitable for raw wastewater.
Strap-on meters	Can apply to outside of pipe. Installation is inexpensive; can be used permanently.	Meter itself is expensive. Readings suspect without independent check; should calibrate meter on pipe.
Magnetic	For raw wastewater or clean water.	
Time of flight	Accurate for clean water but only if installed far enough (2 blocks) downstream from the pump so that minute vapor bubbles are dissipated.	Not for raw wastewater or dirty water.
Doppler	For raw wastewater and dirty water.	Not for clean water.
Magnetic tube type	Same as multiport pitot tube.	Expensive. Requires corporation cock.
Orifice plate	Same as for flow nozzle; for grit-laden water, use segmental orifice plate to provide smooth invert.	Unsuited for rages and stringy material.
Elbow meter	Inexpensive; can construct in the field using any elbow; fair accuracy but good precision (repeatability); stable operation; unobstructed flow.	Must calibrate in place; low differential pressure (use air manometer). See Miller [39].

the cross-sectional area. The accuracy is questionable. Errors may well exceed 15%.

The next two methods require precision and costly equipment. The equipment can sometimes be rented. Some organizations have specialists who can contract the work. The general requirements for the use of tracers are:

- A precise method for introducing the tracer.
- Enough length and/or turbulence so that mixing is complete—a critical requirement.
- A tracer substance that is innocuous (nonhazardous and nontoxic), stable, and not significantly adsorbed on walls or particles.
- A tracer substance that occurs naturally only in very low or insignificant concentrations (low background).
- A means for obtaining small samples downstream.
- A method of chemical analysis that is sensitive, accurate, and free from significant interferences.
- Preliminary testing to (1) establish accuracy; (2) obtain calibration curves for the tracer diluted with the water or wastewater; (3) quantify possible interferences, decay, or sorption on pipe walls or particles so that corrections can be made; and (4) determine the suitability of the means selected to preserve samples for shipment and analysis.

In the slug-addition method, a known mass of tracer is introduced at some upstream point in a manner to promote good mixing. At a point well downstream after thorough mixing, a series of grab samples is obtained and analyzed. Plotting the data yields a curve like that labeled "abd" in Figure 3-9. If the flow rate is constant over the short time needed for all the tracer to pass the downstream point, then tracer introduced equals tracer past the downstream point.

$$M = vC = Q\int_{t_1}^{t_2} (c - c_o)dt \qquad (3\text{-}27)$$

where M is the mass of tracer, v is the small volume of tracer at high concentration C, Q is the uniform flow

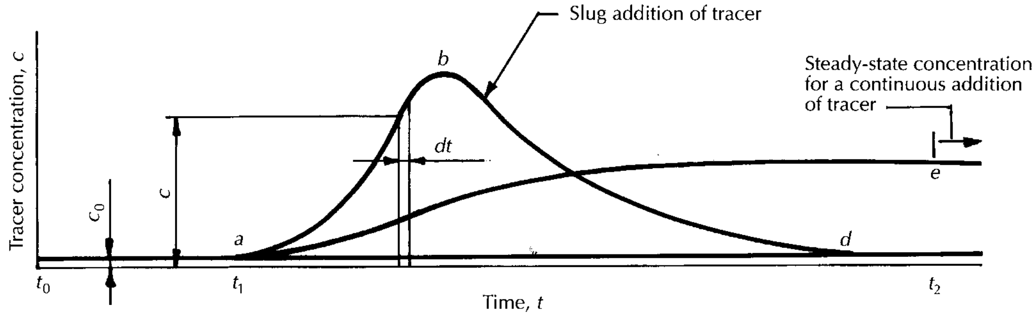


Figure 3-9. Downstream concentration distribution of tracer resulting from: (a) slug addition of tracer (curve abd) and (b) continuous constant-rate injection of tracer (curve ae) at an upstream point.

rate, c is the measured concentration of tracer downstream, and c_0 is "blank" or the background reading of water without tracer. Under ideal conditions, the error in determining Q can be less than 2%. In Figure 3-9, blank, c_0, is shown to be large for clarity. Actually, the peak concentration should be at least 100 times greater than blank.

Accurate results require a sufficient number of samples to identify the time of first appearance and the time of complete passage and to define the shape of the curve clearly. It is prudent to take many samples before the expectation of first appearance and many more samples after expected complete passage. The tracer concentrations should be determined with the same care as described below for the constant-rate injection method.

The average pipe diameter can be found by the slug addition method. Velocity is found by measuring the time when half the tracer has passed the sampling point and measuring the distance between that point and the point of injection. Equation 3-27 defines Q, and the cross-sectional area is determined from Equation 3-2. See, Wright, and Damewood Nevins [40].

In the more accurate continuous constant-rate injection method, a small stream of concentrated tracer is added at the upstream point. If possible, the concentration of tracer past the downstream point should be monitored until uniform instrumental readings, as indicated by point "e" in Figure 3-9, are obtained. If monitoring is impractical, estimate the waiting period for the axial dispersion to stabilize and for a uniform concentration of tracer to invade all water in pockets (such as valve bonnets). Then take a series of grab samples for precise analysis. The remainder of this discussion applies to continuous constant-rate injection, although much of it would apply to slug addition as well.

If the objective is to calibrate a flowmeter, obtain simultaneous readings of the flowmeter in engineering units and in raw sensor analog output. The latter can be measured by means of a digital meter with much greater accuracy than a visual reading of the flow indicator, and the multiplier and offset coefficients to convert sensor signal to flow can be calculated directly. Flowmeter signals typically fluctuate as much as 10% over short periods of time due to turbulence at the sensor and, therefore, many data points are needed to establish a norm. If practical, obtain pump speed and pump motor current draw to compare pump performance with predicted performance and to check the reasonableness of the test results.

If the objective is to calibrate a flowmeter in an open channel, it is useful to obtain simultaneous depth measurements manually for comparison with the flowmeter measurements and to establish an independent flow rating curve. If a flow sensing element is replaced, the manual rating curve allows a quick check of the meter rating data. Record the measurement location on-site and document it in the report so the same point can be identified and used in the future.

The accuracy of the method depends directly upon (1) the accuracy and uniformity of tracer injection and (2) *complete* mixing before the second sampling point. Tracer can be metered with precision either from a positive displacement fluid metering pump or from a Marriotte jar feeding an orifice, as shown in Figure 3-10. The Marriotte jar is useful only for open channels, whereas the pump can inject tracer at high pressure. To ensure accuracy, however, the pressure must not change during injection. It is possible to inject tracer into the low-pressure side of a small pump that "chases" the dye to the point of application with a small stream of water. Constant speed metering pumps can be obtained from Fluid Metering, Inc. [41]

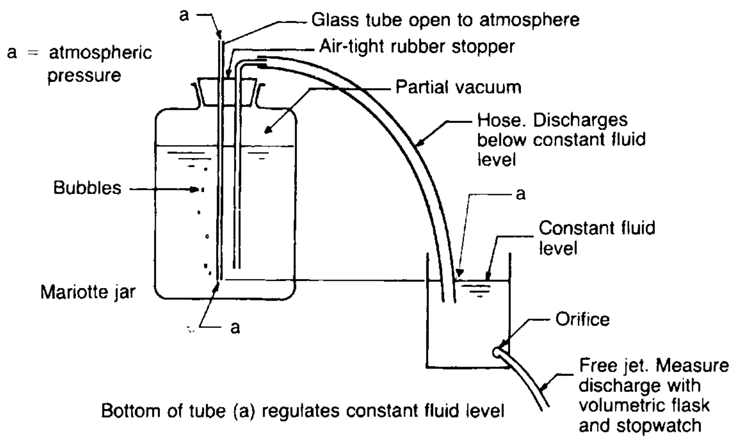


Figure 3-10. Constant head device for metering tracer.

for 12 V battery operation. The flow rates can be adjusted by changing the piston angle to alter the piston stroke length. Even more adjustment in tracer flow rate can be made by changing the concentration of the tracer. The reservoir of tracer should be calibrated either volumetrically or gravimetrically so that the accuracy of dispensing can be measured with a stopwatch.

By the time the tracer reaches the sampling location, mixing must be complete so that the concentration of tracer is uniform. One hundred pipe diameters downstream of the injection point along a straight section of pipe is considered sufficient, but sometimes that distance can be reduced substantially. Dye can be injected at the inlet side of a centrifugal pump and sampled on the discharge side. Or it can be injected into a pipe through a ball valve or a corporation cock (wherein leakage between probe and valve is prevented by means of O-rings) and sampled downstream by the same means. It helps to have several ejection ports in the injection (probe) tubing. Always test for complete mixing by withdrawing samples from various points across the pipe and also by comparing consecutive samples. Extreme care must be taken to ensure that none of the tracer escapes by recirculation or by leaking through check valves. (Close all isolation valves for pumps that are not running during the test.) If the wet well is contaminated with tracer, it will take a long time to remove all of it, and it will compromise the data.

Of the recommended tracers shown in Table 3-2, Rhodamine WT is the best choice for municipal water and wastewater because (1) flow-through fluorometers can be used in the field to determine when stability has occurred and when the more accurate grab sampling procedure can start; (2) the tracer is innocuous, inexpensive, and provides linear instrumental response in concentrations from the limit of detection (about 0.01 ppb depending upon the fluorometer used) to 100 ppb; (3) sorption on walls or particles is usually very slight; (4) background fluorescence is low (usually less than 0.2 ppb); (5) determinations are quick and relatively convenient; and (6) determinations can be made in the field. The disadvantages are (1) fluorometers are expensive and rarely found in chemical laboratories (although they can be rented); (2) Rhodamine is very temperature-sensitive and is attacked by chlorine; and (3) samples should not contain particles or air bubbles, so if the water is dirty, samples must either be centrifuged or tediously filtered through glass or Millipore filters.

If no tracer is lost (through leakage, reaction, absorption, etc.), a material balance yields:

Tracer in = tracer out

$$Qc_o + qC = (Q + q)c \tag{3-28}$$

where Q is flow rate in the pipe before the stream of tracer is added, c_o is blank or the apparent background concentration of tracerlike substance, q is the flow rate of tracer, C is the concentration of the tracer feed solution, and c is the final concentration of the tracer downstream. If Q is greater than $100q$ (as it should be), $Q + q$ approximates Q and Equation 3-38 can be modified to

$$Q = q\frac{C}{(c - c_o)} \tag{3-29}$$

The highest accuracy with Rhodamine WT cannot be achieved, however, without accounting for the

Table 3-2. Recommended Tracers and Analyses

Substance	Use	Analytical method	Concentration (mg/L) Lower detection limit	Advisable range	Comments
Fluoride	Excellent for potable water.	Specific ion electrode.[a]	0.0500	0.5–2.5	If utility adds fluoride to water, no added spike needed, hence it may be the best field tracer. Calibrate fluoride feeder or substitute solution feed and metering jump or Mariotte jar (Figure 3-10).
Glucose	Excellent for potable water.	Colorimetry by Park–Johnson method [42].	1	1–10	Occurs naturally in some fruits and is a food ingested by everyone. Sterilize any tracer solution to be injected into potable water. Little affected by chlorine. Protect samples from bacterial attack.
Rhodamine WT [43]	Excellent for wastewater. Approved for drinking water by the EPA [44].	Fluorometry[b], easiest of all tracer methods for use in laboratory or field.	0.00001	0.0002–0.4	Fluorometers uncommon in commercial laboratories. There is little background fluorescence. Adsorption on pipe walls and particles insignificant.
Lithium chloride	Good for manufacturing plants and water or wastewater.	Electrothermal atomic adsorption spectroscopy, laboratory only.	0.001	0.005–0.025	Very low natural background, used medicinally (but in far larger quantities).
Sodium chloride	Wastewater	Conductivity[c], good for field use.	1	100–1000	High background (1000 μmhos) requires massive spikes (~100 mg/L NaCl) and large flow ($q = \frac{1}{2}\%Q$) of brine.
Sodium chloride		Flame atomic adsorption.	0.002	0.03–1	Must dilute samples.
Chlorine	Water	DPD colorimetry [45] can be used in the field.	—	0.2–4	Usually unacceptable due to rapid reaction and decay.
Ascorbic acid (food grade Vitamin C)	Water	pH (easily used in the field) [40].	Unknown	Unknown	Acceptable by public. Must be tested for interferences.

[a]SPADNS colorimetric method can be used, but it is less convenient, more expensive, and subject to more interferences.

[b]Temperature sensitive: variation of 0.4°C in samples gives 1% error; use a water bath or correct mathematically.

[c]Correct for temperature: use a temperature-conductivity calibration chart in the field.

[d]Wright and Nevins [40].

effects of the flow stream on the dye (such as degradation or sorption) and on the actual fluorescent measurement itself (by masking the background fluorescence). Before and after every flow rate test, background samples should be collected upstream of the injection point and composited in sufficient volume (5 L) to produce a spiked background standard and a blank sample. The spike should be added in the field soon after collection at a concentration reasonably close to that expected in the downstream samples. Use only Class A volumetric glassware for dilution and pipetting.

Wastewater characteristics typically change under varying flow conditions. Solids increase significantly

in raw domestic wastewater as the flow increases from early morning to midmorning, and they can significantly affect the measured dye concentration. Under certain conditions, background sampling is the only way to detect dye recirculation through, for example, leaking valves into the wet well. Furthermore, there may be some unknown material that degrades the dye or absorbs it over time in the flow stream.

These background samples should be analyzed in the same manner as the downstream samples and used to adjust the measured sample dye concentrations as shown by the following equation developed by McDonald [46]:

$$C_{\mathrm{a}} = (C_m - b)\left[\frac{S}{(C_{\mathrm{s}} - b)}\right] \tag{3-30}$$

where C_{a} is the adjusted concentration of the sample, C_{m} is the measured concentration of the sample, b is blank (or background concentration), C_{s} is the measured concentration of the spiked background, and S is the true concentration of the spiked background based on the volumetric dilution calculations. The true flow rate is calculated from

$$Q = q\left(\frac{C}{C_{\mathrm{a}}}\right) \tag{3-31}$$

where q is the flow rate of the dye injected at concentration C.

As an absolute minimum, samples should be collected in groups of three for each flow rate tested. It is often difficult to predict how long it will take to reach steady-state conditions, so it is prudent to obtain extra grab samples to demonstrate that steady-state conditions were actually achieved. The concentration variability between samples in a group should be less than $\pm 2\%$ with complete mixing and steady-state conditions.

A flow-through fluorometer cell is useful for continuously monitoring downstream dye concentrations for determining completeness of mixing, equilibrium, and optimal dye concentration. But if precision is needed (for example, to calibrate flowmeters), use these data only for guidance. Use only grab samples, centrifuged and brought to uniform temperature, for precise results.

Although less than 20 mL of sample is required for measurement, the sample volume must be sufficient to allow for centrifuging or filtering the sample and for rinsing containers. Therefore, 250 to 500 mL of sample should be collected in groups of three and kept in borosilicate glass bottles. Decanting after settling by gravity instead of centrifuging is sometimes adequate, although it is time consuming. Borosilicate cuvettes (test tubes) should be acid-washed before each use and either dried or rinsed with sample prior to determinations. Mark cuvettes so each can always be inserted into the fluorometer with the same orientation, and test each for blank with distilled water. Use only those cuvettes with the same blank reading.

Fluorometers are now made with automatic temperature compensation for flow-through cells, and the flow-through cell can be used with large (1-L) grab samples by pumping the samples through it. But unless enough experience is gained to trust a particular instrument or procedure, it is wiser to put grab samples into small cuvettes and immerse them in a circulating water bath long enough (at least 20 min) for the samples to reach a predetermined stable temperature. After insertion into the fluorometer, obtain the instrument reading as soon as it stabilizes, because the temperature of the sample rises in the instrument due to the heat from the lamp, and readings change 2.6% per degree Centigrade. A liquid-cooled cuvette holder (if one is available or can be custom-made) is very desirable. Run all the samples without pause to obtain the best temperature control. Good advice on the details of fluorometric measurements is given by Turner Designs [47, 48]. This company also rents fluorometers and metering pumps.

Lithium chloride is also a good tracer. Lithium is rare in nature, so the background is low. It is not degraded by sunlight or chemicals, and sorption on walls and particles is less of a problem than it is with Rhodamine. It can be measured by means of electrothermal atomic absorption spectroscopy or by inductively coupled plasma (ICP) emission spectroscopy. These instruments are common in chemistry laboratories, but (unlike fluorometers, which are easily operated) they require considerable skill for operation, and they are more expensive than fluorometers. The samples can, however, be measured by trained personnel in commercial or university laboratories. Usually, samples are acidified and digested, but it seems likely that for constant injection where the concentration is near optimum, the samples can be measured much more economically without the usual preparation. It would be wise to make preliminary tests to determine whether accuracy is thus compromised. The disadvantages include the necessity of taking samples to a laboratory for analysis, high cost per sample, and uncertainty or delay in the field for determining either contamination or stability.

Calculation of Pipe Friction Coefficient

With pipe diameter, headloss, and flow as known quantities, the Hazen–Williams roughness coefficient, C, can be calculated by Equation 3-10a. Rearranging it into a more convenient form gives

$$C = 151\, QD^{-2.63} h_f^{-0.54} \tag{3-32a}$$

where Q is in cubic meters per second, D is the pipe ID in meters, and h_f is meters per 100 meters. In U.S. customary units Equation 3-10b becomes

$$C = 149 QD^{-2.63} h_f^{-0.54} \tag{3-32b}$$

where Q is in gallons per minute, D is in inches, and h_f is feet per 1000 feet.

It is desirable to make two (or, better) three tests, preferably at different flow rates. The C values should be reasonably close. If C is less than about 130, investigate for excessive mineral deposits or obstructions. Note that buried pipe can be both cleaned and relined in place with cement mortar or plastic at a total of 20–40% of the cost for new pipe in a "turn-key" operation (which includes all facets of the job—even a temporary bypass). Cost data and practical field methods for cement-mortar lining have been discussed by Walski [49].

3-10. Flow of Sludges

The preceding equations for friction headloss in pipes (Equations 3-9 and 3-11, for example) are limited to Newtonian fluids—those in which shear stress is proportional to shear rate and thus to velocity (see Figure 3-11). But in thick sludge, the shear stress increases very rapidly at low shear rates and increases only moderately with an increase of shear rate at higher velocity. Consequently, at low velocity the headloss for sludge is many times the headloss for water, whereas at high velocity the headloss for sludge may be only marginally higher than that for water.

In general, the headloss for sludges with less than 2% solids is nearly the same as for water. Most secondary sludge and water treatment plant sludge would fall into this category. The headloss for thicker sludge (usually primary sludge) can be approximated as explained in Section 19-1 by calculating the headloss of water and then multiplying by a factor obtained from Figure 19-4 or 19-5. But sludges are unique and the headloss for a "worst case" could be double the factors shown in Figure 19-3. For long pipelines where pressure loss is important, it is wise to test the friction properties of the sludge. Sludge flow and pumping are considered in greater detail in Chapter 19 and by the EPA [50].

3-11. Unsteady Flow

In all of the preceding sections, it is assumed that the flow rate is steady and continuous and that Equation 3-2 (the continuity equation) is valid, so the flow rate at one point equals that at another. In reality, this assumption is often only an approximation. Pipe material is not rigid and water is not incompressible. When valves are closed or opened quickly or when pumps are started or stopped, pressure waves are generated and travel back and forth in the pipe, which expands and contracts (as does the water) and causes the flow to become unsteady.

These violent pressure changes can have serious consequences on pumps, especially if the pressure changes continue for many cycles and if the head-flow rate curve of the pump is relatively flat. The intersection of the pipe system curve and the pump curve is the operating point of the pump—but only during steady flow when the pressure at discharge is a function only of static and friction head.

If the pressure increases somewhat, for example, because of the partial closure of a valve, the flow decreases greatly. If the pressure is reduced slightly below normal, the flow increases substantially. Thus, a component of flow (in which the water sloshes back and forth) is superimposed upon the average flow

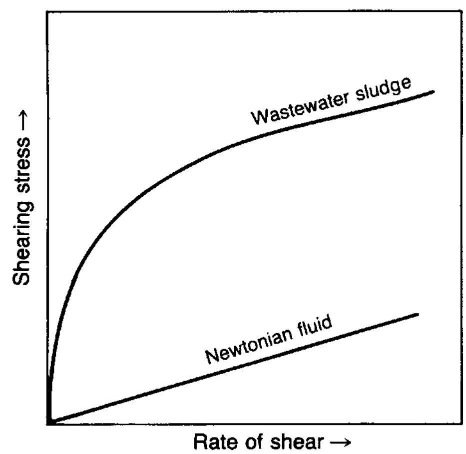


Figure 3-11. Shear rate relationships for representative fluids.

rate of the pump. If that component is in harmony with the natural frequency of the pressure waves in the pipe, resonance occurs, the pressure waves continue, and the pump can be destroyed in a few hours.

The problems associated with pressure waves can be even more severe and insidious with variable-speed pumps because the pumps operate over a considerable portion of the family of head-discharge curves.

The problem can be avoided or alleviated in a number of ways.

- Choose pumps that do not have a flat head-flow rate curve at the operating point.
- Design the system so that the pump does not operate for more than a few seconds at any point where the phenomenon described can occur.
- Make a transient analysis (see Chapters 6 and 7) and design the entire system to limit the magnitude and repetition of pressure waves.

3-12. Model Studies

Some types of problems defy traditional mathematical analysis and must be solved by extraordinary means. One such problem type is the flow regime in forebays and pump intake facilities. For structures of reasonable size that are based upon proven, successful designs, it is usually sufficient to follow established rules or guidelines. For large structures or those that differ significantly from designs known to be successful, a model study is the only means to ensure success.

Model studies of wet wells and/or forebays are indicated whenever:

- The design does not adhere in all respects (such as bay width, bell clearance, side slopes, piping, distance from obstructions, and change in direction of incoming flow) to the guidelines presented in Chapter 12, in ANSI/HI 9.8, or to established designs proven to be successful.
- The flow is greater than about 9000 m^3/h (40,000 gal/min) per pump or 23,000 m^3/h (100,000 gal/min or 140 Mgal/d) for the station.
- Excavation cost for size or depth is at a premium, or where achieving maximum design flow is critical.
- The approach flow is asymmetrical or nonuniform or there is a change in direction of incoming flow due, for example, to some pumps not operating.
- An existing sump is malfunctioning, and retrofitting is required.

Conducted by experienced personnel, model studies can be used to evaluate measures for alleviating the undesirable hydraulic conditions described in Section 27-6.

Model Similitude

True similitude requires that (a) the Froude number, $\boldsymbol{F} = v/(gL)^{0.5}$, a dimensionless function of gravitational and inertial forces; (b) the Reynolds number, $\boldsymbol{R} = vD/\boldsymbol{\nu}$ (Equation 3-12), a dimensionless function of gravitational and viscous forces, and (c) the Weber number, $\boldsymbol{W} = v^2 D\rho/\sigma$, a dimensionless function of inertial and surface tensile forces, be the same for both the prototype and a geometrically similar model. The velocity is v, g is the acceleration due to gravity, L is a length such as the flow depth, D is a length such as the intake diameter, $\boldsymbol{\nu}$ is kinematic viscosity, ρ is mass density, and σ· is surface tension. All dimensionless numbers cannot be satisfied simultaneously unless the model/prototype scale ratio is unity, so a compromise must be made. When a free surface exists, such as in a pump intake basin, flow patterns are influenced primarily by gravitational and inertial forces, and, therefore, similitude is based on the Froude number. Although the Reynolds number is much less important, it must be greater than 3×10^4 at the suction bell mouth to ensure minimum viscous effects. The Weber number should be greater than 120 for minimizing surface tension effects. For practicality in observing dye patterns and obtaining adequate measurements of velocity in the bell, the throat diameter of the suction should be at least 75 mm (3 in.). The linear scale ratio, L_m/L_p, of models is typically about $1/10$, but L_m/L_p for the models used for developing the guidelines in Sections 12-5 to 12-7 was close to $1/4$, so that for equal model and prototype values of $\boldsymbol{F}$, other model/prototype scale ratios were close to:

- Velocity: $v = (L_m/L_p)^{0.5} = 1/2$
- Time: $t = (L_m/L_p)^{0.5} = 1/2$
- Volume: $V = (L_m/L_p)^3 = 1/64$
- Flow rate: $Q = (L_m/L_p)^{2.5} = 1/32$

Construction

The geometry of the model and prototype must be similar, and in particular, a sufficient amount of the approach structure or water body must be included to simulate accurately prototype flow patterns into the sump. One side of the model (or at least a large window) must be constructed of clear plastic or glass for observing flow patterns. Likewise, suction bells and pipes should also be formed from

clear plastic. It is advantageous—in spite of extra cost—to construct the model so that selected dimensions, such as floor clearance for bells, depth of trench, or slope of floors, can be changed (possibly by adding or removing inserts) without rebuilding the model.

Impellers are not modeled, because the objective is to evaluate flow patterns approaching the impeller. Instead, the suction intake is equipped with a swirl meter consisting of four neutrally pitched vanes that rotate freely on a centered shaft. The outer diameter of the vanes should be 75% of the intake pipe diameter and the axial length of the vanes should be 0.6 pipe diameters. The swirl meter should preferably be located about four suction pipe diameters downstream from the mouth of the suction bell.

Measurements

Water level should be measured with a model accuracy of 3 mm ($^1\!/_8$ in.). Flowmeters should be calibrated to achieve an accuracy of at least $\pm 2\%$ of upper range value. Alternatively, a standard ASME orifice could be used without calibration. Very low approach velocities to pump intakes can sometimes be measured with dye by means of a stopwatch and a background grid. Dye is introduced through tubing small enough to be essentially nonintrusive. Higher velocities can be measured with a small propeller meter or by other suitable means at enough points to define the flow pattern and its stability.

For the final design, the distribution of axial velocity in the throat of the suction bell should be taken with velocity traverses along two perpendicular axes with a device capable of a repeatability of $\pm 2\%$ or better. Pitot tubes can be used, but a laser velocity meter with a computer readout, although an incredibly expensive device, provides much more extensive data and is completely nonintrusive.

The intensity of flow rotation, θ, in the intake is the swirl angle computed from

$$\theta = \tan^{-1}\left(\frac{\pi d n}{v}\right) \tag{3-33}$$

where d is pipe diameter, n is revolutions per second of the swirl meter, and v is average axial velocity at the swirl meter. Swirl is unsteady. The meter may only rock slightly for short or long periods followed by slow, rapid, or even reversing rotation. Hence, average rotation is not a complete description, and data on the percentage of time versus swirl angle should be obtained. As the swirl meter may sometimes rotate too rapidly to count revolutions, a good system is to photograph the swirl meter for 10 min with a video camera that overlays time in seconds on the images. This permanent record can be analyzed (at slow speed when necessary) for any type of time-rotation rate distribution wanted.

Vortices are measured visually with the aid of dye and artificial debris by comparing them to Figure 3-12. Vortices are usually unsteady in both location and strength and intermittent in occurrence. Vortices should be observed at, for example, every 15 sec during a 10-min interval to obtain vortex type versus frequency. Video recording is recommended.

Acceptance Criteria

The pump manufacturer, station designer, client, and research organization should agree on the acceptance criteria because the criteria may vary with the size, application, and kind of pumps. In general, the following provisions are satisfactory:

- Surface vortices of Type 3 or more and subsurface vortices of Type 2 or more are unacceptable, except that in some stations they may be acceptable only if they occur less than 10% of the time or only for infrequent pump operating conditions. Be cautious, however, because a weak effect in a small prototype can become stronger in a larger prototype.
- Swirl angles should be less than 5 degrees, except that 6 degrees may be acceptable only if they occur less than 10% of the time or only during infrequent pump operating conditions.
- The time-average velocities at specific points in the bell throat should be within 10% of the area average velocity. Time-varying fluctuations should have a standard deviation of 10% or less.

Although not mentioned in ANSI/HI 9.8-1998, the criteria ought to include "There shall be no cascade into the wet well pool, and air bubbles shall not be permitted to enter any pump intake during normal operation."

Tests

General observations should first be made on the initial design to identify any problems before making time-consuming detailed tests (unless detailed tests are wanted by the client for comparison with a final design). Modify the model until the general observations are consistent with satisfactory design. Then test

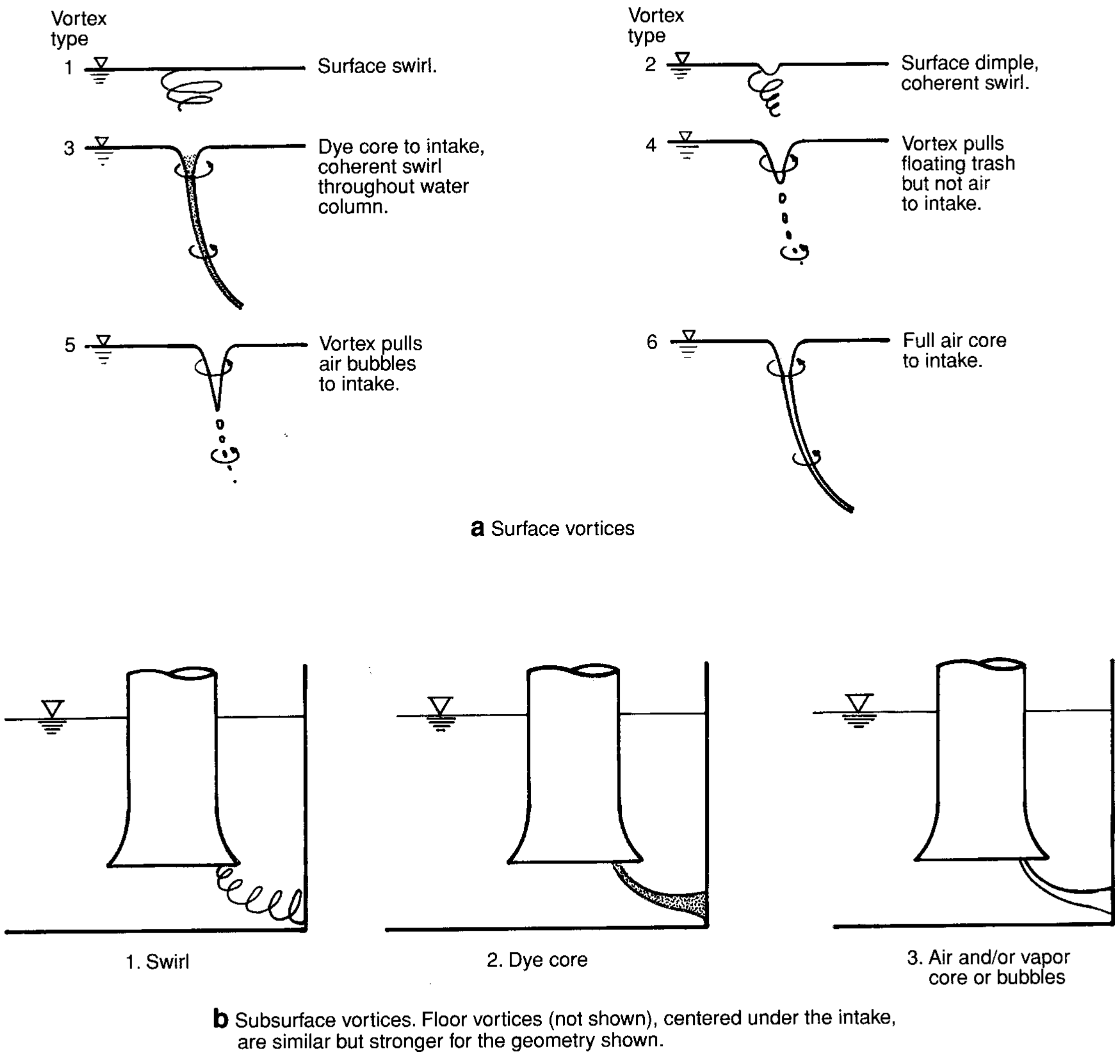


Figure 3-12. Classification of vortices. Courtesy of Alden Research Laboratory, Inc.

operating conditions at minimum, intermediate, and maximum flows and water levels. Include all possible pump operating conditions and combinations of pumps. Selected tests for the formation of free-surface vortices should be made at 1.5 times the Froude number to compensate for possible scale effects. Sometimes strong vortices can organize unexpectedly—at the low velocities that occur with HWL and low flow, for example.

Vortices and swirl angles should be observed in all tests. Axial velocity determinations should be made for each pump—but only in the final design. Document tests of swirling and vortexing with a video camera and include a VHS cassette of the tests with the final report.

Comparison with HI Standards

Part of Section 3-12 is adapted from (and all of it is compatible with) the guidelines in ANSI/HI 9.8-1998.

3-13. Computational Fluid Dynamics (CFD)

Computational fluid dynamics (CFD) is a powerful method for predicting fluid motion in a continuum by using numerical techniques. The Navier–Stokes equations governing fluid motion are valid at all points but, as it is not feasible to calculate flow characteristics at every point, the equations are solved at a

finite number of points called "nodes." Typically, nodes are closely spaced at discontinuities (such as walls and corners) where velocity gradients are large, whereas nodes are spaced farther apart where velocity gradients are low. Node spacing depends on the phenomena and geometry to be studied as well as on the numerical solver and the computer used. Computational time can vary from a few minutes to several days.

An increasing number of engineers are using CFD because programs (called "codes") are being developed for high-speed, relatively low-cost computers. Over 30 codes were available in 2003. Engineers with basic fluid mechanics backgrounds can, with minimal training, use these tools to analyze flow patterns. The most versatile codes include the ability to deal with several types of fluids (compressible, incompressible, Newtonian, non-Newtonian, etc.); change of phase, multiphase, steady and unsteady flow; shock waves; body forces; and even chemical reactions.

Fluid flow is described by partial differential equations of mass, momentum, and energy conservation. Equations that describe fluid properties such as viscosity and density are needed. Using the time-averaging technique, several terms that cannot be related to the mean flow quantities are generated. These quantities are called Reynolds stresses. It is impossible to solve the time-averaged equations unless certain assumptions are made for the Reynolds stresses. Various turbulence models are available depending upon those assumptions, and it is the responsibility of the user to select the appropriate turbulence model. After the set of differential equations is derived for a particular turbulence assumption, these equations are converted to a set of nonlinear equations by discretization. These simultaneous, nonlinear equations are then solved at all nodes to obtain the flow field by iteration to a predetermined level of accuracy.

A limitation of current (2004) codes is their inability to predict turbulence with accuracy. The energy in turbulence is contained in eddies whose sizes can range over ten orders of magnitude. To resolve the smallest eddies in three dimensions, about 10^{30} nodes would be required—well beyond the present limit of 10^6 nodes. Instead, special functions or turbulence models are used to estimate the production, transport, and dissipation of turbulent energy without dramatically increasing the number of nodes. The equations used to model turbulence contain constraints that are adjusted for the type of flow, for example, a free jet or flow over a backward-facing step. Turbulence levels and mean flow characteristics can be predicted for conditions near those for which the code was calibrated and verified. Research is currently being conducted to use direct numerical simulation to minimize assumptions for modeling turbulence.

CFD is useful for predicting overall flow distribution in pump sumps and forebays to within about one bell diameter of a pump intake. Approach flows to a pump intake (or to a pump bay from a forebay) can be influenced by withdrawals by other pumps, and such influences can be readily predicted by CFD. But flow details such as free surface and subsurface vortices cannot be predicted by CFD, and therefore, CFD cannot accurately predict objectionable phenomena such as swirling or asymmetrical flow within pump suction bells. Obtaining accurate data for such phenomena will require physical model studies until CFD codes are developed that can accurately model the smallest turbulent eddies that affect a flow field. The use of CFD in conjunction with physical model testing can sometimes reduce the total cost of modeling.

3-14. References

1. Ten-State Standards, *Recommended Standards for Sewage Works,* Great Lakes–Upper Mississippi River Board of Sanitary Engineers Health Education Service, Inc., Albany, NY (1978) (revised periodically).
2. Benedict, R. P., *Fundamentals of Pipe Flow,* John Wiley, New York (1980).
3. Williams, G. S., and A. Hazen, *Hydraulic Tables,* 1st ed., John Wiley, New York (1905); 3rd ed., 11th printing, Hazen and Sawyer P. C., New York (1965).
4. Bombardelli, F. A. and M. H. Garcia, "Hydraulic design of large-diameter pipes," *Journal of Hydraulic Engineering, American Society of Civil Engineers*, 129, 839–846 (November 2003).
5. Colebrook, C. E., "Turbulent flow in pipes, with special reference to the transition region between the smooth and rough pipe laws," *Journal of the Institution of Civil Engineers,* London, 11, 133 (1938–1939).
6. Moody, L. E., "Friction factors for pipe flow," *Transactions, American Society of Mechanical Engineers,* 66, 671–684 (November 1944).
7. Prandtl, L., *Essentials of Fluid Dynamics,* Chapter 3, Hafner Publishing, New York (1952).
8. Swamee, P. K., and A. K. Jain, "Explicit equations for pipe flow problems," *Journal of the Hydraulic Division, Proc. of the American Society of Civil Engineers,* 102, 657–664 (May 1976).
9. Ackers, P., *Tables for the Hydraulic Design of Stormdrains, Sewers, and Pipe-lines, Hydraulics Research Paper No. 4,* 2nd ed., Department of the Environment, Hydraulic Research Station, Wallingford, Berkshires, England (1969).
10. Jeppson, R., *Analysis of Flow in Pipe Networks,* Ann Arbor Science Publishers, Ann Arbor, MI (1976).
11. AWWA Manual M11. *Steel Pipe—A Guide for Design and Installation,* 3rd ed., American Water Works Association, Denver, CO (1989).

12. Gros, W. F. H., President, The Pitometer Associates, Inc., Chicago, IL, private communications (February/March 1996).
13. Lamont, P. A., "Common pipe flow formulas compared with the theory of roughness," *Journal of the American Water Works Association,* 73, 274–280 (May 1981).
14. Wolfe, T. F., "Advantages of cement linings for cast-iron pipe," *Journal of the American Water Works Association,* 38, 11–15 (January 1946).
15. Miller, W. T., "Durability of cement-mortar linings in cast-iron pipe," *Journal of the American Water Works Association,* 57, 773–782 (June 1965).
16. White, G. C., *The Handbook of Chlorination,* 2nd ed., Van Nostrand Reinhold Co., New York (1986), pp. 356–358.
17. DIPRA, *Handbook of Ductile Iron Pipe,* 6th ed., Ductile Iron Pipe Research Association, Birmingham, AL (1984) (revised periodically).
18. Brater, E. F., and H. W. King, *Handbook of Hydraulics,* 6th ed., McGraw-Hill, New York (1976).
19. Yarnell, D. L., and S. M. Woodward, "The flow of water in drain tile," Bulletin No. 854, Dept. of Agriculture Bulletin v 35, U.S. Government Printing Office (1922).
20. Camp, T. R., "Design of sewers to facilitate flow," *Sewage Works Journal* 18 (1): 3–16 (January 1946).
21. Wilcox, E. R., *A Comparative Test of the Flow of Water in 8-inch Concrete and Vitrified Clay Sewer Pipe,* Bulletin No. 27, Univ. of Washington Eng. Experiment Station (March 1924).
22. Escritt, L. B., *Sewerage and Sewage Treatment. International Practice*, edited and revised by W. D. Haworth, John Wiley, New York (1984).
23. Saatci, A., "Velocity and depth of flow calculations in partially filled pipes," *Journal of the ASCE,* Environmental Div., 116 (6): 1202–1208 (November/December 1990).
24. Wheeler, W., *Advancing Gravity Pipe Design*, unpublished. Available from William Wheeler, 683 Limekiln Road, Doylestown, PA 18901 (January 1994).
25. Wheeler, W., PARTFULL®. For a free copy of this computer program with instructions, send a formatted 1.4 MB, $3\frac{1}{2}$-in. diskette and a stamped self-addressed mailer to 683 Limekiln Road, Doylestown, PA 18901–2335.
26. Cahoon, J. E. See www.coe.montana.edu/ce/joelc/wetwell/
27. Kulin, G., *Recommended Practice for the Use of Parshall Flumes and Palmer-Bowlus Flumes in Wastewater Treatment Plants,* PB85-122745; EPA-600/ 2–84–186, U.S. Department of Commerce, National Technical Information Service, Springfield, VA (1984).
28. Grant, D. M., *Open Channel Flow Measurement Handbook,* 2nd ed., ISCO, Inc., Lincoln, NE (1981).
29. Wells, E. A., and H. B. Gotaas, "Design of Venturi flumes in circular conduits," *Transactions, American Society of Civil Engineers,* 133, 749–771 (1958).
30. Jorgensen, G., private communication (November 1984).
31. Walski, T., "Measuring differential pressure accurately in water distribution systems," *WATER/Engineering & Management,* 131, 28–30 (October 1984).
32. Walski, T. M., *Application of Procedures for Testing and Evaluating Water Distribution Systems,* U.S. Army Engineers Waterways Experiment Station, Technical Report, Vicksburg, MS (1984).
33. Shelley, P. E., and G. A. Kirkpatrick, "Sewer flow measurement—a state-of-art assessment." EPA-600/2-75-027, Contract No. 68-03-0426, Municipal Environmental Research Laboratory, Office of Research and Development, U.S. Environmental Protection Agency, 26 W. Martin Luther King Dr., Cincinnati, OH 45268 (November 1975).
34. Smith, E. C., private communication (December 1984).
35. Connell, J. E., private communication (January 1985).
36. DIN Number 1944, *Acceptance Test on Centrifugal Pumps (V.D. Rule for Centrifugal Pumps),* Deutsche Institut für Nommung (German Standards Institute), Berlin; available in English from Heyden & Son, Philadelphia, PA (October 1968).
37. Smith, E. C., private communication (February 1985).
38. Walski, T. M., *Analysis of Water Distribution Systems,* Van Nostrand Reinhold, New York (1984).
39. Miller, R. W., *Flow Measurement Engineering Handbook,* McGraw-Hill, New York (1983).
40. Nevins, T. R., C. E. Wright, and M. Damewood, " 'C'-ing inside the pipe with tracers," American Water Works Association, *Opflow*, Vol. 29, No. 9 (September 2003).
41. Fluid Metering, Inc., 29 Orchard St., P.O. Box 179, Oyster Bay, NY 11771.
42. Park, J. T., and M. J. Johnson, "A sub-micro determination of glucose," *Journal of Biological Chemistry,* 181, 149–151 (November 1949).
43. Rhodamine WT, obtainable from Crompton & Knowles Corp., 7535 Lincoln Ave., Skokie, IL 60076.
44. "Flow measurements," *Fluorometric Facts,* FR-1084-IM, Turner Designs, 845 W. Maude Ave., Sunnyvale, CA 94086 (n.d.).
45. *Standard Methods for the Examination of Water and Wastewater,* American Public Health Association, Albany, NY (revised periodically).
46. McDonald, R., Senior Scientist, Brown and Caldwell, Pleasant Hill, CA, private communication (1995).
47. "A practical guide to flow measurement," *Fluorometric Facts*, Turner Designs, Inc., 845 W. Maude Ave., Sunnyvale, CA 94086 (Rev. November 1995).
48. "Flow measurements in sanitary sewers by dye dilution," *Fluorometric Facts,* SS-7-80, Turner Designs Inc., 845 W. Maude Ave., Sunnyvale, CA 94086 (n.d.).
49. Walski, T. M., *Cost of Water Distribution System Infrastructure Rehabilitation, Repair, and Replacement,* Technical Report EL-85-5, Department of the Army Waterways Experiment Station, Vicksburg, MS (1985).
50. *EPA Process Design Manual for Sludge Treatment and Disposal,* EPA 625/1-79-011, pp. 14-1–14-57, U.S. Environmental Protection Agency, Municipal Environmental Research Laboratory, Office of Research and Development, Center for Environmental Research Information, Technology Transfer, Cincinnati, OH (September 1979).

Chapter 4

Piping

BAYARD E. BOSSERMAN II
JAMES C. DOWELL
ELIZABETH M. HUNING
ROBERT L. SANKS

CONTRIBUTORS
Thomas Flegal
Earl L. Heckman
Charles D. Morris
W. Stephen Shenk

The types of pipe, fittings, and associated materials commonly used in pumping stations and force and transmission mains for the pressurized flow of water, wastewater, and sludge are discussed in this chapter. The emphasis is on pipe 100 mm (4 in.) in diameter or larger, but small pipe for fuel, seal water, and plumbing is included.

The chapter is conceptually presented in four parts:

- Selection of pipe material
  - Exposed piping (piping within the pumping station), Section 4-1
  - Buried piping (force mains and transmission pipelines), Section 4-2
- Descriptions of pipe, fittings, joints, gaskets, and linings
  - Ductile iron pipe, Section 4-3
  - Steel pipe, Section 4-4
  - Plastic pipe, Section 4-5
  - Asbestos cement pipe, Section 4-6
  - Reinforced concrete pressure pipe, Section 4-7
- Design of piping, Section 4-8
  - Tie rods for flexible couplings
  - Wall thickness in exposed pipe
  - Hangars, supports, and spacing
- Special piping and plumbing, Section 4-9
  - Water
  - Diesel fuel service
  - Sewers
  - Chlorine
  - Air
  - Design of plumbing systems

This organization makes it possible to select the type of pipe to be used with no need to read more than a fraction of the chapter. After the type of pipe is chosen, skip to the appropriate section (Sections 4-3 to 4-7) for a detailed description of the piping. Pipe design and the design examples in Section 4-8 emphasize exposed piping. The design of buried piping is limited to generalities because that subject is so well covered in the literature [1, 2].

References to a standard or to a specification are given here in abbreviated form—code letters and numbers only (such as ASME B36.10). Double designations such as ANSI/AWWA C115/A21.15 indicate that AWWA C115 is the same as ANSI A21.15. Most standards are revised periodically, so obtain the latest edition. There are many standards that are important in design that are not specifically mentioned in the text, and designers must be alert to their requirements.

4-1. Selection of Exposed Pipe

Factors to be considered in the selection of pipe (whether exposed or buried) include the following:

- Properties of the fluid
  - Corrosive or scale-forming properties [3, 4, 5, 6, 7, 8]
  - Unusual characteristics, for example, viscosity of sludges
- Service conditions
  - Pressure (including surges and transients)
  - Corrosive atmosphere for exposed piping
  - Soil loads, bearing capacity and settlement, external loads, and corrosion potential for buried piping
- Availability
  - Sizes
  - Thicknesses
  - Fittings
- Properties of the pipe
  - Strength (static and fatigue, especially for water hammer)
  - Ductility
  - Corrosion resistance
  - Fluid friction resistance of pipe or lining
- Economics
  - Required life
  - Maintenance
  - Cost (fob plus freight to job site)
  - Repairs
  - Salvage value

Most of these factors are discussed here. If some that are not considered could influence selection, consult the literature and manufacturers' representatives.

Material

The great reserves of strength, stiffness, ductility, and resistance to water hammer; the wide range of sizes and thicknesses; and the wide variety of fittings available make either steel or ductile iron pipe (DIP) virtually the only logical choices for pump manifold piping. (See Section 2-2 for a definition of *ductile iron*.) The properties of these two pipe materials, compared in Table 4-1, are so similar that, in most situations, price is (or should be) the determining factor in the choice between DIP and steel. In the United States, DIP up to 750 mm (30 in.) seems to be preferred in the East, although steel pipe is often used in larger sizes. Steel pipe is used in all sizes in the West where freight makes DIP expensive. Some designers prefer steel pipe because the mitered and welded fittings allow greater flexibility in layout. A 48-degree

Table 4-1. Comparison of Pipe for Exposed Service

Pipe	Advantages	Disadvantages/limitations
Ductile iron (DIP)	Yield strength: 290,000 kPa (42,000 lb/in.^{2}) E = 166 × 106 kPa (24 × 106 lb/in.^{2}) Ductile, elongation ≈10% Good corrosion resistance Wide variety of available fittings and joints Available sizes: 100–1350 mm (4–54 in.) ID Wide range of available thicknesses Good resistance to water hammer	Maximum pressure: 2400 kPa (350 lb/in.^{2}) High cost, especially for long freight hauls No diameters above 1350 mm (54 in.) Difficult to weld Class 53 is the thinnest allowed for American flanged pipe (with screwed flanges in the U.S.) Corrosion resistance low unless lined
Steel	Yield strength: 207,000–414,000 kPa (30,000 – 60,000 lb/in.^{2}) Ultimate strengths: 338,000–518,000 kPa (49,000 – 75,000 lb/in.^{2}) E = 207 × 106kPa (2500 lb/in.^{2}) Diameters to 3.66m (12 ft) Widest variety of available fittings and joints Custom fittings can be mitered and welded Excellent resistance to water hammer Low cost	

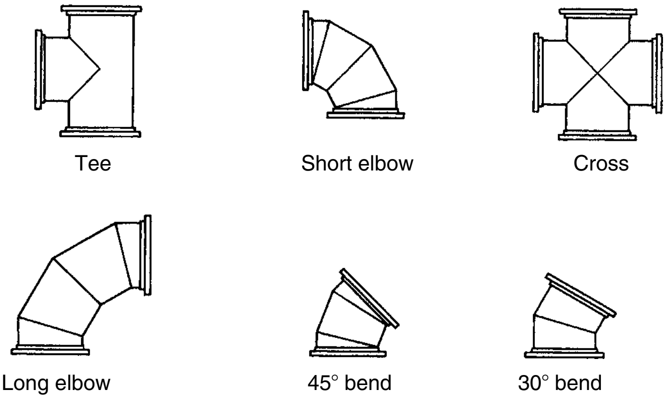


Figure 4-1. Typical mitered steel fittings.

bend, for example, is as easy to fabricate as a 45-degree bend (see Figure 4-1). Steel pipe has the further advantage of more convenient modification. DIP fittings are available only in standard configurations as described in AWWA C110 and C153.

Joints

The most practical connections in exposed piping are flanged, bolted joints (Figure 4-2) because they are rigid, cannot be deflected, and can resist a considerable amount of bending moment. There must, however, be enough couplings (Figures 4-3 to 4-5, for example) to make assembly and disassembly easy and to allow for inaccuracies in spool lengths and position. The connection in Figure 4-5 allows for considerable deflection.

Sleeve couplings (such as Dresser®) provide no longitudinal restraint (except for unreliable friction), so unless the pipe is otherwise anchored against significant longitudinal movement, the couplings must be harnessed as shown in Figure 4-3a and b. The number and size of tie rods are determined by the root area in the threads of the rods, as demonstrated in Example 4-1 on p. 4.28. Harnessing adds to the laying length requirement and cost of the joint, so omit unneeded harnesses where both ends are restrained externally.

Flanges

Inattention to the many and varied flange standards can result in misapplication with regard to pressure ratings and *costly*, embarrassing mismatches of flange bolt circles and bolt holes. The following is a summary of flange standards and designs encountered in the United States. Most of the following commentary concerns flanges with maximum designation of Class 300, although flange ratings as high as Class 2500 exist.

Flanges for steel pipe are always welded. Schedule 40, according to ASME B36.10, is considered to be the "standard" wall thickness for pipe up to 600 mm (24 in.). But for 600-mm (24-in.) or larger steel pipe, most U.S. manufacturers produce selected OD cylinders with fractional inch plate. Deviations from a pipe maker's norms are expensive. To avoid blunders, the wise designer becomes familiar with the sizes of pipe readily available and with the many codes listed in this chapter for pipe and connections. For example, Schedule 40 pipe can be threaded if it is steel, but not if it carries flammable fluids and never if it is polyvinyl chloride (PVC). In the United States flanges for DIP are always screwed to the barrel by the pipe manufacturer, and because Class 53 is the thinnest DIP allowed by ANSI/AWWA C115/A21.15-83 for threaded flanges, it is therefore the most common thickness used within a pumping

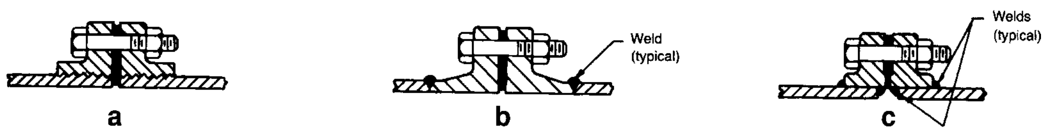


Figure 4-2. Common flanged joints for exposed or buried piping. (a) Threaded cast flange for ductile iron pipe; (b) welded neck flange for steel pipe; (c) slip-on flange for steel pipe. Courtesy of Wilson & Company, Engineers & Architects.

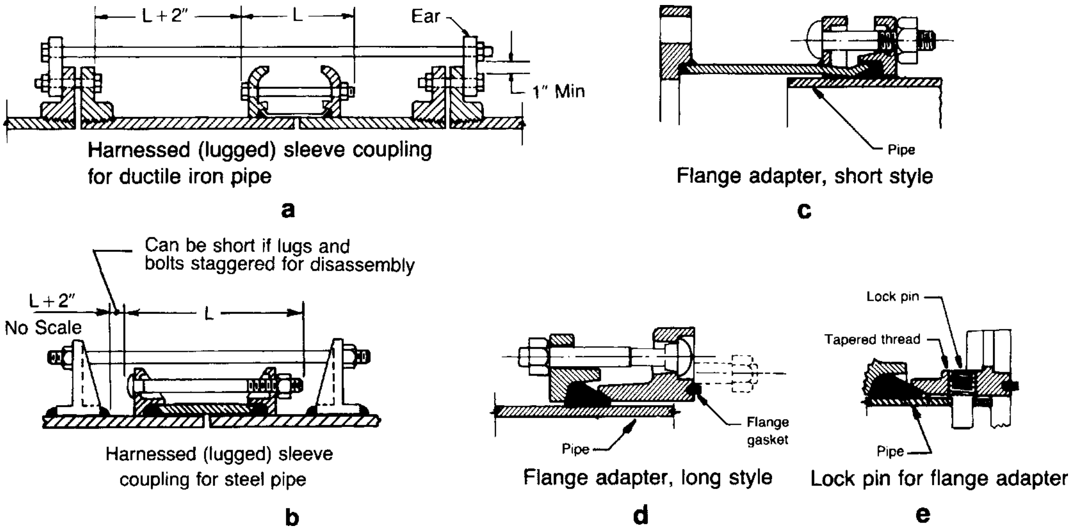


Figure 4-3. Sleeve couplings and flanged adapters. Unless restrained, the joints can separate. After Dresser Manufacturing Division.

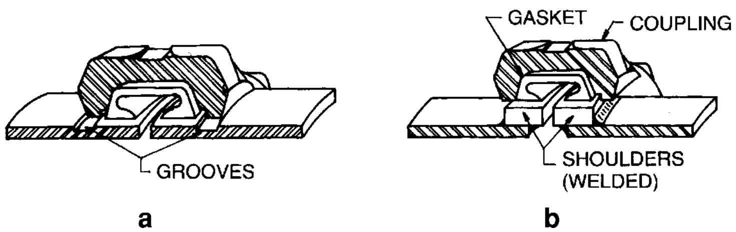


Figure 4-4. Grooved-end (Vitaulic®) couplings. (a) Grooved pipe; (b) welded shoulders.

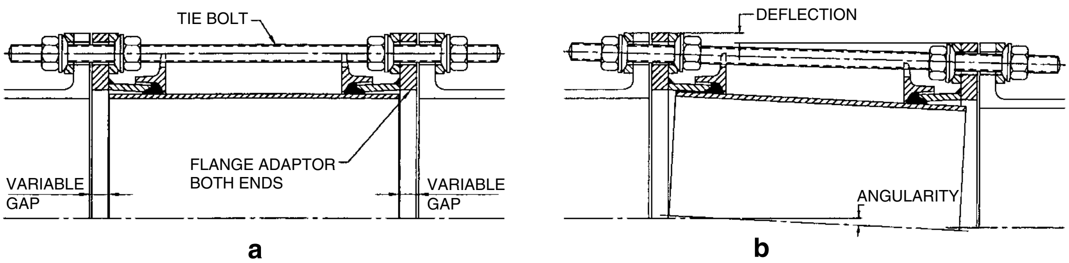


Figure 4-5. Equipment connection fitting. (a) Undeflected; (b) deflected. Courtesy of Viking Johnson.

station. Grooved-end couplings also require Class 53 DIP for diameters of 400 mm (16 in.) or less, but even thicker pipe is required for larger sizes according to AWWA C151 and C606.

For yard piping or for transmission or force mains, mechanical joints or push-on joints would be preferred, and Class 50 DIP or any of the several other pipe materials would probably be used. At 2 m/s (6.5 ft/s) velocity, the discharge in Class 50 DIP is greater than in Class 53 by about 2 to 6% and the headloss is less by about 1 to 3%. Because the concern here is mainly with the pumping station proper, the tables in Appendix B are limited to standard weight for steel and Class 53 for DIP; see the manufacturers' literature for other piping.

Flange Standards

The flange standards most frequently encountered in water works piping designs are described in the following documents:

- ASME B16.1, *Cast Iron Pipe Flanges and Flanged Fittings*. This standard covers Classes 25, 125, 250, and 800.
- ASME B16.5, *Pipe Flanges and Flanged Fittings ½ through NPS 24*. This standard covers Classes 150, 300, 400, 600, 900, 1500, and 2500.
- ASME B16.47, *Large Diameter Steel Flanges NPS 26 through NPS 60*.
- ASME B16.42, *Ductile-Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300*.
- AWWA C207, *Steel Pipe Flanges for Water Works Service—Sizes 4-Inch through 144-Inch*. Four pressure rating designs, designated Classes B, D, E, and F, are described.
- AWWA C115, *Flanged Ductile-Iron Pipe with Threaded Flanges*.

Flanges constructed in accordance with ASME or ANSI standards do *not* match flanges constructed in accordance with AWWA C207 in terms of pressure rating. Some (but not all) combinations match in terms of bolt circle and bolt hole number and diameter.

To make matters more confusing, the ASME B16.47, Series B (formerly API 605) flanges do not match ASME B16.1, B16.5, B16.42, or B16.47 (Series A) or AWWA standards in any particular. Finally, there is a Manufacturer's Standardization Society (MSS) standard SP 44 that matches ASME and AWWA in terms of flange OD, bolt circle, number of bolt holes, and bolt hole size but does not match in flange thickness. MSS-SP 44 does not match ASME B16.47 (Series B) in anything. For the sake of brevity, the following discussion is limited to AWWA and ASME B16.1, B16.5, B16.42, and B16.47 (Series A) flanges.

Facings

Important differences between AWWA C207 and ASME B16.1, B16.5, and B16.47 are:

- AWWA steel flanges are flat-faced, while ASME B16.5 and B16.47 steel flanges have a raised face. That is, the ASME flange has a projection inside the bolt circle that is about 1.5 mm ($^1/_{16}$ in.) higher than the portion of the flange outside the bolt circle.
- Above 1200-mm (48-in.) size, ASME B16.1 and B16.47 and AWWA Class B, D, and E flanges do not match in bolt hole size, but they do match in bolt circle and number of bolt holes.

When mating steel flanges to cast-iron flanges, be sure the steel flange has the raised face removed. Steel to cast-iron flange connections should have flat-faced flanges. If the raised steel face is not removed, the stress induced by bolting the flanges together can crack the cast iron.

Dimensions (flange OD, bolt circle diameter, and number and size of bolt holes) of AWWA Class D flanges through 600-mm (24-in.) size are the same as those for ASME B16.5, Class 150. Dimensions of AWWA Class D flanges 650-mm (26-in.) through 1500-mm (60-in.) size are the same as those for ASME B16.47, Class 150. AWWA Class D flange dimensions match ASME B16.1, Class 125, cast-iron flanges in all sizes.

Cast-iron flanges ANSI B16.1, Class 125, have the same drilling and bolting dimensions as ASME B16.5, Class 150, steel flanges in sizes 600 mm (24 in.) and smaller. Cast-iron flanges ASME B16.1, Class 250, have the same drilling and bolting dimensions as ASME B16.5, Class 300, steel flanges in sizes 600 mm (24 in.) and smaller. Class 125/150 flanges do *not* match Class 250/300 flanges in any size.

AWWA Class E flanges have a pressure rating of 1900 kPa (275 lb/in.^{2}), but they conform to the dimensions of ASME B16.1, Class 125. Only AWWA Class F flanges conform to ASME Class 250/300 dimensions (1200 mm or 48 in. and smaller). The dimensions (OD, bolt circle, diameter, number of bolts, bolt size, etc.) of ASME B16.47 Class 300 flanges match those of AWWA C207 Class F flanges and ASME B16.1, Class 250 flanges in sizes 36 in. and smaller.

ASME B16.47 describes two series of flanges. Series A specifies flanges for general use. Series B specifies compact flanges which have smaller bolt circle diameters and match API 605 flanges. The two series of flanges are not interchangeable. The Series A Class 150 flanges match the dimensions (flange OD, bolt circle diameter, number and sizes of bolt holes) of AWWA C207 Class D and E, AWWA C115, ASME B16.1 Class 125, and ASME B 16.42 Class 150 flanges in sizes 1500 mm (60 in.) and smaller. The Series A Class 300 flanges match the dimensions of AWWA C207 Class F, ASME B16.1 Class 250, and ASME B16.42 Class 300 flanges in sizes 900 mm (36 in.) and smaller. The flange standards corresponding to various AWWA and ASME pressure class ratings, materials, and pipe sizes are summarized in Table 4-2.

Table 4-2. Flange Standards

Pressure class			Maximum pipe size		
kPa	lb/in.^{2}	Material	mm	in.	Equivalent[a] ASME standard
860	125	C.I.[c]	2400	96	B16.1, CL. 125
1030	150	Steel[b]	600	24	B16.5, CL. 150
1730	250	C.I.[b]	1200	48	B16.1, CL. 250
2070	300	Steel[b]	600	24	B16.5, Cl. 300
AWWA "D"		Steel[c]	3600	144	B16.1, CL. 125
AWWA "E"		Steel[c]		144	B16.1, CL. 125
AWWA "F"		Steel[c]		48	B16.1, CL. 250; and B16.5, CL. 300

[a]Equivalent in bolt circle and spacing and flange OD.

[b]Raised face.

[c]Flat face.

Pressure Ratings

Temperature is not mentioned in AWWA specifications for flanges, but there is a pressure–temperature relationship for ASME flanges in which temperatures above 38°C (100°F) reduce the allowable pressure rating. In ASME 16.5, a Class 150 steel flange has a pressure rating of 197 kPa (285 lb/in.^{2}) at 38°C (100°F). Hence, Class 150 flanges are adequate for any lower pressure at the temperatures usually encountered in traditional water works piping.

Some pressure ratings (static plus water hammer pressure) for various types of flanges usually encountered in water works piping are summarized in Table 4-3. Refer to AWWA C207, ASME B16.1, ASME B16.5, ASME B16.42, AWWA C115, and ASME B16.47 for flange dimensions and to ASTM A307 and A193 and ASME B31.1 and B31.3 for bolts and nuts.

Sleeve (such as Dresser®) couplings provide no longitudinal restraint (except for unreliable friction), so unless the pipe is otherwise anchored against significant longitudinal movement, the couplings must be harnessed as shown in Figures 4-3a and b. The number and size of tie rods are determined by the root area in the threads of the rods, as demonstrated in Example 4-1. Harnessing adds to the laying length requirement and cost of the joint, so omit unneeded harnesses.

Flanged adapters (Figure 4-3c and d) accomplish the same purpose as sleeve couplings but are more economical for two reasons: (1) shorter laying lengths, and (2) fewer flanges. Longitudinal movement can be restrained by the use of lock pins (Figure 4-3e), but if there are likely to be many repetitions of longitudinal force, the lock-pin holes in the pipe may wear and eventually fail. Flanged adapters, however, are useful and unquestionably excellent for externally restrained pipe.

Grooved-end couplings (such as Victaulic®) are the most economical of the flexible joints because (1) the laying length required is so short, and (2) two flanges are omitted. Grooved ends can only be used in thick-walled pipe (see Tables 4-4 and 4-5), but thin steel pipe can be adapted for these couplings by welding shoulders to the pipe as shown in Figure 4-4. Grooved-end couplings allow expansion and contraction of ± 3 mm ($^1\!/_8$ in.) or a total travel of 6 mm ($^1\!/_4$ in.) at each coupling. Consequently, special expansion joints can usually be eliminated. Standards for grooved-end fittings are contained in AWWA C606. Grooved-end couplings for steel and ductile iron pipe are manufactured of malleable or ductile iron, ASTM A 47 or A 536, respectively. Carbon steel bolts should conform to ASTM A 183. Stainless steel bolts should conform to ASTM A 193, Class 2. Gaskets should conform to ASTM D 2000.

Sleeve couplings, flanged adapters, and, to some degree, grooved-end couplings can be purposely deflected. However, use them only to provide for accidental misalignment, expansion, and ease of installation and disassembly. Use bends or other fittings for calculated deflection angles. To prevent strain from being transmitted to equipment (such as pump casings), use the types that do not transmit shear.

At times, transitions from one pipe material to another are required, for example, conversion from ductile iron for exposed service to polyvinyl chloride (PVC) for buried service. Adapters for this purpose are commercially available for almost any type of pipe material transition. Consult the pipe manufacturer for the particular application.

Table 4-3. Pressure Ratings of Flanges

			Temperature range		Pressure rating			
							Pipe diameter	
Class	Standard	Material	°C	°F	kPa	lb/in.^{2}	mm	in.
125	ASME B16.1	C.I.	−29 to +66	−20 to +150	1380	200	300	12
					1030	150	350–1200	14–48
150	ASME B16.5	Steel	−29 to +66	−20 to +100	1970	285		24–42
250	ASME B16.1	C.I.	−29 to +66	−20 to +150	3450	500	≤500	12
					2070	300	350–1200	14–18
300	ASME B16.5	Steel	−29 to +66	−20 to +100	5100	740	600–1050	24–42
"B"	AWWA C207	Steel	−29 to +66	—	590	86		
"D"	AWWA C207	Steel	−29 to +66	—	1210	175	100–300	4–12
					1030	150	>300	>12
"E"	AWWA C207	Steel	−29 to +66	—	1900	275		
"F"	AWWA C207	Steel	−29 to +66	—	690	100		

Gaskets

Gaskets required for ductile iron or gray cast-iron flanged joints are 3.2 mm (1⁄8 in.) thick, and those for steel flanges can be either 1.6 mm (1⁄16 in.) or 3.2 mm (1⁄8 in.) thick (see "Gaskets" in Sections 4-3 and 4-4 for details).

Thickness

Temperatures exceeding 871 to 927°C (1600 to 1700°F) erode the ductility and strength of ductile iron, so if they are used at all, welds must be made with great care to avoid overheating. Hence, flanges for pipe made in the United States are screwed to the barrel by the pipe manufacturer. The thinnest allowable DIP for threaded barrels is Class 53. The thickness of Class 53 DIP is given in Tables B-1 and B-2 (Appendix B). Spools (short lengths of pipe) can sometimes be obtained with integrally cast flanges that may permit Class 50 pipe to be used. Flanges for ductile iron or gray iron valves and fittings are always cast integrally. Some European manufacturers

Table 4-4. Minimum Wall Thickness for Grooved-End Couplings in Ductile Iron Pipe

Nominal size of pipe		Ductile iron pipe thickness class per AWWA C151 and C606
mm	in.	
100–400	4–16	53
450	18	54
500	20	55
600	24	56

Table 4-5. Minimum Wall Thickness for Grooved-End Couplings in Steel Pipe per AWWA C606

Nominal size of pipe		Thickness[a,b]		ASME B36.10 Schedule No.
mm	in.	mm	in.	
75	3 ID	4.7	0.188	40[c]
100	4 ID	5.2	0.203	40[c]
150	6 ID	5.6	0.219	40[c]
200	8 ID	6.0	0.238	20[c]
250	10 ID	6.4	0.250	20
300	12 ID	7.0	0.279	30[c]
350	14 OD	7.1	0.281	20[c]
400	16 OD	8.0	0.312	20
450	18 OD	8.0	0.312	20
500	20 OD	8.0	0.312	20[c]
600	24 OD	9.5	0.375	30[c]

[a]Thin pipe can be used with shouldered couplings (see Figure 4-4b).

[b]Standard weight pipe up to 600 mm (24 in.) can be used.

[c]Nearest Schedule number.

do weld flanges to DIP barrels, and this practice permits the use of Class 50 instead of Class 53 pipe.

Steel pipe can be obtained in a very wide range of thicknesses and diameters [9]. Flanges are welded to the barrel of steel pipe. Thicknesses for carbon steel vary from Schedule 10 to Schedule 160 and from "standard" to "double extra heavy." Standard weight pipe is shown in Tables B-3 and B-4 (Appendix B). Standard weight and Schedule 40 are identical for diameters up to and including 250 mm (10 in.). All larger sizes of standard weight pipe have a wall thickness of 9.5 mm (3/8 in.)

Depending on pipe size and internal pressure, steel pipe may have to be reinforced at tees, wyes, and other openings. As discussed in Section 4-4, reinforcement can be one of three kinds: collar, wrapper, or crotch.

Linings and Coatings

Considering its low cost, long life, and sustained smoothness (see "Friction Coefficients" in Section 3-2), cement-mortar lining for both ductile iron and steel pipe is the most useful and the most common. Standard thicknesses for shop linings are given both in Table 4-6 and (for DIP) in Tables B-1 and B-2. Pipe can, after cleaning, be lined in place with the thicknesses given in Table 4-7. Although cement mortar is normally very durable, it can be slowly attacked by very soft waters with low total dissolved solids content (less than 40 mg/L), by high sulfate waters, or by waters undersaturated in calcium carbonate. For such uses, carefully investigate the probable durability of cement mortar and consider the use of other linings. The marble test [10] is a simple, accurate, and inexpensive means for determining the calcium carbonate saturation. Sulfate and total dissolved solids are also easily tested in almost any water laboratory. Because the standard, shop-applied mortar linings are relatively thin, some designers prefer to specify shop linings in double thickness.

In specifying mortar lining, match the pipe ID with the required valve ID, particularly with short-body butterfly valves in which the valve vane protrudes into the pipe. If the ID is too small, the valve cannot be fully opened.

Other linings and uses are given in Table 4-8. The plastics are usually about 0.3 mm (0.012 in.) thick. Glass lining is about 0.25 mm (0.010 in.) thick. In general, the cost of cement mortar is about 20% of that of other linings, so other linings are not justified except where cement mortar would provide unsatisfactory service. One exception is glass, which is supreme for raw primary sludge because it inhibits grease and solids buildup and is highly resistant to chemical attack. Applying the glass lining to the pipe requires a temperature of about 760°C (1400°F), which can safely be applied to DIP provided that the original thickness is at least Class 54 for 150- to 500-mm (6- to 20-in.) pipe or Class 56 for 100-mm (4-in.) pipe. The interior must be very smooth, so the pipe is either bored (from Class 56 to Class 53, for example) or blasted with abrasive grit. Manufacturers may differ in their thickness requirements, so consult them before writing a specification. Steel pipe should be seamless because weld seams cannot be covered. Welded pipe can, however, be used if the weld seams are machined smooth. Any end connection found on DIP can be used with glass lining, but only plain-end, flanged, or grooved-end connections can be used for glass-lined steel pipe. Steel bell and spigot joints deform under the heat required. Because glass lining

Table 4-6. Thickness of Shop-Applied Cement-Mortar Linings

Nominal pipe diameter		Lining thickness: Ductile iron pipe[a]		Steel pipe[b]		
mm	in.	mm	in.	mm	in.	
100–250	4–10	1.6	1/16	6.4	1/4	
300	12		1.6	1/16	7.9	5/16
350–550	14–22	2.4	3/32	7.9	5/6	
600	24		2.4	3/32	9.5	3/8
750–900	30–36	3.2	1/8	9.5	3/8	
1050–1350	42–54	3.2	1/8	12.7	1/2	
>1350	>54		—	—	—	—

[a]Single thickness per AWWA C104. Linings of double thickness are also readily available.

[b]Per AWWA C205.

Table 4-7. Thickness of Cement-Mortar Linings of Pipe in Place per AWWA C602

Nominal pipe diameter		DIP or gray cast iron (new or old pipe)		Steel pipe			
				Old pipe		New pipe	
mm	in.	mm	in.	mm	in.	mm	in.
100–250	4–10	3.2	1/8	6.4	1/4	4.8	3/16
300	12	4.8	3/16	6.4	1/4	4.8	3/16
350–550	14–22	4.8	3/16	7.9	5/16	6.4	1/4
600–900	24–36	4.8	3/16	9.5	3/8	6.4	1/4
1050–1350	42–54	6.4	1/4	9.5	3/8	9.5	3/8
1500	60	—	—	9.5	3/8	9.5	3/8
1650–2250	66–90	—	—	12.7	1/2	11.1	7/16
>2250	>90	—	—	12.7	1/2	12.7	1/2

Table 4-8. Linings for Ductile Iron and Steel Pipe

Lining material	Reference standard	Recommended service
Cement mortar	AWWA C104, C205	Potable water, raw water and wastewater, activated and secondary sludge
Glass	None	Primary sludge, very aggressive fluids
Epoxy	AWWA C210	Raw and potable water
Fusion-bonded epoxy	AWWA C213	Potable water, raw water and wastewater
Coal-tar epoxy	AWWA C210	Not recommended for potable water
Coal-tar enamel	AWWA C203	Potable water
Polyurethane	AWWA C222	Raw wastewater, water
Polyethylene	ASTM D 1248	Raw wastewater

becomes very expensive for pipes larger than 250 mm (10 in.), other linings (such as cement mortar) should ordinarily be considered.

Usually, ductile iron pipe is furnished with an exterior asphaltic coating. This asphaltic coating is only 1 mil thick and provides little or no corrosion protection. If the pipe is to be painted, either a bare pipe or a shop-applied primer coat compatible with a finish coat of enamel or epoxy should be specified.

Fittings

Some standard iron fittings are shown in Figure 4-6. A comprehensive list of standard and special fittings is given in Table 4-9. Greater cost and longer delivery times can often be expected for special fittings. Fittings are designated by the size of the openings, followed (where necessary) by the deflection angle. A 90-degree elbow for a 250-mm (10-in.) pipe would be called a 250-mm (10-in.) 90-degree bend (or elbow). Reducers, reducing tees, or reducing crosses are identified by giving the pipe diameter of the largest opening first, followed by the sizes of other openings in sequence. Thus, the reducing outlet tee in Figure 4-7 might be designated as a 250-mm $\times$ 200-mm $\times$ 250-mm tee (10-in. $\times$ 8-in. $\times$ 10-in. tee).

Most steel fittings are mitered (Figure 4-1), but wrought fittings (Figure 4-7) are also used. A comprehensive list of customary mitered and wrought fittings is given in Table 4-10 (refer also to AWWA M11 [9]).

Expansion, Contraction, and Vibration

Pump vibration or expansion and contraction of the pipe may, at times, be significant. Compensation for these types of movement may be made by installing flexible connectors. These connectors are usually constructed of rubber and may be bolted directly to the flanges of equipment and piping (refer to Chapter 22 and Figure 22-14). Many connectors for these

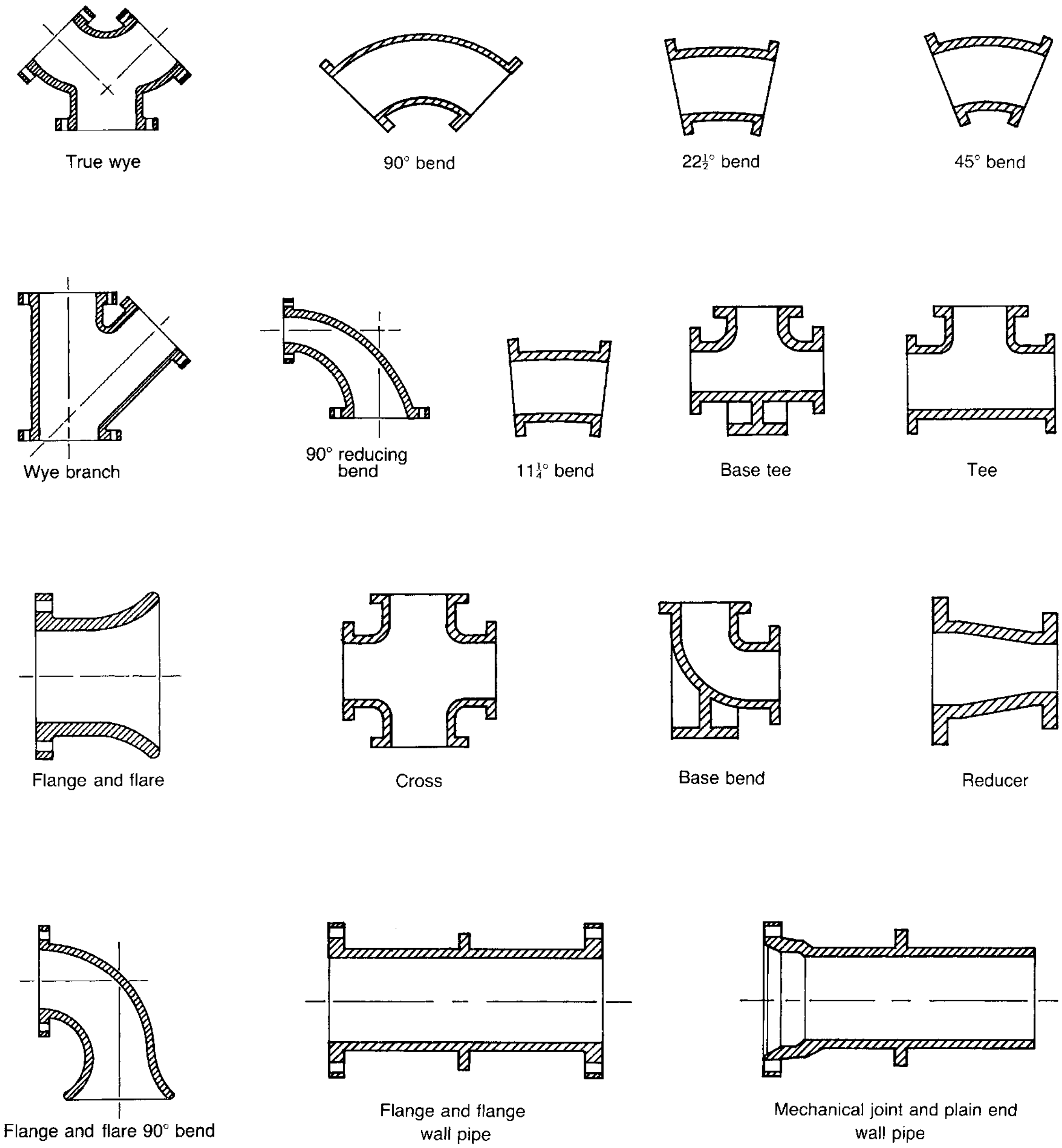


Figure 4-6. Ductile or gray iron flanged fittings.

applications are corrugated, but various shapes are available.

Summary: Pumping Station Piping

The joints in piping within a pumping station should almost universally be bolted flanges augmented with a few strategically placed grooved-end couplings (e.g., Victaulic®) or sleeve pipe couplings (e.g., Dresser®) to permit disassembly and allow for misalignment (see Figures 4-3 to 4-5). Connections to equipment (such as pumps) should be flexible to prevent the transmission of undue forces, including shear. Mason radial ply joints of resilient material (rubber, Viton, or Buna N) are excellent. Occasionally other joints are used, but only with many grooved-end couplings added for ease in disassembly.

Table 4-9. Ductile Iron and Gray Cast-Iron Fittings, Flanged, Mechanical Joint, or Bell and Spigot[a]

Standard fittings	Special fittings
Bends (90°, 45°, 22 ½°, 11 ¼°)	Reducing bends (90°)
Base bends	Flared bends (90°, 45°)
Caps	Flange and flares
Crosses	Reducing tees
Blind flanges	Side outlet tees
Offsets	Wall pipes
Plugs	True wyes
Reducers	Wye branches
Eccentric reducers	
Tees	
Base tees	
Side outlet tees	
Wyes	

[a]Sizes from 100 to 1350 mm (4 to 54 in.).

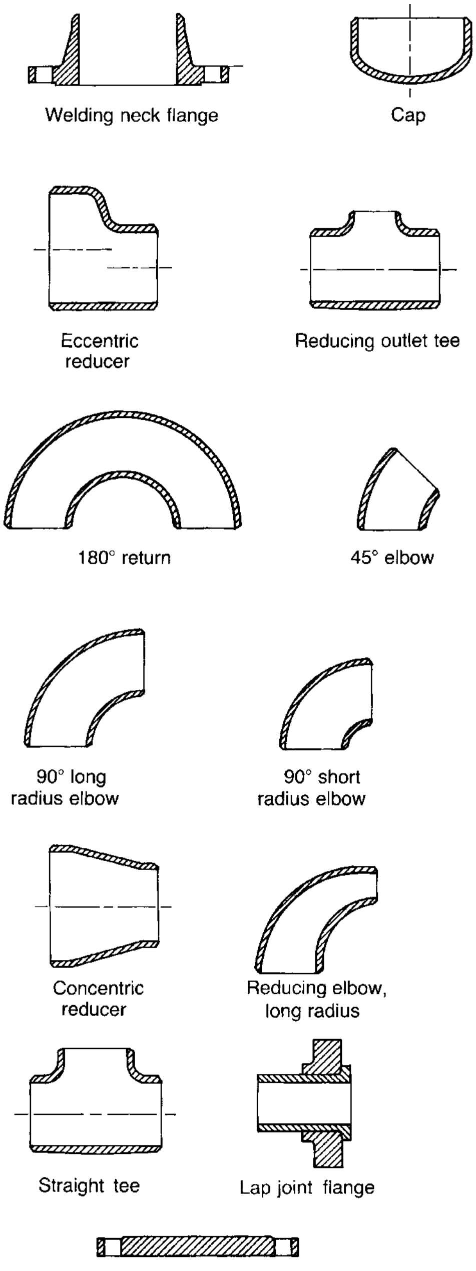


Figure 4-7. Wrought (forged) steel fittings for use with welded flanges.

4-2. Selection of Buried Piping

Buried piping must resist internal pressure, external loads, differential settlement, and the corrosive action of soils. The profile, the velocity of flow, and the size and stiffness of the pipe all affect water hammer; thus, they affect the design of the pumping station itself as well as that of the force main. General factors affecting piping selection are listed in Section 4-1.

Material

As buried pipes are supported by the trench bottom, there are more possibilities for selecting piping materials and joints. As force mains are much longer than the pipe within a pumping station, substantial savings are possible by choosing the most economical pipe. Recommended materials are compared in Table 4-11. Plastics other than PVC are used for acids and chemicals. For wastewater service, both acrylonitrile butadiene styrene (ABS) and PVC are commonly used, but PVC pipe is the only plastic commonly used for potable water service.

Reinforced concrete pressure pipe, generically described in Table 4-11, is divided into four types described in Table 4-12.

Practical Selection

Considering experience with, and characteristics of, the various pipe materials available for wastewater force mains, the following materials are recommended. Use HDPE or PVC up to a diameter of 600 mm (24 in.) for wastewater force mains. Note, however, that if the pressure is 75% or more of the working pressure of the pipe or if there are repetitive pressure cycles, as with cycling constant speed pumps off and on, experience has shown that the material

Table 4-10. Steel Fittings

Mitered fittings	Wrought fittings
Crosses	Caps
2-Piece elbows, 0–30° bend	45° elbows
3-Piece elbows, 31–60° bend	90° elbows, long radius
4-Piece elbows, 61–90° bend	90° elbows, short radius
4-Piece, long radius elbows	90° reducing elbows, long radius
Laterals, equal diameters	
Laterals, unequal diameters	Multiple-outlet fittings
Reducers	Blind flanges
Eccentric reducers	Lap joint flanges
Tees	Slip-on flanges
Reducing tees	Socket-type welding flanges
True wyes	Reducing flanges
	Threaded flanges
	Welding neck flanges
	Reducers
	Eccentric reducers
	180° returns, long radius
	Saddles
	Reducing outlet tees
	Split tees
	Straight tees
	True wyes

may fail. For larger pipe, use polyurethane or fusion-bonded, epoxy-coated ductile iron or steel pipe. Consider RCPP with a PVC lining above a diameter of 900 or 1050 mm (36 or 42 in.). If there is a possibility of air entering the force main, line the top of the RCPP with PVC to guard against the potential for acid corrosion. Careful evaluation with regard to internal pressures, external loads, corrosiveness of the wastewater, and corrosiveness of the soil is required in selecting the pipe material for a given project or application.

Joints

Because buried piping such as a force main or water line is fully laterally supported, the joints should be flexible to accommodate minor settlement; hence, less expensive joints, better suited for burial than bolted flanged joints, should be used (see Figures 4-8 and 4-9).

Many kinds of joints are available (see Figures 4-8 through 4-14), and several common ones are listed in Table 4-13. The check marks (✓) show the type of pipe for which they are suited. For buried service, sliding couplings (such as bolted sleeve type per AWWA C219, i.e., Dresser®) need not be harnessed if thrust blocks are installed at bends or if there is a sufficient length of pipe on both sides for soil friction to resist the thrust (see DIPRA [11]). Grooved-end couplings (see Figure 4-4) allow for slight changes in alignment (about 5 degrees per coupling), but do not deliberately make use of this flexibility; use bends or ball joints instead.

For ductile iron pipe, push-on and mechanical joints are the most commonly used for buried service. These joints allow for some pipe deflection (about 2 to 5 degrees depending on pipe size) without sacrificing watertightness. Except in certain unstable soils, buried joints are not required to support the weight of the pipe.

Gaskets

Gaskets for various joints are described in Sections 4-3 to 4-5.

Thickness

The thickness of buried pipe must be sufficient to resist the external soil and overburden loads as well as internal pressure. If there is no traffic loading and the burial is shallow (say, 2 m or 6 ft), ductile iron pipe Class 50 can often be specified where internal pressure is less than the pipe pressure rating.

The stresses in (and deflection of) pipe walls due to soil and overburden loads are complex functions of (1) pipe material, diameter, and thickness; (2) bedding conditions; (3) soil properties; (4) trench width and depth; (5) frost; and (6) the overburden load and its distribution. Although not difficult, the calculations are too involved to be included here. Refer to:

- AWWA M11 [9] for steel pipe
- DIPRA [11] for DIP
- ASTM D 2774 and Uni-Bell Handbook [12] for PVC pipe
- AWWA C401 and C403 for AC pipe
- AWWA M9 [13] for RCPP
- Spangler [14, 15] for the most extensive discussion

Linings, Coatings, and Cathodic Protection

The discussion of lining for exposed service applies to buried service as well. Buried glass-lined pipe should

Table 4-11. Comparison of Pipe for Buried Service

Pipe	Advantages	Disadvantages/limitations
Ductile iron pipe (DIP)	See Table 4-1; high strength for supporting earth loads, long life.	See Table 4-1; may require wrapping or cathodic protection in corrosive soils.
Steel pipe	See Table 4-1; high strength for supporting earth loads.	See Table 4-1; poor corrosion resistance unless both lined and coated or wrapped, may require cathodic protection in corrosive soils.
Polyvinyl chloride (PVC) pipe	Tensile strength (hydrostatic design basis) = 26,400 kPa (4000 lb/in.^{2}); E = 2,600,000 kPa (400,000 lb/in.^{2}); light in weight, very durable, very smooth; liners and wrapping not required; good variety of fittings available or can use ductile or cast-iron fittings with adapters; can be solvent-welded; diameters from 100 to 375 mm (4 to 36 in.).	Maximum pressure = 2400 kPa (350 lb/in.^{2}); little reserve of strength for water hammer if ASTM D 2241 is followed (AWWA C900 includes allowances for water hammer); limited resistance to cyclic loading as the fatigue limit is very low; unsuited for outdoor use above ground.
High-density polyethylene (HDPE) pipe	Tensile strength (hydrostatic design basis) = (8600 to 11,000 kPa (1250 to 1600 lb/in.^{2}); light weight, very durable, very smooth; liners and wrapping not required for corrosion protection; usually jointing method is thermal butt fusion, which develops the full strength of the pipe; flanges can be provided; diameters from 100 through 1575 mm (4 through 63 in.); low installed cost.	Subject (as are many plastics, including PVC) to permeation by low-molecular-weight organic solvents and petroleum products; unsuited for manifold piping for pumping stations; scratches on the pipe wall can significantly reduce service life; requires careful bedding and compaction beneath the springline; cannot be solvent welded or threaded.
Asbestos cement pipe (ACP)	Yield strength: not applicable; design based on crushing strength (see ASTM C 296 and C 500); E = 23,500,000 kPa (3,400,000 lb/in.^{2}); rigid, light weight in long lengths; low cost; diameters from 100 to 1050 mm (4 to 42 in.); compatible with cast-iron fittings; pressure ratings from 1600 to 3100 kPa (225 to 450 lb/in.^{2}) for large pipe 450 mm (18 in.) or more.	Attacked by soft water, acids, sulfates; requires thrust blocks at elbows, tees, and dead ends; maximum pressure = 1380 kPa (200 lb/in.^{2}) for pipe up to 400 mm (16 in.); brittle and requires care to install.
Reinforced concrete pressure pipe (RCPP)	Several types available to suit different conditions (see Table 4-10); high strength for supporting earth loads; wide variety of sizes and pressure ratings; low cost; sizes from 600 to 3660 mm (24 to 144 in.).	Attacked by soft water, acids, sulfides, sulfates, and chlorides, often requires protective coatings; water hammer can crack outer shell, exposing reinforcement to corrosion and destroying its strength with time; maximum pressure = 1380 kPa (200 lb/in.^{2}).
Vitrified clay	Unexcelled for gravity sewers if properly bedded; no limit to life. Inert to ordinary chemicals; moderately smooth and strong; no leaks with polyurethane joints.	Costly compared to plastic (plastic has largely replaced clay); suitable only for gravity sewers; brittle and requires care to install; unsuited for unstable soils.

Table 4-12. Reinforced Concrete Pressure Pipe (RCPP)

Type	Specification	Description
Steel cylinder	AWWA C300	Steel cylinder within a reinforcing cage; walled inside and out with dense concrete, steel joint rings welded to cylinder, rubber gasket joint
Prestressed steel cylinder	AWWA C301	Cement-mortar-lined steel cylinder, wire wrapped and coated with mortar or concrete; two types differ slightly
Noncylinder	AWWA C302	Circumferentially reinforced concrete pipe without a steel cylinder and not prestressed
Bar-wrapped steel cylinder	AWWA C303	Cement-mortar-lined steel cylinder helically wound with continuous pretensioned steel bars and coated with mortar

be DIP, but if steel is used it should be plain-end pipe joined by mechanical couplings.

Steel and sometimes ductile iron pipe corrode on contact with some soils. Asbestos cement and concrete are attacked by acidic waters or by high-sulfate soils. In aggressive environments the pipe can be coated (see Section 4-3 for details).

In addition to the protective coatings, cathodic protection may also be required. Corrosion of iron and steel pipe is an electrochemical reaction. Cathodic protection consists of introducing a controlled DC current to oppose the natural destructive current. An alternative method is to bury sacrificial anodes that also reverse the current. Cathodic protection of iron and steel pipes is commonly used in aggressive soils or where there are stray electrolytic currents. Stray electrolytic currents can occur near buried power transmission lines and near other pipelines equipped with cathodic protection systems. Most petroleum and gas transmission lines have impressed-current cathodic protection. Any ferrous metal in the vicinity becomes an anode, loses metal, and

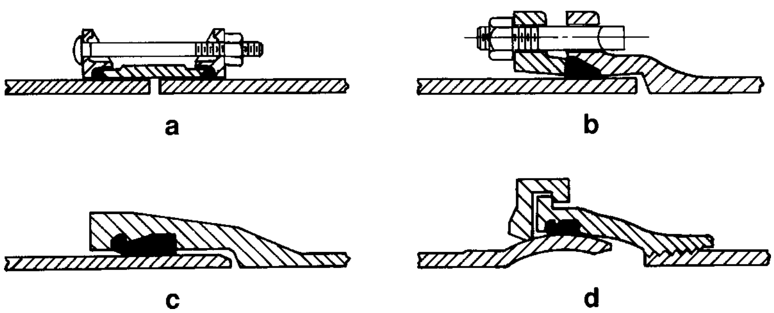


Figure 4-8. Couplings and joints for ductile iron pipe for buried service. (a) Sleeve (Dresser®) coupling per AWWA C219 (also for asbestos cement pipe); (b) mechanical joint; (c) push-on joint; (d) ball joint (also for exposed service). Courtesy of Wilson & Company, Engineers & Architects.

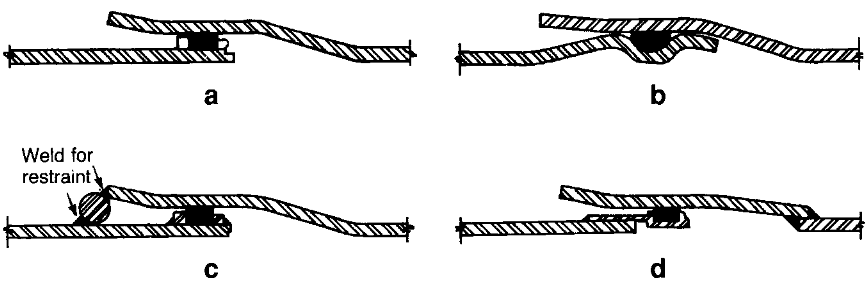


Figure 4-9. Rubber-gasketed push-on joints for steel pipe in buried service. (a) Fabricated joint; (b) rolled-groove joint; (c) tied joint; (d) Carnegie-shaped joint with weld-on bell ring. After AWWA M11 [9].

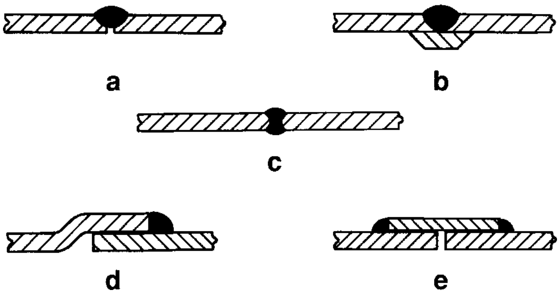


Figure 4-10. Welded joints for steel pipe in buried or exposed service. (a) Vee butt weld; (b) vee butt weld with weld ring; (c) double butt weld; (d) cup-type weld; (e) sleeve or butt strap weld. Courtesy of Wilson & Company, Engineers & Architects.

Figure 4-11. Couplings and joints for polyvinyl chloride (PVC) pipe in buried service. (a) Twin-gasketed coupling (also for asbestos cement); (b) bell and spigot joint. Courtesy of Wilson & Company, Engineers & Architects.

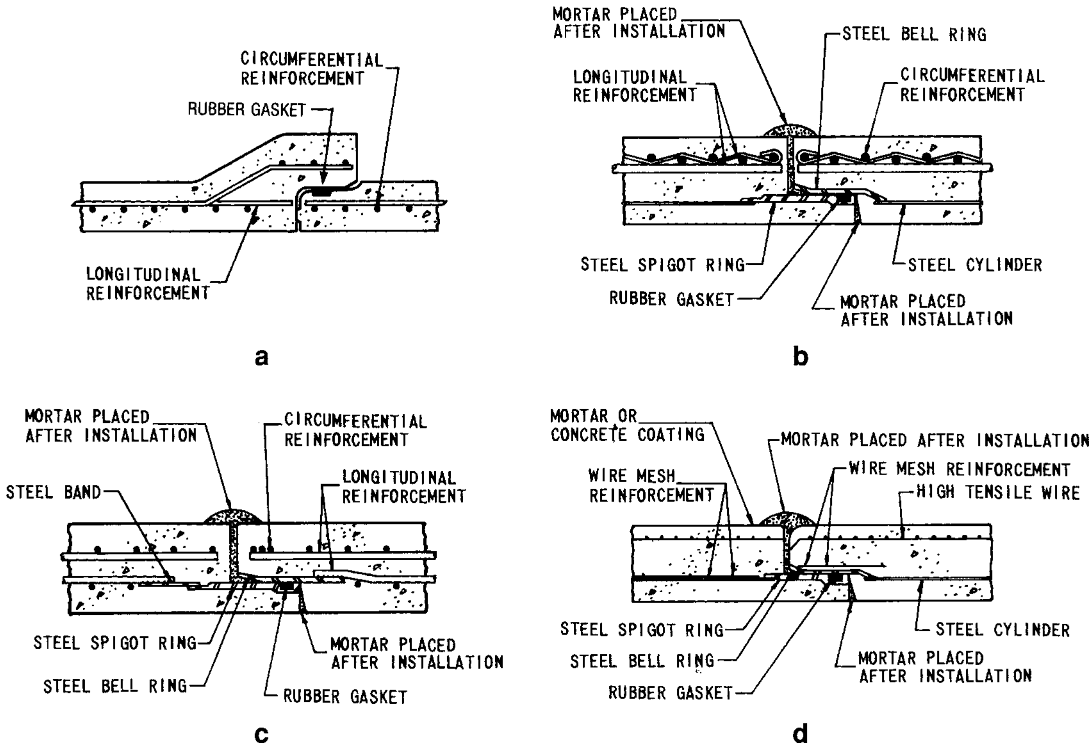


Figure 4-12. Push-on joints for concrete pipe in buried service. (a) Reinforced pressure pipe with rubber and concrete joint; (b) reinforced cylinder pipe with rubber and steel joint; (c) reinforced pressure pipe with rubber and steel joint; (d) prestressed embedded cylinder pipe with rubber and steel joint. Courtesy of Wilson & Company, Engineers & Architects.

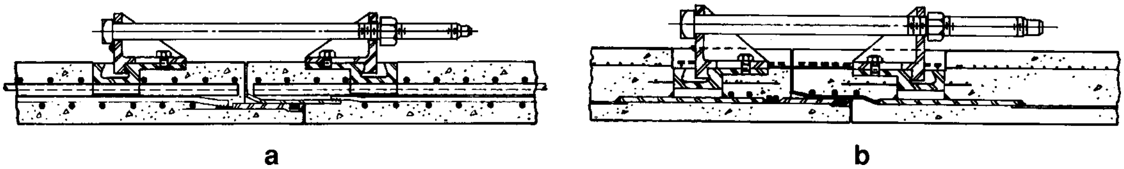


Figure 4-13. Restrained rigid joints for concrete pipe in exposed, buried, or subaqueous service. (a) Reinforced pressure pipe with rubber and steel joint; (b) reinforced cylinder pipe with rubber and steel joint. Courtesy of Wilson & Company, Engineers & Architects.

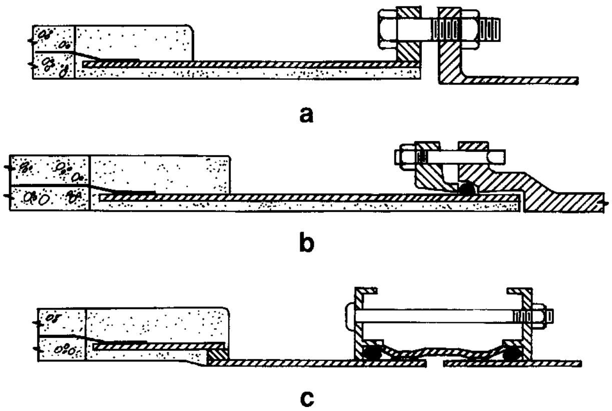


Figure 4-14. Adapters for connecting concrete pipe to ductile iron or steel pipe. (a) Flanged adapter; (b) mechanical joint adapter; (c) sleeve (Dresser®) coupling adapter. After Millcon Corp.

eventually fails unless it, too, is protected. If in doubt, consult a corrosion specialist. Minimum voltage requirements are determined by experience and field studies of ambient conditions such as moisture, chemical content, and resistivity of the soil. A more detailed discussion of cathodic protection is included in AWWA M11 [9] and NACE Standards RP-01-69 and RP 05-72.

Asbestos cement pipe is usually used without interior or exterior coatings. If corrosion can occur, consider other materials.

Reinforced concrete pressure pipe carrying corrosive water can be lined with coal-tar enamel or coal-tar epoxy. (Coal-tar enamels and epoxies are not used in potable water service.) These linings require maintenance, however, and should be used only where the pipeline can be removed from service periodically for inspection and where the pipeline is large enough in diameter to permit painting in place. Alternatively, the pipe can be lined by securing plastic sheets with T-shaped ribs to the pipe forms. The ribs lock the liner securely into the finished pipe. Field joints in the liner are made by heat-welding plastic strips after each joint of pipe has been installed in the trench. The pipe could also be encased in plastic to protect it from corrosive (high-sulfate or acidic) soils.

Fittings

The ductile and gray cast-iron fittings shown in Table 4-9 are also available with mechanical joints. Fittings for other kinds of pipe are given in Table 4-13. Ductile iron fittings are used for asbestos cement pipe and PVC (AWWA C900). Fabricated steel fittings can also be used.

Economics

The cost of the pipe at the job site is only one factor of many. Others include:

- Size
- Weight and length of sections, difficulty of handling, and machinery required
- Bedding. Flexible pipe, such as plastic sewer pipe, requires special bedding conditions (see Uni-Bell Handbook [12]).
- Type of joint and amount of field labor

Table 4-13. Common Types of Joints for Buried Pipe

Joint	Figure	Type of joint[a]	Type of pipe: Ductile iron	Steel	PVC	AC	RCPP
Flanged joint	4-1, 2	R	✓	✓			
Lugged sleeve coupling (Dresser™), harnessed	4-3a, b	F	✓	✓			
Sleeve coupling (Dresser™), unharnessed	4-8a	S	✓	✓			
Mechanical joint	4-8b	F	✓				
Push-on joint	4-8c	F	✓				
Push-on joint	4-9	F		✓			
Ball (also for exposed service)	4-8d	F	✓				
Twin-gasketed coupling	4-11a	F			✓	✓	
Bell and spigot	4-11b	F, S			✓		
Grooved-end coupling (Victaulic™)	4-4a	F	✓	✓			
Welded joint	4-10	R		✓			
RCPP							
Push-on	4-12	S					✓
Lugged, rigid	4-13	S					✓
RCPP to iron or steel							
Flanged	4-14a	R					✓
Mechanical or sliding	4-14b, c	S			✓		✓

[a]R = rigid; S = sliding or slip; F = flexible.

- Maintenance (frequency and difficulty of repair)
- Water hammer control (different for different materials)
- Friction coefficient
- Shoring requirement for laborers to enter trenches deeper (depending on soil stability) than 1.2 or 1.5 m (4 or 5 ft)—especially severe for trenches deeper than 3 m (10 ft). Consult OSHA regulations for shoring requirements.

Selection is further complicated by local conditions that significantly affect the cost of one material versus another, and these conditions may change radically with time. Carefully investigate the cost and availability of all types of suitable piping. For example, API steel pipe is made in great quantity and is frequently readily available and cheaper than other steel pipe. Coatings and linings can be applied to API pipe in the same manner as applied to any other steel pipe.

4-3. Ductile Iron Pipe (DIP)

Detailed descriptions of DIP, fittings, joints, installation, thrust restraint, and other factors relating to design as well as several important ASME/AWWA specifications are contained in the DIPRA handbook [11] (see also Section 4-1 and Tables 4-1 and 4-11).

Materials

Cast-iron pipe is manufactured of an iron alloy centrifugally cast in sand or metal molds. Prior to the early 1970s, most cast-iron pipe and fittings were gray iron, a brittle material that is weak in tension. But now all cast-iron pipe except soil pipe (which is used for plumbing) is made of ductile iron in which the graphite is formed into spheroids by the addition of magnesium and heat treatment, which makes it about as strong as steel. Cast-iron fittings are still available in gray iron as well as ductile iron.

Tolerances, strength, coatings and linings, and resistance to burial loads are given in AWWA C151/A21.51.

A special abrasion-resistant ductile iron pipe for conveying slurry and grit is available from several manufacturers. Regular ductile iron pipe (AWWA C151) has a Brinell hardness (BNH) of about 165. By comparison, abrasion-resistant ductile iron has a BNH of about 280. The sizes available are 150 through 600 mm (6 through 24 in.) per AWWA C151 in the standard AWWA wall thickness classes.

Available Sizes and Thicknesses

As shown in Tables B-1 and B-2, the available sizes range from 100 to 1350 mm (4 to 54 in.). The stan-

dard length is 5.5 m (18 ft) in pressure ratings from 1380 to 2400 kPa (200 to 350 lb/in.^{2}).

Thickness is specified by class, which varies from Class 50 to Class 56 (see the DIPRA handbook [11] or ANSI/AWWA C150/A21.50). Thicker pipe can be obtained by special order.

Joints

Buried joints should be of the mechanical joint or rubber gasket push-on type. Various types of restrained joints for buried service are also available.

Exposed joints should be flanged (AWWA C115 or ASME B16.1) or grooved end (AWWA C606) (refer to "Joints" in Section 4-1).

Gaskets

Gaskets for ductile iron or cast-iron flanges should be rubber, 3.2 mm ($^1/_8$ in.) thick.

Gaskets for grooved-end joints are available in ethylene propylene diene monomer (EPDM), nitrile (Buna N), halogenated butyl rubber, Neoprene™, silicone, and fluorelastomers. EPDM is commonly used in water service and Buna N in wastewater or sludge service.

Gaskets for ductile iron push-on and mechanical joints described in AWWA C111 are vulcanized natural or vulcanized synthetic rubber. Natural rubber is suitable for water pipelines but deteriorates when exposed to wastewater or sludge.

Flange Interfaces with Certain Flanged Valves

Wafer and lug style valves frequently cannot be used with ductile iron pipe because the inside bore of the flange is too large to provide sufficient sealing area for the valve face. Some ASME flanged butterfly valves and certain flanged double-door, silent, and single-disc check valves also cannot be used with ductile iron pipe for the same reason. The problem is that the valve body or flange seals must set against a metal surface to be effective. With some wall thickness classes of ductile iron pipe, a portion of the valve seal does not set against any surface because the pipe ID is so large. With such valves, a minimum wall thickness for the ductile iron pipe of Special Class 53 to Special Class 56 may have to be used. There is no general rule that can be applied. The inside diameter of the valve seal must be compared to the inside diameter of the pipe, and the proper pipe wall thickness must then be selected.

Fittings

Dimensions of cast-iron and ductile iron flanged fittings are covered by ASME B16.1, ASME B16.42 and AWWA C110, and fittings for abrasion-resistant pipe are generally furnished in one of the following categories:

- *Type 1:* Low alloy fittings with a minimum hardness of about 260 BNH, with flanged, grooved-end, or mechanical joints.
- *Type 2:* Special ductile iron fittings with a minimum hardness of about 400 BNH and flanged or mechanical joints.
- *Type 3:* Ni-Hard® fittings with a minimum hardness of about 550 BNH, with plain end or mechanical joints.

Linings and Coatings

For a discussion of linings, refer to Section 4-1.

Although ductile iron is relatively resistant to corrosion, some soils (and peat, slag, cinders, muck, mine waste, or stray electric current) may attack the pipe. In these applications, ductile iron manufacturers recommend that the pipe be encased in loose-fitting, flexible polyethylene tubes 0.2 mm (0.008 in.) thick (see ANSI/AWWA C105/A21.5). In some especially corrosive applications, a coating such as adhesive, hot-applied extruded polyethylene wrap may be required.

An asphaltic coating (on the outside) approximately 0.025 mm (0.001 in.) thick is a common coating for DIP in noncorrosive soils. In corrosive soils, consider the following coatings for protecting the pipe:

- Plastic wrapping (AWWA C105)
- Hot-applied coal-tar enamel (AWWA C203)
- Hot-applied coal-tar tape (AWWA C203)
- Hot-applied extruded polyethylene [ASTM D 1248 (material only)]
- Coal-tar epoxy
- Cold-applied tape (AWWA C209)
- Fusion-bonded epoxy (AWWA C213)
- Polyolefin tape coating (AWWA C214)
- Extruded polyolefin pipe coating (AWWA C215)
- Heat shrinkable cross-linked polyolefin coatings for special sections of pipe (AWWA C216)
- Polyurethane linings and coatings (AWWA C222)

Each of the coatings is discussed in detail in the AWWA specifications. Because each has certain lim-

ited uses, consult the NACE standards for the particular service.

4-4. Steel Pipe

The principal advantages of steel pipe include high strength, the ability to deflect without breaking, ease of installation, shock resistance, lighter weight than ductile iron pipe, ease of fabrication of large pipe, the availability of special configurations by welding, the variety of strengths available, and ease of field modification (see Section 4-1 and Tables 4-1 and 4-11).

Material

Conventional nomenclature refers to two types of steel pipe: (1) mill pipe, and (2) fabricated pipe.

Mill pipe includes steel pipe of any size produced at a steel pipe mill to meet finished pipe specifications. Mill pipe can be seamless, furnace butt welded, electric resistance welded, or fusion welded using either a straight or spiral seam. Mill pipe of a given size is manufactured with a constant outside diameter and an internal diameter that depends on the required wall thickness.

Fabricated pipe is steel pipe made from plates or sheets. It can be either straight- or spiral-seam fusion-welded pipe, and it can be specified in either internal or external diameters. Note that spiral-seam, fusion-welded pipe may be either mill pipe or fabricated pipe.

Steel pipe may be manufactured from a number of steel alloys with varying yield and ultimate tensile strengths. Internal working pressure ratings vary from 690 to 17,000 kPa (100 to 2500 lb/in.^{2}) depending on alloy, diameter, and wall thickness. Specify steel piping in transmission mains to conform to AWWA C200, in which there are many ASTM standards for materials (see ASME B31.1 for the manufacturing processes).

Available Sizes and Thicknesses

The available diameters range from 3 to 6000 mm ($\frac{1}{8}$ to 240 in.), although only sizes up to 750 mm (30 in.) are given in Tables B-3 and B-4. Sizes, thicknesses, and working pressures for pipe used in transmission mains range from 100 to 3600 mm (4 to 144 in.) and are given in Table 4-2 of AWWA M11 [9].

Manufacturers should be consulted for the availability of sizes and thicknesses of steel pipe (see also Table 4-2 in AWWA M11, which shows a great variety of sizes and thicknesses).

According to ASME B36.10,

- Standard weight (STD) and Schedule 40 are identical for pipes up to 250 mm (10 in.). All larger sizes of standard weight pipe have walls 9.5 mm ($\frac{3}{8}$ in.) thick [see Tables B-3 and B-4 for standard weight pipe from 3 to 750 mm ($\frac{1}{8}$ to 30 in.)]. For 300-mm (12-in.) pipe and smaller, the ID approximately equals the nominal diameter. For larger pipe, the OD equals the nominal diameter.
- Extra strong (XS) and Schedule 80 are identical for pipes up to 200 mm (8 in.). All larger sizes of extra-strong-weight pipe have walls 12.7 mm ($\frac{1}{2}$ in.) thick.
- Double extra strong (XXS) applies only to steel pipe 300 mm (12 in.) and smaller. There is no correlation between XXS and schedule numbers. For wall thickness of XXS, which (in most sizes) is twice that of XS, see ANSI B36.10.

For sizes 350 mm (14 in.) and larger, most pipe manufacturers use spiral welding machines and, in theory, can fabricate pipe to virtually any desired size. But in practice most steel pipe manufacturers have selected and built equipment to produce given OD sizes. For example, one major U.S. manufacturer uses a 578.6-mm (22 $\frac{25}{32}$ -in.) OD cylinder for a nominal 525-mm (21-in.) pipe. Any deviation from manufacturers' standards is expensive. To avoid confusion, either show a detail of pipe size on the plans or tabulate the diameters in the specifications. For cement-mortar-lined transmission mains, AWWA C200, C205, C207, and C208 apply.

Manifold piping in pumping stations should be considered a large special fitting or a series of fittings connected together. Dependence on AWWA C200 alone is inadequate for designing such headers or manifolds because it does not address the following:

- Reinforcement at openings (tees, laterals, branches); see "Pipe Wall Thickness" in Section 4-8
- Wall thickness for grooved-end couplings (see Table 4-5)
- Thrust harness lugs for flexible pipe couplings (see Example 4-1)
- Additional wall thickness required at elbows and other fittings because of stress intensification factor (see the following subsection on "Fittings").

Thus, the design of steel manifolds depends on a combination of a number of factors.

Steel pipe must sometimes either be reinforced at nozzles and openings (tees, wye branches) or a greater

wall thickness must be specified. A detailed procedure for determining whether additional reinforcing is required is described in Chapter II and Appendix H of ASME B31.3. If additional reinforcement is necessary, it can be accomplished by a collar or pad around the nozzle or branch, a wrapper plate, or crotch plates. These reinforcements are shown in Figure 4-15, and the calculations for design are given in AWWA M11 [9].

Steel manifold piping 500 mm (20 in.) and smaller must inevitably be made in accordance with ASME B36.10. Material would usually conform to ASTM A 53, A 135, or API 5L. For pipe 600 mm (24 in.) and larger, a fabricator might elect to use pipe conforming to ASTM A 134 or A 139 as well. As shown in Tables B-3 and B-4, the size of the pipe (nominal diameter) approximates the ID for 300-mm (12-in.) pipe and smaller, but size equals the OD for 350-mm (14-in.) pipe and larger. For pipe larger than 500 mm (20 in.), steel pipe size can be specially fabricated to any size.

The ID of steel pipe should match the ID of iron valves, particularly butterfly valves. One way is to select pipe one size larger than the nominal pipe size and line it with cement mortar so that the ID of a mortar-lined pipe matches the nominal pipe size.

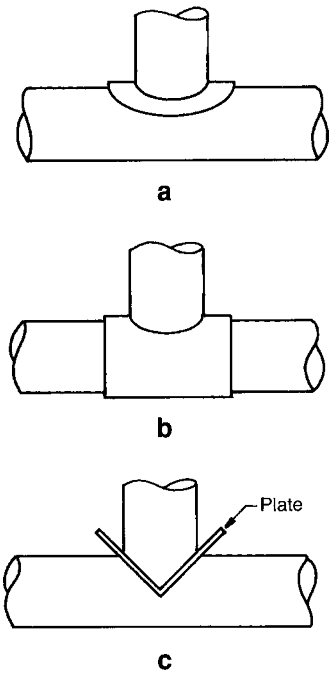


Figure 4-15. Reinforcement for steel pipe openings. (a) Collar plate; (b) wrapper plate; (c) crotch plates.

Consider, for example, a requirement for a steel header with a nominal 400-mm (16-in.) ID. A pipe fabricator could use an ASME B36.10 standard weight pipe with a true OD of 457 mm (18 in.). The wall thickness of the steel cylinder is 9.5 mm (0.375 in.), which gives an ID of 438 mm (17.25 in.). A 13-mm ($\frac{1}{2}$ -in.) mortar lining provides a net ID of 413 mm (16.25 in.), which is close to the desired size, and if the mortar lining were 16-mm ($\frac{5}{8}$ -in.) thick, the ID would be exactly 406 mm (16 in.). As shown in Table 4-6, the minimum thickness of the cement-mortar lining is 7.9 mm ($\frac{5}{16}$ in.).

Obtaining internal diameters in even sizes for steel pipe smaller than 350 mm (14 in.) can be done by using plastic linings. For example, a 250-mm (10-in.) standard weight pipe has an OD of 273 mm (10.75 in.) and a wall thickness of 9.27 mm (0.365 in.); for a minimum cement-mortar thickness of 6.4 mm ($\frac{1}{4}$ in.), the ID is only 242 mm (9.5 in.). A practical alternative is to use a plastic lining, such as fusion bonded epoxy, that makes the ID equal to the nominal 250-mm (10-in.) diameter.

Some benefits of using standard weight ASME B36.10 pipe are that (1) the pipe is readily available, and (2) the wall thickness is:

- often sufficient, so that reinforcements at openings (wrappers, collars, or crotch plates) are unnecessary;
- often sufficient in pumping stations for use with thrust harnesses on flexible pipe couplings, so that additional wall thickness is unnecessary; and
- sufficient for use with AWWA C606 grooved-end couplings without additional reinforcement at the pipe ends.

Joints

Joints for steel pipe are listed in Table 4-13. For buried service, bell and spigot joints with rubber gaskets or mechanical couplings (with or without thrust harnesses) are preferred. Welded joints are common for pipe 600 mm (24 in.) and larger. Linings are locally destroyed by the heat of welding, so the ends of the pipe must be bare and the linings field-applied at the joints. The reliability of field welds is questionable without careful inspection, but when properly made they are stronger than other joints.

Fittings

Specifications for steel fittings can generally be divided into three classes, depending on the joints used and the pipe size:

- Threaded (ASME B16.3 or B16.11) but only for pipe 75 mm (3 in.) and smaller
- Flanged, welded (ASME B16.9)
- Fabricated (AWWA C208).

Fittings larger than 75 mm (3 in.) should conform to ASME B16.9 ("smooth" or wrought) or AWWA C208 (mitered); avoid threaded fittings larger than 75 mm (3 in.). The ASME B16.9 fittings are readily available up to 300 to 400 mm (12 to 16 in.) in diameter. Mitered fittings are more readily available and cheaper for larger fittings.

The radius of a mitered elbow can range from 1 to 4 pipe diameters. The hoop tension concentration on the inside of elbows with a radius less than 2.5 pipe diameters may exceed the safe working stress. This tension concentration can be reduced to safe levels by increasing the wall thickness, as described in ASME B31.1, AWWA C208, and *Piping Engineering* [16]. Design procedures for mitered bends are described in ASME B31.1 and B31.3.

Gaskets

Gaskets for steel flanges are usually made of cloth-inserted rubber either 1.6 mm ($^1\!/_{16}$ in.) or 3.2 mm ($^1\!/_8$ in.) thick and are of two types: (1) ring (extending from the inside diameter of the flange to the inside edge of the bolt holes), and (2) full face (extending from the inside diameter of the flange to the outside diameter).

For mechanical and push-on joints, refer to "Gaskets" in Section 4-4.

Flange Interfaces with Certain Flanged Valves

Wafer- and lug-style valves frequently cannot be used with mortar-lined steel pipe because the inside bore of the flange is too large for sufficient sealing area at the valve face. Some ASME flanged butterfly valves and certain flanged double-door, silent, and single-disc check valves also cannot be used with mortar-lined steel pipe for the same reason. The problem is that the valve body or flange seals must set against a metal surface to obtain an effective seal. With some mortar-lined pipe, a portion of the valve seal sets against the mortar lining. With such valves, steel pipe per ANSI B3.10 with linings such as epoxy or polyurethane should be used with the proper associated flanges.

Linings and Coatings

Cement mortar is an excellent lining for steel pipe (see Table 4-6 and AWWA C205 for the thickness required; refer to "Linings and Coatings" in Section 4-1 and to "Linings" in Section 3-3 for further discussion).

Steel pipe can also be protected with mortar, but soil conditions affect the necessary thickness of the mortar coating. Corrosive soils may require mortar coatings of 25 mm (1 in.) or more regardless of pipe size. Alternatively, a hot-applied extruded polyethylene coating with heat-shrink jackets for joints that complies with ASTM D 1248 and AWWA C216 is an excellent coating for both steel and ductile iron pipe. Other coatings frequently used with buried steel pipe include polyethylene tape per AWWA C214, extruded polyolefin per AWWA C215, and polyurethane per AWWA C222.

As another alternative, epoxy-lined and -coated steel pipe or nylon-lined and -coated steel pipe could be used. Because these linings are only 0.3- to 0.6-mm (0.012- to 0.020-in.) thick, the ID of bare pipe is only slightly reduced by such linings (see Tables B-1 and B-2). Epoxy-lined steel pipe is covered by AWWA C203, C210, and C213 standards. Nylon-lined and -coated steel pipe is described in AWWA C224. Additional linings and coatings for steel pipe include:

- Polyolefin tape coating (AWWA C214)
- Extruded polyolefin pipe coating (AWWA C215)
- Heat-shrinkable cross-linked polyolefin coatings for special section of pipe (AWWA C217)
- Cold-applied petrolatum tape (AWWA C217)

The supplier must be consulted to determine the limitations of sizes and lengths of pipe that can be lined with epoxy. Flange faces should not be coated with epoxy if flanges with serrated finish per AWWA C207 are specified.

4-5. Plastic Pipe

In the United States, where it is used in both water and wastewater service, polyvinyl chloride (PVC) is the most commonly used plastic pipe. It is a polymer extruded under heat and pressure into a thermoplastic that is nearly inert when exposed to most acids, alkalies, fuels, and corrosives, but it is attacked by ketones (and other solvents) sometimes found in industrial wastewaters. It has a high strength-to-weight ratio and is durable and resilient, but it lacks the stiffness necessary for exposed service and is susceptible to flotation in groundwater conditions. Most types of PVC pipe should not be exposed to direct sunlight (see ANSI/AWWA C900). Some designers have had poor experiences with solvent-welded

flanges and no longer use them on buried PVC pressure pipe in any size. PVC pipe conforming to AWWA C900 or C905 with ductile iron fittings can be used for buried service in sizes 1000 mm (24 in.) or smaller. For larger pipe, AWWA C905 may not be sufficiently conservative in some applications. A proper analysis of static and transient internal and external pressures is required. Note that the fatigue limit for PVC is very low, and the pipe could be vulnerable to rupture from excessive pressure cycles. HDPE pipe is better for cyclic loading.

High-density polyethylene (HDPE) pipe and polyvinyl chloride (PVC) are both suitable for use in potable water service and raw wastewater service. HDPE, in general, is much less sensitive to surge pressures than PVC because of the long molecular chains in the plastic material. See AWWA C906 for the method of calculating the required pressure class. The heat-fused butt joint system for HDPE pipe is a satisfactory joint that is easy to install in the field.

A key consideration in specifying either PVC or HDPE is their susceptibility to scratches in the pipe wall that can be caused by dragging the joined sections of pipe for long distances over the ground. Rocks usually cause scratches about 3-mm ($^1/_8$ in.) deep and sometimes much deeper. In HDPE, scratches 0.19-T (T is the pipe wall thickness) deep reduce the cyclic loading life span by 90%, whereas scratches 0.04-T deep have little influence on fatigue strength. Specifying that pipe must be dragged over smooth surfaces, such as railroad ties, is one way to avoid deep scratches.

In addition, careful attention must be paid in specifications and construction inspection to ensure that the pipe is fully bedded, with no voids or poorly compacted areas beneath the springline. Because of its permeability to organic solvents with low molecular weight (gasoline, for example), HDPE should not be used for potable water beside roads where gasoline could penetrate into the ground. Other than the above considerations, HDPE is ideal for systems with moderate pressures. Although the cost of the pipe itself is about the same as it is for steel, its installed cost is likely to be less than that of steel because of the ease and speed of handling.

Many designers refuse to specify PVC pipe for the small (75 mm or 3 in. and smaller) piping that, for example, conveys seal water inside pumping stations. The primary reason is due to the difficulty of getting the PVC pipe installers to do a proper job. Specific problems that occur include:

- Inadequate or excessive amounts of solvent cement applied to the joints. See ASTM D 2855.
- Inadequate curing time allowed before moving pipe with solvent-cemented joints. It is wise to specify eight hours of curing time before pipe can be moved (in spite of manufacturers' claims that two hours is adequate).
- Forcing and springing the pipe into position to make up for errors in initially installing the pipe. The overstress in the pipe can result in breakage after a time lapse of less than a year. Adequate inspection *during installation* could prevent the overstress, but inspection after installation is useless.

Because of these problems, copper, steel, or stainless steel for the small-size service piping is a better choice where applicable. These metallic piping systems are far more resistant to poor installation practices than is plastic piping. Furthermore, it is preferable to avoid exposed plastic pipe in any pumping station because of the hazard of melting (or even supporting combustion) in fires (see NFPA 820). Plastic pipe may, however, be the only practical selection for chlorine solution or other chemical services.

Other plastic piping materials include:

- Acrylonitrile-butadiene-styrene (ABS)
- Chlorinated polyvinyl chloride (CPVC)
- Polypropylene (PP)
- Polyethylene (PE)
- Polyvinylidene fluoride (KYNAR™ or PVDF)
- Fiberglass-reinforced plastic (FRP).

Most of these materials are used for corrosive chemicals, but some may have special uses for water, wastewater, or sludge. Consult manufacturers for properties, joints, and fittings. Investigate installations before specifying such materials. The impact strength of most plastics decreases when exposed to sunlight. Consequently, be wary of using plastic pipe in exposed outdoor service unless it is coated with an ultraviolet-resistant paint such as a polyurethane. However, FRP should be installed only where it is exposed and easily inspected; if it is buried, it should be used with both very conservative safety factors and considerable caution.

Available Sizes and Thicknesses

The range of available sizes is given in Table 4-11, but consult the appropriate ASTM standards (D 1785 and D 2241) to find exact sizes, thicknesses, and pressure ratings.

Wall thickness design for PVC pipe is defined by two separate sets of nomenclature: (1) standard dimension ratios (SDRs), and (2) schedules. The ratio of pipe outside diameter to wall thickness is called the "SDR." For PVC pipe, SDR is calculated by dividing the average outside diameter of the pipe by the minimum wall thickness. The available thicknesses are SDR 35 through SDR 14 (refer to ANSI/ASTM D 2241 and AWWA C900 and C905 for a complete discussion of SDRs and corresponding pressure ratings; refer to ANSI/ASTM D 1785 for a complete discussion of the wall thicknesses and pressure ratings for the various schedules of PVC pipe).

Joints

Joints of PVC can be solvent-welded, flanged, push-on with rubber gaskets, or threaded. Threads should be used only for 100-mm (4-in.) pipe and smaller, and the thinnest threaded pipe should be Schedule 80 (see ANSI/ASTM D 2464). Solvent-welded Schedule 40 pipe is stronger than threaded Schedule 80 pipe, and solvent-welded Schedule 80 pipe is the strongest of all.

Gaskets

Gaskets for PVC flanges should be 3.2-mm (1/8-in.)-thick ethylene propylene rubber (EPR), full-faced, with a Durometer hardness of 50 to 70, Shore A. When connecting PVC flanges to raised-face metal flanges, remove the raised face on the connecting metal flange to protect the PVC flange from the bolting moment.

Valve Pressure Rating

The maximum allowable working pressure of PVC valves is even lower than that of Schedule 80 threaded piping. Most PVC valves are rated at 1040 kPa (150 lb/in.^{2}) at a temperature of 38°C (100°F). The maximum recommended pressure for any flanged plastic pipe system (PVC, CPVC, PP, PVDF) is the same.

Fittings

Threaded, flanged, or solvent-welded fittings are used in exposed and buried service for piping smaller than 100 mm (4 in.). Class 125 mechanical joint ductile or cast-iron fittings should be used in buried applications for pipes 100 mm (4 in.) and larger. The adapters must be installed in the manner prescribed by the manufacturer.

Fittings for PVC pipe include tees, crosses, wyes, reducers, and 22.5-, 45-, and 90-degree bends.

Criteria for Selection of PVC Pressure Pipe

Recommended criteria for using PVC pressure pipe is as follows:

- *Seventy-five mm (3 in.) and smaller, exposed and buried service:* Schedule 80 per ASTM D 1784 and D 1785. Other ASTM standards applicable to PVC pipe are D 2464, Schedule 80 Threaded Fittings; D 2467 Schedule 80 Socket Type Plastic Fittings; and D 2564 Solvent Cements for PVC Pipe. Based on the experience and observations of various engineers, this pipe should not be used in applications in which the operating pressure exceeds 550 kPa (80 lb/in.^{2}). Use copper or steel pipe for water service; use PVC pipe only for chemical service.
- *One hundred and 150 mm (4 and 6 in.), exposed service:* Schedule 80 per ASTM D 1748 and D 1785. Use solvent-welded—not threaded—joints. Based on the experience and observations of various engineers, this pipe should not be used in applications in which the operating pressure exceeds 345 kPa (50 lb/in.^{2}). Use PVC only for chemical service piping. Use steel or ductile iron for water piping and other services.
- *Two hundred mm (8 in.) and larger, exposed service:* PVC pipe should not be used at all in such sizes.
- *One hundred through 300 mm (4 through 12 in.), buried service:* AWWA C900. Fittings should be cast or ductile iron.
- *Three hundred fifty through 900 mm (14 through 36 in.), buried service:* AWWA C905. This standard may not be sufficiently conservative in its pressure ratings for some applications. For a pressure class of 1035 kPa (150 lb/in.^{2}), consider using an SDR of 18 for 350- to 600-mm (14- to 24-in.) pipe and an SDR of 21 for 750- to 900-mm (30- to 36-in.) pipe. These SDR values are based on a surge allowance of 345 kPa (50 lb/in.^{2}) over the pipe pressure class, with a safety factor of 2.5 for 350- to 600-mm pipe and a safety factor of 2.0 for 750- to 900-mm pipe.
- *Never use plastic pipe for air or compressed gas service*. (It melts.)

4-6. Asbestos Cement Pipe (ACP)

Asbestos cement pipe, available in the United States since 1930, is made by mixing portland cement and asbestos fiber under pressure and heating it to produce a hard, strong, yet machinable product. Over 480,000 km (300,000 mi) of ACP is now in service in the United States, and, according to a mid-1970s survey, more than a third of pipe then being installed was ACP [8].

In recent years, attention has been focused on the hazards of asbestos in the environment and, particularly, in drinking water. The debate continues with one set of experts advising of the potential dangers and a second set of experts claiming that pipes made with asbestos do not result in increases in asbestos concentrations in the water. Studies have shown no association between water delivered by ACP and any general disease, but fear may be as important as reality, so consult with owners and local health authorities before deciding whether to specify ACP.

In January 1986, the EPA published a proposed regulation banning further manufacture and installation of ACP, but it was made clear that the proposed action was based on the hazard of inhaling asbestos fibers during manufacture and installation of the pipe—not because it contaminated drinking water. For a while after October 1987, the EPA had been reassessing the January proposal [8], but the proposed ban was overturned by a U.S. Federal Appeals court in 1991. Consequently, ACP is still being manufactured and used.

Available Sizes and Thicknesses

Available sizes are given in Table 4-11. Refer to ASTM C296 and AWWA 401, 402, and 403 for thicknesses and pressure ratings and AWWA C401 and C403 for detailed design procedures. AWWA C401, for 100- to 400-mm (4- to 16-in.) pipe, is similar to AWWA C403 for 450- to 1050-mm (18- to 42-in.) pipe. The properties of asbestos cement for distribution pipe (AWWA C400) and transmission pipe (AWWA C402) are identical. Nevertheless, under AWWA C403 the *suggested* minimum safety factor is 2.0 for operating pressure and 1.5 for external loads, whereas the safety factors under AWWA C402 are 4.0 and 2.5, respectively. So the larger pipe has the smaller safety factors.

Section 4 in AWWA C403 justifies this difference on the basis that surge pressures in large pipe tend to be less than those in small pipes. But surge pressures are not necessarily a function of pipe diameter (see Chapters 6 and 7). The operating conditions, including surge pressures, should be evaluated before the pipe class is selected. It is the engineer's prerogative to select which of the safety factors should apply. Per AWWA C400, safety factors should be no less than 4.0 and 2.5 if no surge analysis is made. The low safety factors given in AWWA C403 should be used only if all loads (external, internal, and transient) are carefully and accurately evaluated.

Joints and Fittings

The joints are usually push-on, twin-gasketed couplings (Figure 4-11a), although mechanical and rubber gasket push-on joints can be used to connect ACP to iron fittings.

Ductile iron fittings conforming to ANSI/AWWA C110/A21.10 are used with ACP, and adapters are available to connect ACP to flanged or mechanical ductile iron fittings. Fabricated steel fittings with rubber gasket joints can also be used. ACP may be tapped with corporation stops, tapping sleeves, or service clamps.

4-7. Reinforced Concrete Pressure Pipe (RCPP)

Reinforced concrete pressure pipe can be made to meet special strength requirements by using a combination of a steel cylinder and steel cages, by using one or more steel cages, or by prestressing with spiral rods (as shown in Figures 4-12 and 4-13 and in Table 4-12).

A distressing number of failures of prestressed concrete cylinder pipe (AWWA C301) have occurred in the United States within the last two or three decades. The outer shell of concrete cracks, which allows the reinforcement to corrode and subsequently fail. Therefore, do not depend on AWWA specifications or on manufacturers' assurances, but do make a careful analysis of internal pressure (including water hammer) and external loads. Make certain that the tensile strain in the outer concrete shell is low enough so that cracking either will not occur at all or will not penetrate to the steel under the worst combination of external and internal loading. Note again that strength is not the issue. The concern is for the tensile strain in the outermost concrete. When cracks do occur, they appear to penetrate to a depth of about 18 to 25 mm (0.75 to 1 in.), so a clear cover of at least 38 mm (1.5 in.) should be specified. Water hammer must be carefully analyzed and controlled. Wire-wrapped, prestressed concrete pipe has the worst history of failure. Reinforced concrete pipe is better, but if any significant water hammer can occur, ductile iron or steel is

best. For salt or brackish water, PVC, HDPE, or ductile iron with cathodic protection should be used.

Sizes and Joints

Sizes range from 600 to 3600 mm (24 to 144 in.), as shown in Table 4-11.

Joints are shown in Figures 4-12 and 4-13. Joints for buried service include:

- Rubber gasket and concrete (Figure 4-12a)
- Rubber gasket and steel (Figure 4-12b, c, d)
- Lugged rubber gasket and steel (Figure 4-13).

Note that rubber gasket and concrete joints (Figure 4-12) should be used only for pressures less than 380 kPa (55 lb/in.^{2}). The other joints can withstand pressures up to 2800 kPa (400 lb/in.^{2}), but consult manufacturers for joint pressure ratings.

Wall Thickness Design

The wall thickness design should be based on both external trench loads and on a detailed hydraulic analysis of the pumping system, including water hammer and surge pressure. Surge pressures should not be able to induce even hairline cracks in the external surface. A variety of wall thicknesses and reinforcing designs are available for each pipe diameter.

Some of these types of pipe have severe internal pressure limitations. The AWWA standards cited in Table 4-12 should be consulted.

Fittings

Standard fittings are tees, crosses, 45-degree wyes, eccentric reducers, concentric reducers, flange and mechanical joint adapters to connect concrete pipe to steel or ductile iron (Figure 4-14), and bends from 7.5 to 90 degrees in 7.5-degree steps. RCPP can be tapped by drilling a hole into the pipe and then strapping a threaded or flanged tapping saddle to the pipe. Alternatively, a threaded steel outlet connection can be cast in the pipe wall during manufacture. Some pipe designers prefer to use fabricated steel specials for fittings and for any pipe segment containing an outlet or nozzle.

Linings

Pipe that is only partly full of wastewater or in which air can enter by any means (temperature changes, vortices in the wet well, or leaks) requires protection against corrosion caused by bacterial action. Sometimes, pipe can fail in only a few years. Anaerobic, sulfate-reducing bacteria (such as *Desulfovibrio*) living in the slime below the water surface utilize the oxygen in sulfate and create hydrogen sulfide, which escapes from the water surface to the atmosphere above. Aerobic bacteria (e.g., *Thiobacillus*) on the sides and soffit of the pipe convert the hydrogen sulfide to sulfuric acid at a pH of 2 or even less. *Thiobacillus* cannot live in pipe that is always full, so it is important to keep force mains (whether concrete, steel, or ductile iron) full at all times. Therefore, where a force main terminates at a manhole, design the connection so that no part of the force main is exposed to air. Either run the force main up through the bottom of the manhole, or if it enters from the side, either (1) set the invert of the downstream sewer above the crown of the force main, or (2) use plastic pipe near the manhole.

Lining and coating systems for pipes include various brush- or spray-applied epoxies, resins, polyurethanes, and coal tars. Coatings such as coal-tar epoxies have a history of poor performance where hydrogen sulfide attack can occur. A more effective (and more costly) lining system consists of PVC liner sheets (such as Ameron Tee-Lock®) and high-density polyethylene (HDPE) that are made with keys or ribs projecting from one side of the sheet. The smooth PVC face is attached directly to the forms prior to casting the concrete. The ribs project into the concrete and form a mechanical bond with it. PVC liners can be cast around the entire circumference of pipe and the longitudinal joint heat-fused to form a 360-degree liner, but it is cheaper to line only the portion above low-water level. These liners offer maximum protection for concrete pipe and manholes subject to corrosive environments, and they have a 40-year record of success. If the workmanship and inspection of the welding at the joints is good, the system will have a long, trouble-free life. The designer's problems are: (1) to write specifications that ensure workers are not hurried through this important task, and (2) to ensure that inspection is of high caliber. PVC liners are usually applied only to pipes in sizes of 900 mm (36 in.) and larger and usually only to the upper 270 degrees of the full circumference. Corrosion-resistant materials such as vitrified clay pipe, PVC, or HDPE should be used for smaller sewer pipes. See Section 25-11 for more discussion of concrete protection.

4-8. Design of Piping

The emphasis in this section is on the piping within the pumping station (i.e., exposed) and problems such as

pipe thickness, flange bolts, and pipe supports. The design of external (i.e., buried) piping is limited to generalities because there is an extensive body of excellent literature on such problems [9, 11–15].

In selecting a pipe size, be aware that it is the outside—not the inside—diameter that is fixed. The inside diameter varies with the wall thickness, whereas the outside diameter does not. This is true for all sizes of ductile iron, copper, brass, and plastic pipe. It is also true for most steel pipe used in pumping stations.

Exposed Piping

The selection of pipe size governed by hydraulics is given in Example 3-1. Other practical considerations that depend on available pipe diameters and wall and lining thicknesses are discussed in Sections 4-3 and 4-4 under "Available Sizes and Thicknesses."

Tie Rods

Mechanical couplings, mechanical joints, and push-on joints (Figures 4-8, 4-9, 4-11, and 4-12) must be restrained from sliding apart either by soil friction (if the pipe is buried) or by tie rods (if the pipe is exposed) as shown in Figures 4-3 and 4-13. Always design rods and bolts for tensile stress at the net cross-sectional area at the root of the threads (see Table 4-14).

Table 4-14. Root Areas for Threaded Rods

Nominal rod diameter		Root area for coarse threads	
mm	in.	mm^2	in.^{2}
10	3⁄8	43.9	0.068
13	1⁄2	81.3	0.126
16	5⁄8	130.3	0.202
19	3⁄4	194.8	0.302
22	7⁄8	270.3	0.419
25	1	356.1	0.552
28	1 1⁄8	447.1	0.693
31	1 1⁄4	573.5	0.889
35	1 3⁄8	679.4	1.053
38	1 1⁄2	834.2	1.293
41	1 5⁄8	977.4	1.515
44	1 3⁄4	1125	1.744
47	1 7⁄8	1321	1.048
50	2	1479	2.292
63	2 1⁄2	2397	3.716
75	3	3626	5.672

Bending

Pipe between tie rods is subject to bending due to the weight of metal and the liquid contents. The formula for bending stress is

$$s = \frac{Mc}{I} = \frac{M}{S} \tag{4-1}$$

where s is stress in Newtons per square meter (pounds per square inch), M is moment in Newton-meters (pound-inches), c is distance ($d/2$) from the centerline to outermost surface, I is moment of inertia in meters to the fourth power (inches to the fourth power), and S is the section modulus, I/c, in cubic meters (cubic inches). For circular pipes,

$$I = \frac{\pi(d^4 - d_i^4)}{64} \tag{4-2}$$

where d is pipe outside diameter, and d_i is pipe inside diameter. For rectangular cross-sections,

$$I = \frac{bd^3}{12} \tag{4-3}$$

where b is width in meters (inches) and d is depth in meters (inches).

Example 4-1
Design of Tie Rods for a Sleeve Pipe Coupling

Problem: Determine the number and size of tie rods required for a sleeve coupling in a 600-mm (24-in.) DIP under a maximum static and surge pressure of 1070 kPa (155 lb/in.^{2}).
Solution: The outside diameter of the pipe is 655 mm (25.8 in.); because the pressure acts on the gross face area of the end of the pipe, use the OD (not the ID) to compute thrust.

SI Units

$$\text{Pipe area} = \frac{\pi(0.655)^2}{4} = 0.337\,\text{m}^2$$

U.S. Customary Units

$$\text{Pipe area} = \frac{\pi(25.8)^2}{4} = 523\,\text{in.}^2$$

$$\text{Thrust} = 1070\,\text{kPa} \times 0.337\,\text{m}^2 = 361\,\text{kN}$$

$$\text{Thrust} = 155\,\text{lb/in.}^2 \times 523\,\text{in.}^2 = 81,100\,\text{lb}$$

Materials recommended in AWWA M11 [9] are (1) rods or bolts (ASTM A 193, Grade B7 or equivalent with a yield stress of 725,000 kPa (105,000 lb/in.^{2}); (2) welded lugs [ASTM A 283, Grade B or ASTM A 285, Grade C or equivalent with a yield stress of 186,000 kPa (27,000 lb/in.^{2}) or ASTM A 36 steel with a yield stress of 248,000 kPa (36,000 lb/in.^{2})]; see the ASTM standards for stresses.

Reduce the tensile yield stress at the root of the threads by a safety factor of 2 to obtain the allowable working stress. The total root area required is then

$$A = \frac{361\,\text{kN}}{362{,}000\,\text{kPa}} = 0.0010\,\text{m}^2$$

$$A = \frac{81{,}000\,\text{lb}}{52{,}000\,\text{lb/in.}^2} = 1.56\,\text{in.}^2$$

Many combinations of rod sizes and numbers can be used. Try several and choose a suitable one. (Note, however, that the resulting bending stresses in the pipe shell should be checked. Using two large-diameter rods, for example, causes greater bending stresses than using four smaller ones. Sometimes the pipe wall thickness must be increased to reduce the stress.) From Table 4-14, the root area of a 22-mm ($\frac{7}{8}$ -in.) bolt is 270 mm^2 or 0.000270 m^2 (0.419 in.^{2}); thus, the number of rods required is:

$$N = \frac{0.0010\,\text{m}^2 \times 10^6}{270\,\text{mm}^2} = 3.7$$

$$N = \frac{1.56\,\text{in.}^2}{0.419\,\text{in.}^2} = 3.7$$

Use four 22-mm rods.

Use four $\frac{7}{8}$ -in. rods.

Note that the rod material is high-strength steel (ASTM 193 Grade B7). If lower strength steel (carbon or stainless) were used, the number and/or size of rods would be quite different.

The rods can be supported by lugs welded to the pipe, which is excellent with steel pipe but can be done with ductile iron pipe only if great care is taken not to overheat the iron. An alternate detail for ductile iron pipe is to bolt an "ear" (as in Figure 4-3a) to the pipe flange so that the tie rod clears both the pipe flange and the pipe coupling. From the DIPRA handbook [11] and manufacturers' catalogs, the critical dimensions are (1) flange bolt circle [749 mm (29.5 in.)], (2) number and size of bolts [20 bolts, 32 mm (1 $\frac{1}{4}$ in.) in diameter], (3) flange OD [813 mm (32.0 in.)], (4) flange thickness [48 mm (1.88 in.)], and (5) coupling OD [782 mm (30.8 in.)]. If a clearance between the rod and flange of, say, 8 mm ($\frac{5}{16}$ in.) is chosen, the rod is centered 19 mm ($\frac{3}{4}$ in.) from the flange OD. The ear can be designed (with an adequate safety factor applied to the yield stress) to withstand the calculated force in the tie rod, but a more sensible design is to size the ear so that yield stress is reached simultaneously in the ear and tie rod. The bending moment in the ear is equal to the lever arm times the force in the tie rod.

SI Units

$$M = 0.019\,\text{m}(2.7 \times 10^{-4}\,\text{m}^2 \times 7.25 \times 10^8\,\text{N/m}^2) = 3.720\,\text{N}\cdot\text{m}$$

U.S. Customary Units

$$M = 0.75\,\text{in.}(0.419\,\text{in.}^2 \times 105{,}000\,\text{lb/in.}^2) = 33{,}000\,\text{lb}\cdot\text{in.}$$

Rearranging Equations 4-1 and 4-3 for section modulus and using the yield stress for A 36 steel gives:

$$bd^2 = \frac{6 \times 3.720\,\text{N}\cdot\text{m}}{248{,}000{,}000\,\text{N/m}^2} = 9.00 \times 10^{-5}\,\text{m}^3$$

$$bd^2 = \frac{6 \times 33{,}000\,\text{lb}\cdot\text{in.}}{36{,}000\,\text{lb/in.}^2} = 5.5\,\text{in.}^3$$

Let $b = 63\,\text{mm} = 0.063\,\text{m}$.

$$d = \sqrt{\frac{9.0 \times 10^{-5}}{0.063}} = 0.038\,\text{m}$$

Use 63 mm × 38 mm plate.

Let $b = 2.5\,\text{in}$.

$$d = \sqrt{\frac{5.5}{2.5}} = 1.48\,\text{in}.$$

Use 2 ½ in. × 1 ½ in. plate.

Pipe Wall Thickness

The hoop (circumferential) tensile stress in metal pipe due to the working pressure should not exceed 50% of the yield strength. The working pressure plus surge pressure due to water hammer should not exceed 75% of the yield strength or the mill test pressure. Depending on the grade of steel used, the yield strength can lie between 248,000 and 414,000 kPa (36,000 and 60,000 lb/in.). The yield strength of DIP is more uniform—a minimum of 290,000 kPa (42,000 lb/in.^{2}).

The hoop tensile stress is given by the equation

$$s = \frac{pd}{2t} \tag{4-4}$$

where s is the allowable circumferential stress in kilopascals (pounds per square inch), p is the pressure in kilopascals (pounds per square inch), d is the outside diameter of the iron or steel cylinder in millimeters (inches), and t is the thickness of the iron or steel cylinder in millimeters (inches). (Theoretically, D should be the inside diameter, but the outside diameter is conservatively specified in most codes, partly because the ID is not known initially.) The longitudinal stress in a straight pipe is half of the circumferential stress.

Hangers and Supports

Supports or hangers must carry the weight of the pipe and fluid in exposed piping systems. The location of supports and hangers depends on the pipe size, joint systems, piping configurations, location of valves and fittings, weight of pipe and liquid, beam strength of the pipe, and the structure available to support the weight and all other static and dynamic forces, including expansion and contraction.

Supports or hangers must also carry the lateral forces due to earthquake. Either design vertical supports to resist the horizontal force of pipe and fluid or augment vertical supports and hangers with horizontal ones. Note that tension in wall anchors must resist the full horizontal force. Typical pipe supports are shown in Figure 4-16, and a few of the many hangers manufactured are shown in Figure 4-17.

Some general design precautions for pipe support system design include the following:

- Because pump casing connections to piping systems are rarely designed to support any load transmitted through the connection, supports should be provided on both the suction and discharge side of pumps to prevent pipe loads being transmitted to the pump casing.
- Flexible pipe couplings are recommended at pump inlets and outlets. They are useful because (1) they are "forgiving" and allow the contractor to level the pump and make minor horizontal adjustments; (2) properly selected, they isolate the pump from slight movements of the pipe; and (3) they reduce vibration and, to a limited extent, noise.
- If expansion and contraction of the piping could occur, the pipe supports should allow movement.
- Flexible joints, such as mechanical joints and couplings, must be supported on both sides because they are not designed to transmit loads. Flexible couplings must be constrained with lugs and lug bolts to prevent longitudinal movement caused by internal pipeline pressure.

ASME B31.1 describes hanger and support spacings for steel pipe with a minimum wall thickness of standard weight. A maximum bending stress of 16,000 kPa (2300 lb/in.^{2}) and a maximum deflection (sag) of 2.5 mm (0.1 in.) for pipes filled with water is assumed in the ASME spacings. The spacings given in Table 4-15 should not be exceeded.

Hanger rod sizes should be

- at least 9.5 mm (⅜ in.) for pipe 50 mm (2 in.) and smaller;
- at least 13 mm (½ in.) for pipe 63 mm (2 ½ in.) and larger.

Design hanger rods for a maximum working stress of 20 to 40% of the yield stress at the root area of the threaded ends (see Table 4-14). If the hangers are

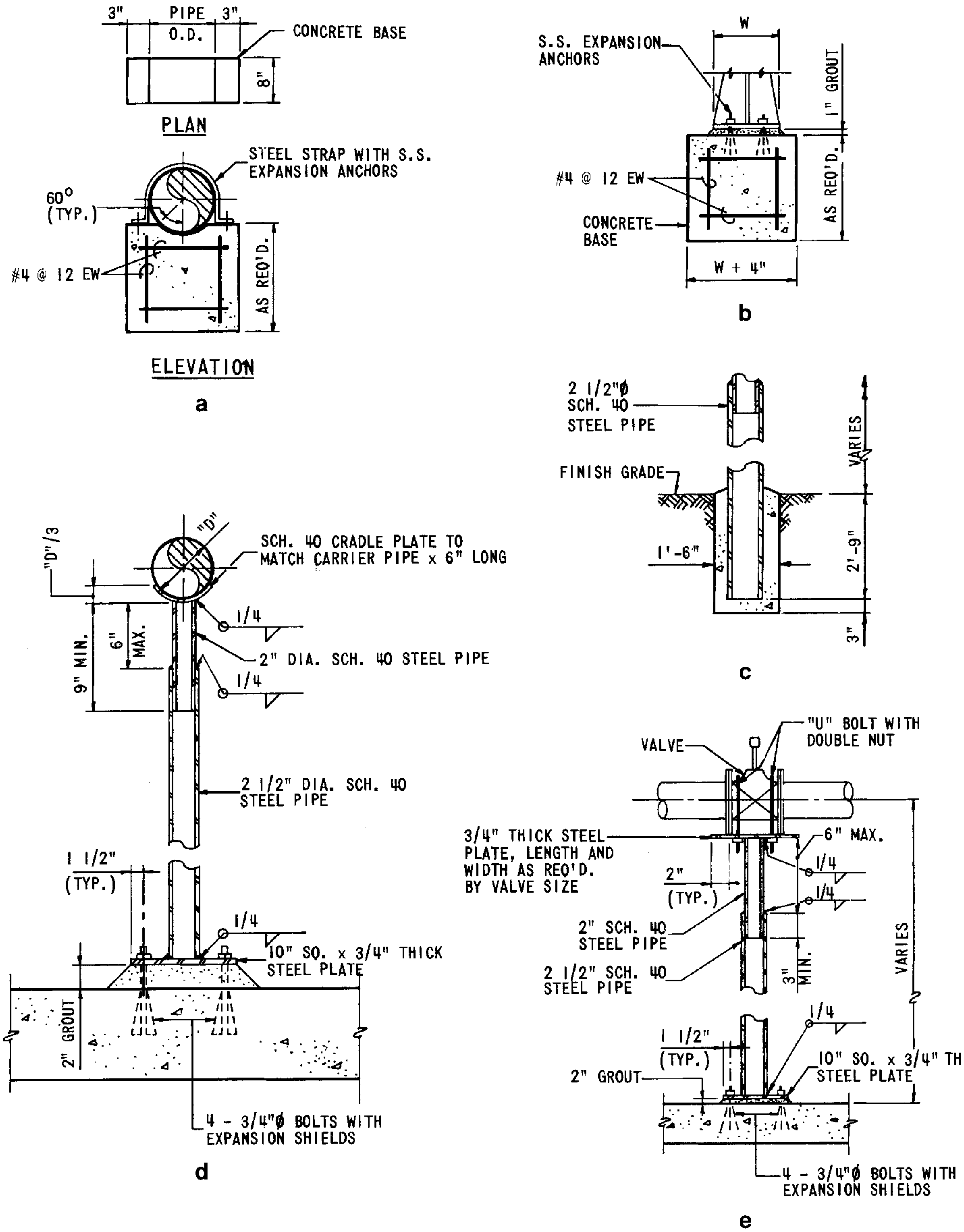


Figure 4-16. Pipe supports. (a) Concrete pipe support; (b) concrete base elbow support; (c) steel pipe support base in soil; (d) steel pipe support on floor; (e) steel valve support on floor. For floor supports, provide similar horizontal supports to resist seismic forces. Courtesy of Wilson & Company, Engineers & Architects.

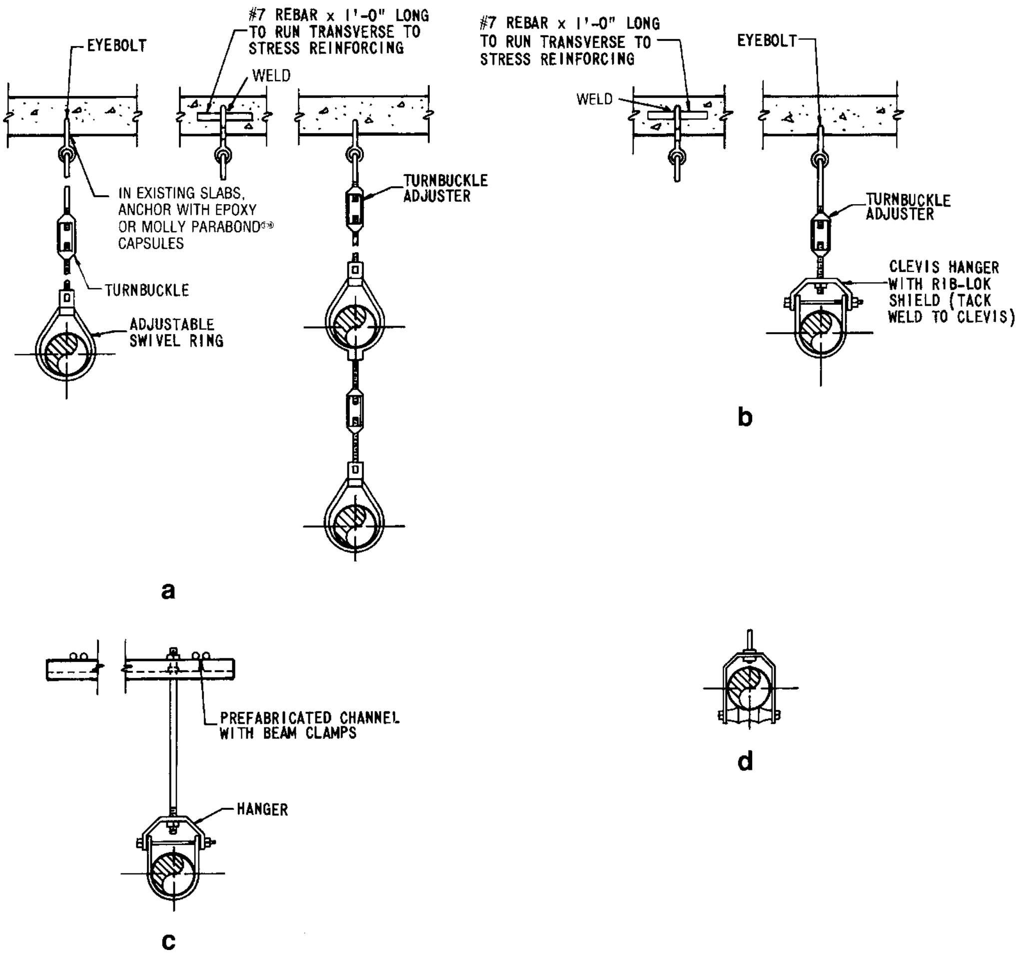


Figure 4-17. Pipe hanger details. (a) Swivel rings (for pipe that requires no protecting shield); (b) clevis hanger [for flexible pipe, add a shield (e.g., a similar pipe split in half) between the pipe and hanger]; (c) clevis hanger (use for hanging from steel joists); (d) adjustable swivel roller (use for pipe that expands and contracts). Courtesy of Wilson & Company, Engineers & Architects.

embedded in or supported by concrete, make sure the embedment is as strong as the threaded rod or limit the allowable load to the specifications in UBC Table 19-D. Capsule anchors containing a two-part resin (such as Emhart Molly Parabond™) are superior to expansion anchors, which use friction to resist being pulled out. Some engineers do not trust glued anchors, however, because good and dependable workmanship is critical. Embedded eye bolts should be welded closed in addition to being welded to the reinforcing bars, as shown in Figure 4-17a and b.

Hanger rod sizing for plastic pipe should be the same as for steel pipe, but the spacing of hangers should be as recommended by the plastic pipe manufacturer. Spacings are typically half those for steel and ductile iron pipe.

In addition to installing supports or hangers for straight runs of pipe, provide additional hangers or supports at

- concentrated loads, such as valves;
- fittings;
- both sides of nonrigid joints; and
- pipe connected to suction and discharge casings of pumps.

A wide variety of standard hangers and supports is available [17]. Combining Equation 4-1 for stress due to bending ($s = Mc/I$) and bending moment for simple spans ($M = wL^2/8$) and rearranging, the equation for calculating pipe support spacings based on stress (for straight pipe runs) becomes

Table 4-15. Support Spacing for Iron or Steel Pipe per ANSI B31.1

Nominal pipe size		Span			
		Water service		Steam, gas, or air service	
mm	in.	m	ft	m	ft
25	1	2.1	7	2.7	9
50	2	3.0	10	4.0	13
75	3	3.7	12	4.6	15
100	4	4.3	14	5.2	17
150	6	5.2	17	6.4	21
200	8	5.8	19	7.3	24
300	12	7.0	23	9.1	30
400	16	8.2	27	10.7	35
500	20	9.1	30	11.9	39
600	24	9.8	32	12.8	42

$$\text{Spacing} = L = \left(\frac{8sI}{wc}\right)^{0.5} \tag{4-5}$$

where L is the spacing in meters (inches), s is the allowable stress in Newtons per square meter (pounds per square inch), I is the moment of inertia from Equation 4-2, w is the weight of pipe and water per unit length in Newtons per meter (pounds per inch), and c is the distance from the neutral axis to the outer fiber (*d/2*) in meters (inches).

The equation of sag for a uniformly loaded simple span is

$$\delta = \frac{5wL^4}{384EI} \tag{4-6}$$

and rearranging to find L gives

$$L = \left(\frac{384EI\delta}{5w}\right)^{0.25} \tag{4-7}$$

where E, the modulus of elasticity, is given in Table A-10. Be careful to use compatible units.

Buried Piping

Buried pipes must support external structural loads, including the weight of the soil above the pipe plus any superimposed wheel loads due to vehicles if the pipeline crosses a runway, railway, or roadway.

External Loads

The two broad categories for external structural design are rigid and flexible pipe. Rigid pipe supports external loads because of the strength of the pipe itself. Flexible pipe distributes the external loads to surrounding soil and/or pipeline bedding material. Consider DIP, steel, and PVC to be flexible, whereas AC and RCPP are rigid. (FRP is also rigid, but refer to Section 4-5 for a warning about buried service.)

Supporting strengths for flexible conduits are generally given as loads required to produce a deflection expressed as a percentage of the diameter. Ductile iron pipe may be designed for deflections up to 3% of the pipe diameter according to AWWA C150/ANSI A21.50. Until recently, plastic pipe manufacturers generally agreed that deflections up to 5% of the diameter were acceptable. Some manufacturers now suggest that deflections up to 7% of the diameter are permissible. Many engineers, however, believe these values are much too liberal and use 2 to 3% for design.

Pipeline bedding conditions affect the safe supporting strength of both rigid and flexible conduits. Screenings, silt, or other fine materials are unsuitable for stable pipeline bedding and should be avoided particularly where the groundwater level may rise above the trench bottom. Better systems range from (1) merely shaping the trench bottom to (2) using select bedding material to (3) supporting the pipe on a monolithically poured concrete cradle. A stable, granular bedding material can be achieved with a

well-graded crushed stone with a maximum particle size of $^3/_4$ in. and containing not less than 95% by weight of material retained on the No. 8 sieve.

The design of pipe to resist external loads is involved because it depends on the stiffness of the pipe, the width and depth of trench, the kind of bedding, the kind of soil, and the size of pipe. Discussions are given in AWWA M11 [9], AWWA M9 [13], the DIPRA handbook [11], by Spangler [14, 15], and in many other publications.

Thrust Blocks

Where changes in flow direction occur, the exposed pipes must be restrained against the resultant thrust. Changes in flow direction occur at all bends, tees, plugs, caps, and crosses. Joint systems, such as flanges, welds, and grooved couplings, are designed to provide restraint up to the manufacturer's rating. But if the joint system is inadequate to contain the calculated thrust, external restraint is needed. Thrust calculations are illustrated in Example 3-5.

Buried pipes with mechanical or push-on joints require thrust blocking at deflections, bends, tees, plugs, or other changes in flow direction. Thrust blocks are constructed of cast-in-place concrete poured in the trench during pipeline installation. They are designed to act as horizontal spread footings that distribute the resultant force to the trench wall. The required area of the thrust block can be determined from the resultant force acting on the bend (see Example 3-5) and the allowable soil bearing pressure. For the resultant force, use whichever of the following is greater: (1) the total (working plus water hammer) pressure or (2) the pipeline test pressure. The allowable horizontal bearing pressure can best be found by calculating the Rankine passive pressure from the principles of soil mechanics, or it can be found in tables [9] (see also DIPRA [11] or UBC Table 29-B).

The concrete in thrust blocks should have a minimum compressive strength of 13,800 kPa (2000 lb/in.^{2}) at 28 days. The bearing area should be poured directly against undisturbed earth. Figures 4-18 and 4-19 show typical thrust block details.

The use of thrust blocks should be considered with great care. They are only as good as the stability of the soils used for reaction backing. In locations where the soils may be disturbed by future excavations (yard piping, treatment plant sites, busy streets, etc.), reliance on thrust blocks (particularly for large diameters and high-pressure pipeline systems) is not a very good idea. Instead, the use of restrained (lugged or harnessed) joints and trench friction is a better approach.

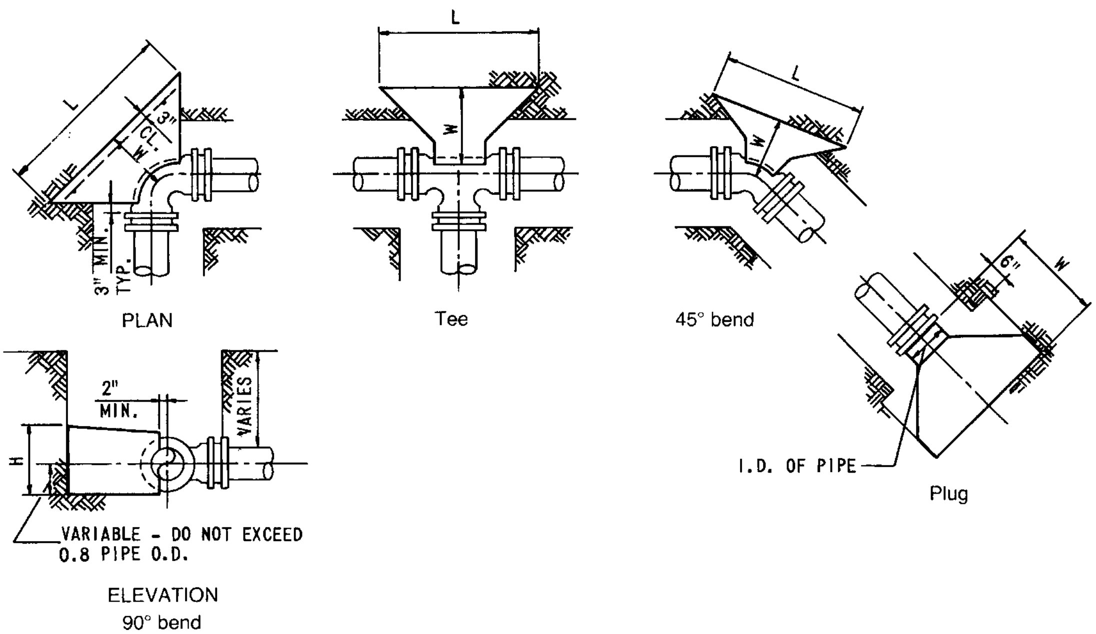


Figure 4-18. Typical underground thrust block details. Courtesy of Wilson & Company, Engineers & Architects.

Example 4-2
Hanger Rod Sizing and Spacing

Problem: A 600-mm (24-in.) steel pipe of standard weight is filled with water. The allowable tensile stress (including the factor of safety) in the hanger rods is 62,000 kPa or $62 \times 10^6\ \mathrm{N/m^2}(9000\ \mathrm{lb/in.^2})$. Assume that a factor of safety of 1.5 is to be applied to the ASME maximum bending stress in the pipe, so that stress is limited to $1.07 \times 10^7\ \mathrm{N/m^2}(1500\ \mathrm{lb/in.^2})$.
Solution: Calculate *I* from Equation 4–4 (see Table B-3 or B-4 for dimensions and weight of the pipe barrel).

SI Units

$$I = \frac{\pi}{64}[(0.610)^4 - (0.591)^4]$$
$$= 8.08 \times 10^{-4}\ \mathrm{m}^4$$

U.S. Customary Units

$$I = \frac{\pi}{64}[(24)^4 - (23.25)^4]$$
$$= 2070\ \mathrm{in.}^4$$

Calculate w, the total mass of pipe and water in kilograms per meter (pounds per foot) (see Tables B-3 and B-4 for the mass of the pipe barrel and for the inside cross-sectional area).

$$w = 141\ \mathrm{kg/m} + 0.273\ \mathrm{m}^2 \times 1000\ \mathrm{kg/m^3}$$
$$= 414\ \mathrm{kg/m}$$

$$w = 94.6\ \mathrm{lb/ft} + 2.94\ \mathrm{ft}^2 \times 62.4\ \mathrm{lb/ft^3}$$
$$= 278\ \mathrm{lb/ft} = 23.2\ \mathrm{lb/in.}$$

Transforming mass to force,

$$w = 414\ \mathrm{kg/m}\frac{9.81\ \mathrm{N}}{\mathrm{kg}} = 4060\ \mathrm{N/m}$$

From Equation 4-3 the maximum spacing is

$$L = \sqrt{\frac{8 \times 1.07 \times 10^7\ \mathrm{N/m^2} \times 8.08 \times 10^{-4}\ \mathrm{m^4}}{4060\ \mathrm{N/m} \times 0.305\ \mathrm{m}}}$$
$$= 7.47\ \mathrm{m}$$

$$L = \sqrt{\frac{8 \times 1500\ \mathrm{lb/in.^2} \times 2070\ \mathrm{in.^4}}{23.2\ \mathrm{lb/in.} \times 12\ \mathrm{in.}}}$$
$$= 299\ \mathrm{in.} = 24.9\ \mathrm{ft}$$

From Equation 4–6, the span for the allowable sag of 2.5 mm (0.10 in.) is

$$L = \left(\frac{384 \times 2.07 \times 10^{11}\ \mathrm{N/m^2}}{5} \times \frac{8.08 \times 10^{-4}\ \mathrm{m^4} \times 0.0025\ \mathrm{m}}{4060\ \mathrm{N/m}}\right)^{0.25}$$
$$= 9.35\ \mathrm{m}$$

$$L = \left(\frac{384 \times 2.9 \times 10^7\ \mathrm{lb/in.^2}}{5} \times \frac{2070\ \mathrm{in.^4} \times 0.2\ \mathrm{in.}}{23.3\ \mathrm{lb/in.}}\right)^{0.25}$$
$$= 379\ \mathrm{in.} = 31.5\ \mathrm{ft}$$

So stress, not deflection, governs. Although the allowable span from Table 4–15 is 9.8 m (32 ft) based on an allowable stress of 15,900 kPa ($2300\ \mathrm{lb/in.^2}$), use a lesser spacing, about 7.3 m (24 ft), for an allowable stress of 10,400 kPa ($1500\ \mathrm{lb/in.^2}$) in this problem. The force per hanger is

$$F = 4060\ \mathrm{N/m} \times 7.3\ \mathrm{m}$$
$$= 30{,}000\ \mathrm{N}$$

$$F = 23.2\ \mathrm{lb/in.} \times 12\ \mathrm{in./ft} \times 24\ \mathrm{ft}$$
$$= 6700\ \mathrm{lb}$$

and the required area at the root of threads is

$$A = \frac{30{,}000\,\text{N}}{62 \times 10^6\,\text{N/m}^2} = 4.8 \times 10^{-4}\,\text{m}^2 = 480\,\text{mm}^2 \qquad A = \frac{6700\,\text{lb}}{9000\,\text{lb/in.}^2} = 0.74\,\text{in.}^2$$

From Table 4–14, use rods 31 mm (1 ¼ in.) in diameter. However, if the rods are to be anchored only by embedment in concrete, note that UBC Section 2624 allows a maximum load of only 14,000 N (3200 lb) per bolt. That requirement would permit a spacing of only

$$L = \frac{14{,}000\,\text{N}}{4060\,\text{N/m}} = 3.45\,\text{m} \qquad L = \frac{3200\,\text{lb}}{278\,\text{lb/ft}} = 11.5\,\text{ft}$$

Cleanouts

Cleanouts should be installed in sludge and slurry lines that carry grease, grit, or other substances (such as lime) that may form deposits in the pipe (typical designs are shown in Figures 4-20 and 4-21). Lime-carrying waters are especially troublesome, so use troughs instead of pipe where possible.

Special cleanouts are required for flowmeters, and a suitable system is shown in Figure 20-12.

Pig Launching and Recovery

Typical pig launching and recovery stations are shown in Figures 4-22 and 4-23. The pig stop, Figure 4-24, is placed in the pig retrieval pipe prior to the pigging operation. The 2 ½ D barrel length is standard, but if space is tight, 1 ½ D can be used (although not with deZurik pigs). Pressure gauges are unnecessary. The best way to determine whether the pig has gone is to isolate the launch barrel and insert a rod through the

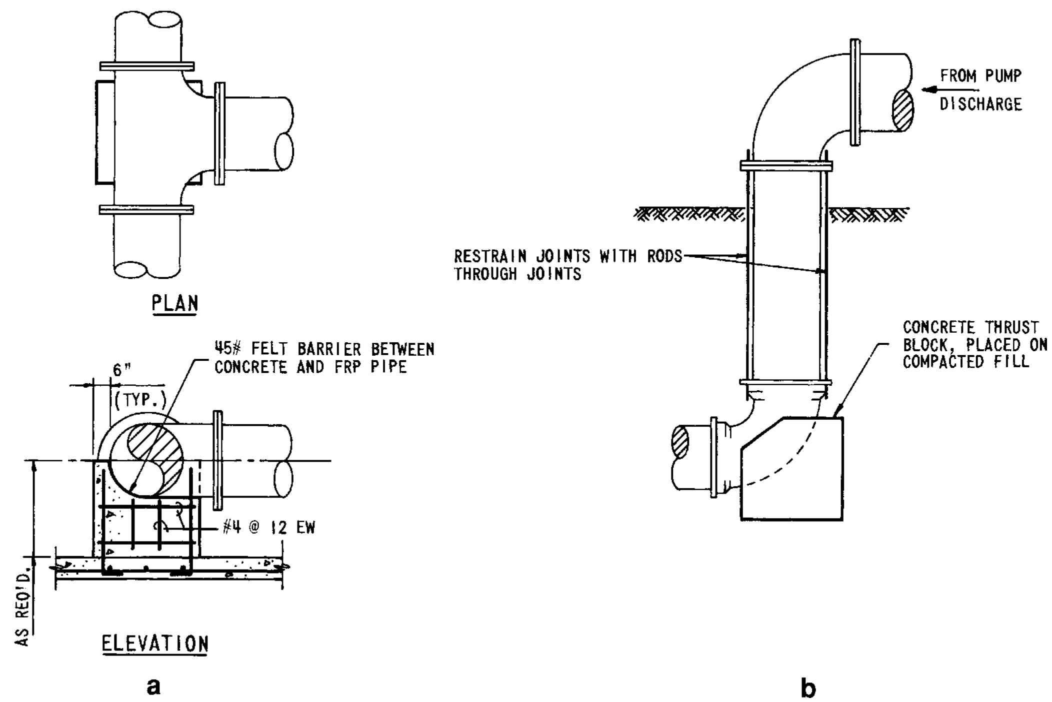


Figure 4-19. Typical thrust blocks for exposed pipes. (a) Combination thrust block pipe support; (b) thrust block/pipe support for emerging pipe. Courtesy of Wilson & Company, Engineers & Architects.

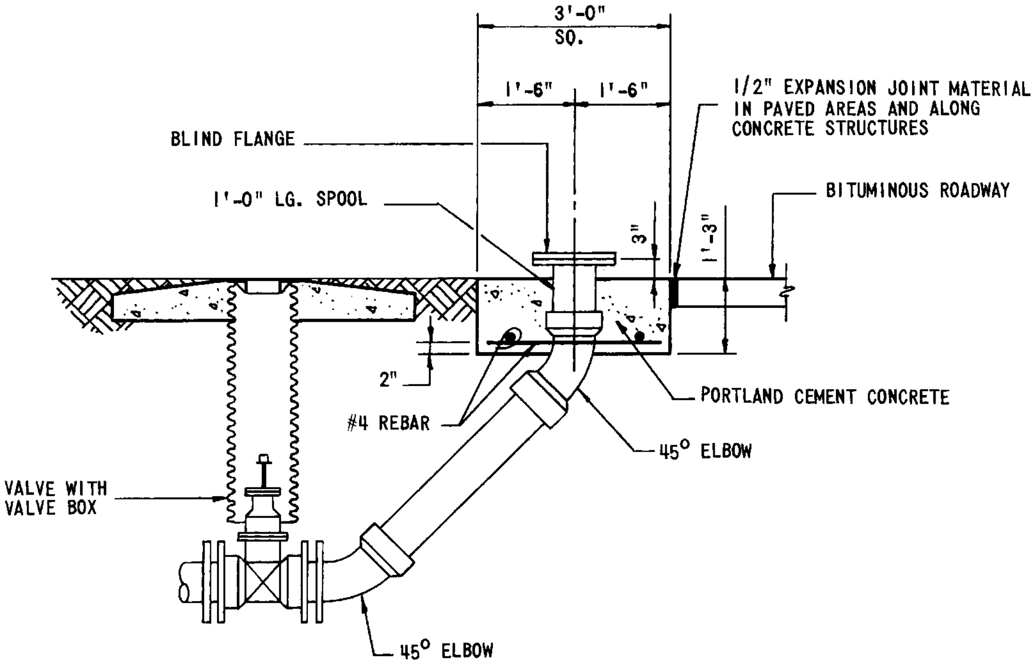


Figure 4-20. Underground pipe cleanout. Courtesy of Wilson & Company, Engineers & Architects.

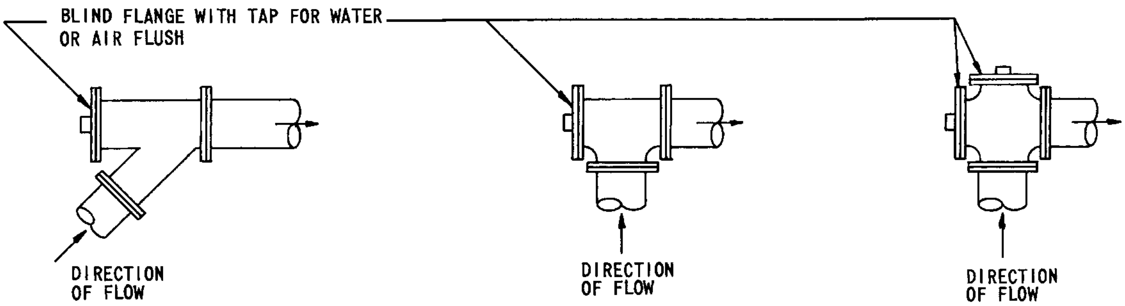


Figure 4-21. In-line cleanouts for exposed pipe. Courtesy of Wilson & Company, Engineers & Architects.

launching valve quick-connect. At the retrieval end, listen for the sound of the pig arriving.

4-9. Special Piping and Plumbing

The requirements for conveying small flows of wash-down water, fuel, cooling water, and sump drainage are entirely different from the foregoing. Pipes for fuel, wash water, seal water, and air are small and comparatively inexpensive even when constructed of materials such as copper or stainless steel. Special piping materials are listed in Table 4-16.

Water

If water is needed for lavatories or wash-down, galvanized steel with threaded connections is practical because the pipe would rarely exceed 38 mm (1 ½ in.) in diameter. Pipe fittings are usually made of malleable iron. Thread compound or Teflon™ tape should be applied to threads. There should be enough strategically placed unions to allow for dismantling and replacement. Threaded pipe must have a wall thickness no less than standard weight per ASME B36.10.

Copper tubing, with joints soldered with tin-antimony (ASTM B 32, Grade Sb5), is also practical.

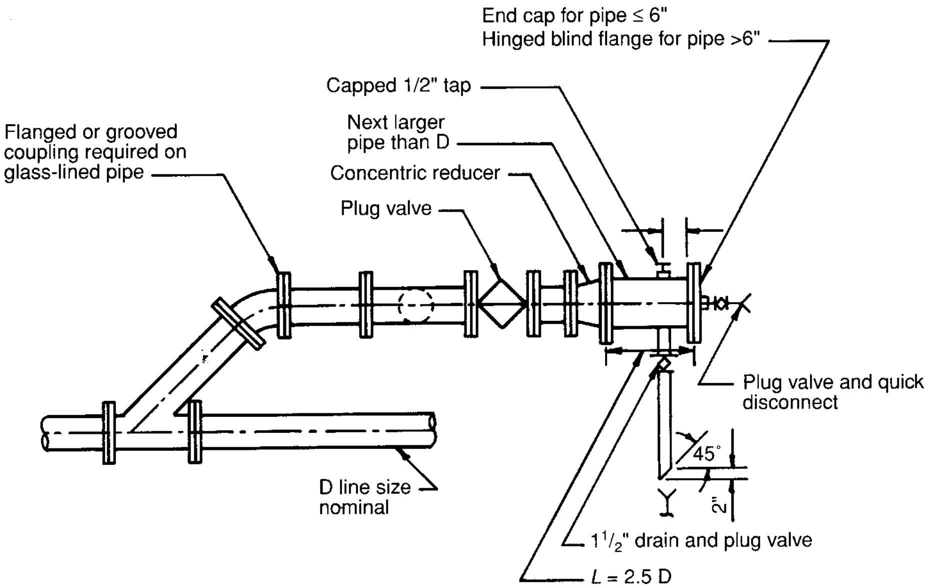


Figure 4-22. Typical pig launching station. Courtesy of Brown and Caldwell.

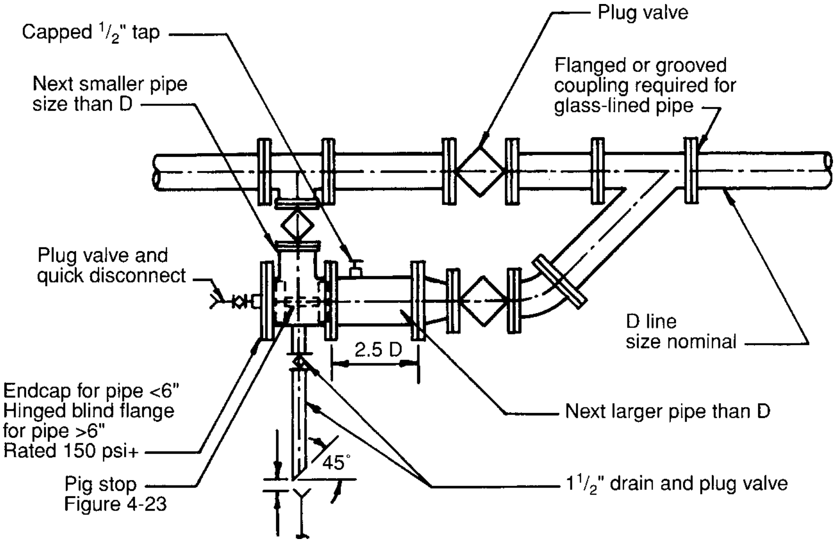


Figure 4-23. Typical pig retrieval station. Courtesy of Brown and Caldwell.

Do not use lead-based solder in potable water service. Copper is corroded by hydrogen sulfide, so copper or brass piping and tubing should be used with caution in areas of wastewater pumping stations exposed to a hydrogen sulfide atmosphere. Copper tubing also should not be allowed to come in contact with prefabricated wood trusses that have been treated with ammonium sulfate fire-retardant material, because the ammonium sulfate is very corrosive to copper. Require that the copper tubing be attached to wood trusses by pipe hangers, and not just attached directly to or laid directly on the wood.

PVC could be used in Schedule 40 for solvent-welded joints or Schedule 80 for threaded joints, but it requires closely spaced supports. Since galvanized steel pipe within corrosive atmospheres, such as wet wells, corrodes quickly, a more satisfactory material is stainless steel or PVC.

Stainless-steel pipe in most pumping station services is covered by ASTM A 312 for 3 through

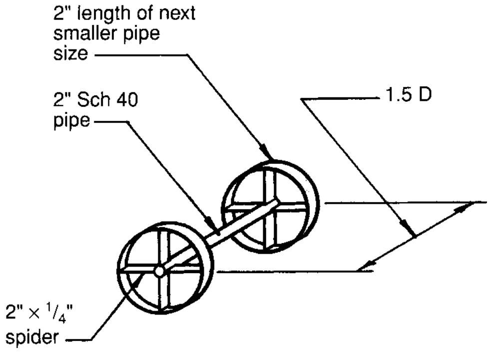


Figure 4-24. Typical pig stop. Courtesy of Brown and Caldwell.

750 mm ($\frac{1}{8}$ through 30 in.) and A 778 for 75 through 1200 mm (3 through 48 in.) for material; for dimensions, it is covered by ASME B36.19 (not ASME B36.10). Note that the minimum wall thickness for threaded pipe is Schedule 40S. Fittings 50 or 75 mm (2 or 3 in.) and smaller should be threaded and should conform with ASTM A 403 and ASME B 16.3 (see the previous discussion on the applicability of threaded joints). Fittings larger than 75 mm (3 in.) should be butt welded, grooved end, or flanged and should conform to ASTM A 403 or A 774, and ASME B16.9.

The wall thicknesses for the pipe schedules for stainless-steel pipe are as follows:

- Schedule 40S for stainless-steel pipe is the same as Schedule 40 for carbon steel pipe in sizes of 10 in. and smaller.
- Schedule 40S for stainless-steel pipe is the same as standard weight ($\frac{3}{8}$ -in. wall) for carbon steel pipe in sizes of 12 in. and larger.
- Schedule 80S for stainless-steel pipe is the same as Schedule 80 for carbon steel pipe in sizes of 8 in. and smaller.

Note that the letter S always follows the Schedule number for stainless steel, for example, 40S.

Threaded Joints

Threaded malleable iron fittings and couplings (ASME B16.3) can be used with steel pipe and are usually cheaper than forged steel fittings conforming to ASME B16.11. Threading conforms to ASME B1.20.1 for both malleable iron and forged steel fittings.

Forged steel fittings (90-degree and 45-degree elbows, tees, crosses, couplings, caps, plugs, and bushings) conforming to ASME B16.11 are available in sizes of 3 through 100 mm ($\frac{1}{8}$ in. through 4 in.) Assembling and disassembling threaded fittings and joints larger than 50 to 75 mm (2 to 3 in.) is labor-intensive, so only small pipes are connected by threaded fittings.

Unions at strategic locations are needed for disassembly. Steel unions conform to MSS SP-83; malleable iron unions conform to ASME B16.39.

Van Stone Flanges

Van Stone flanges are especially economical for stainless-steel piping. The flange can be carbon steel or ductile iron because it is not in contact with the liquid and, hence, does not have to be corrosion-resistant unless the concern is for external corrosion. The joint is made by roll flaring the end of the pipe per ASTM F 2015. A slip-on flange can then be placed over the pipe to make the connection. Such flanges are especially attractive with stainless-steel pipe, but where flanged joints would be exposed to corrosive conditions (as in a wastewater pumping station wet well), specify stainless-steel flanges.

Diesel Fuel Service

Both the Uniform Fire Code and the Standard Fire Prevention Code require that steel piping for flammable and combustible liquids have a wall thickness determined in accordance with ASME B31.3 and B31.4 with a minimum wall thickness of standard weight. ASME B31.1, B31.3, and B31.4 also contain design guidelines for flammable fluid piping, which are summarized as follows:

- *Prohibited pipe:* (1) furnace butt-welded steel, (2) cast iron, (3) copper, (4) brass, (5) aluminum, and (6) thermoplastic (above ground).

Table 4-16. Piping Materials and Standards

Abbreviation	Piping material	Standard
ABDI	Abrasion-resistant ductile iron	None
ABS	Acrylonitrile-butadiene-styrene	ASTM D 1527, D 2661, D 2751, D 2680, F 409, F 545
ACP	Asbestos-cement pipe	AWWA C400, C402; ASTM C 296
ALUM	Aluminum pipe	ASTM B 241, B 361
ARCI	Acid-resistant cast iron	ASTM A 518
BR	Brass	ASTM B 43
CASST	Carpenter 20 stainless steel	ASTM B 464, B 474
CCP	Concrete cylinder pipe (bar-wrapped)	AWWA C303
CISP	Cast-iron soil pipe	CISPI 301; ASTM A 74, A 888
CML	Cement-mortar-lined steel pipe	AWWA C200, C205, C208; ASTM A 53, A 134, A 135, A 139, A 234, A283, A 570, A 572
CMP	Corrugated metal pipe	AASHTO M36
CPE	Corrugated polyethylene	ASTM F 405, F 667; AASHTO M252
CPVC	Chlorinated polyvinyl chloride	ASTM D 1784, F 437, F 439, F 441, F 493
CTEL	Coal-tar-enamel-lined steel	AWWA C200, C203; ASTM A 53, A 134, A 135, A 139, A 234
CTXL&C	Coal-tar epoxy-lined and -coated steel pipe	AWWA C200, C210; ASTM A 53, A 134, A 135, A 139, A 234
CU	Copper	ASTM B42, B 75, B 88
DIP	Ductile iron pipe	AWWA C150, C151, C110, C111, C115; ASTM A 395, A 436
DWV	Drain, waste, and vent copper tubing	ASTM B 306
FRP	Fiberglass-reinforced plastic pipe	ASTM D 2310, D 2992, D 2996, D 2997; AWWA C950
FXL&C	Fusion-bonded epoxy-lined and-coated steel pipe	AWWA C213; ASTM A 53, A 134, A 135, A 139, A 324
GALVS	Galvanized steel pipe	ASTM A 53
HDPE	High-density polyethylene	AWWA C906
PCCP	Prestressed concrete-cylinder pipe	AWWA C301
PP	Polypropylene	ASTM D 2146, D 4101
PPLS	Polypropylene-lined steel pipe	ASTM F 1545
PTFELS	Polytetrafluoroethylene-(Teflon™)-lined steel pipe	ASTM F 1545
PVC	Polyvinyl chloride (pressure) pipe	ASTM D 1784, D 1785, D 2464, D 2466, D 2467, D 2564; AWWA C900, C905
PVC (G)	Polyvinyl chloride (gravity) pipe	ASTM D 2688, D 3034
PVDF	Polyvinylidene fluoride (KYNAR™) pipe	None
PVDFLS	PVDF-lined steel pipe	ASTM F 1545
RCP	Reinforced concrete pipe (gravity)	ASTM C 76, C 655
RCPP	Reinforced concrete pressure pipe	AWWA C300, C302; ASTM C 361
RPMP	Reinforced plastic mortar pipe	ASTM D 3517, D 3262, AWWA C950
SP	Steel pipe	ASTM A 53, A 134, A 135, A 139, A 324, A 283, A 570, A 572, A 795; AWWA C200
SST	Stainless-steel pipe	ASTM A 240, A 312, A 403, A 774, A 778
VCP	Vitrified clay pipe	ASTM C 700
WSP	Welded steel pipe	AWWA C200, C208
YOL	Yolloy	ASTM A 714

- *Prohibited connections:* (1) cast, malleable, or wrought iron threaded couplings, (2) cast-iron flanges, unless integral with cast-iron valves, pressure vessels, and other equipment.
- *Required:* (1) double welding for slip-on flanges and (2) flat facing connecting steel flanges to cast-iron flanges.
- *Recommended materials:* (1) seamless steel such as ASTM A 53 Type S, (2) welded steel with straight seam conforming to ASTM A 53 Type E, ASTM A 134, ASTM A 155, or (3) API 5L electric resistance or double-submerged arc welded with allowable stresses no higher than for ASTM A 53 Type S with an

appropriate longitudinal joint efficiency factor included.

- *Recommended:* (1) welded joints between steel components where practicable; (2) where bolted flanged joints are necessary, gaskets suitable for the service; (3) where threaded joints are necessary, at least Schedule 80 (extra strong) pipe with extreme care in assembly to ensure leaktightness; (4) steel or ductile iron for valves, fittings, and other piping components for systems within plants or buildings that contain equipment with open flame or parts that operate at temperatures over 260°C (500°F); and (5) black steel—never galvanized steel, because the zinc contaminates the fuel (see ASTM D 975, Appendix X2.7) and the galvanizing sometimes flakes off and can clog fuel metering orifices.

The UFC requires that underground piping incorporate flexible joints where the piping leaves the dispensing location and just before connecting to the tank fittings. Swivel joints incorporate ball bearings and permit rotation and movement from one to many degrees of freedom.

Sewers

Cast-iron soil pipe (CISP) 50 through 375 mm (2 through 15 in.) and larger is suitable for use as drainpipe inside buildings if it is installed above the floor with no-hub ends and with neoprene sealing sleeves and Type 301 or 303 stainless-steel clamps (in accordance with CISPI 301). For buried service or service under slabs and buildings, use hub-and-spigot ends (see also ASTM A 74 standards). In lieu of CISP, ABS pipe is frequently used; for drainpipe smaller than 50 mm (2 in.) use steel, copper, or PVC.

Dry Chlorine Gas

As recommended in ASME B31 and by the Chlorine Institute [18], make the piping arrangements as simple as possible. Keep the flanged or screwed joints to a minimum. Slope the pipe to allow drainage, avoid low spots, allow for expansion due to temperature changes, and be sure that the pipe is well supported.

Details of materials and construction can be obtained from Pamphlet No. 6 of the Chlorine Institute [18]. Other publications may also be applicable to pumping stations [19–26]. After assembly and pressure testing, chlorine gas piping must be thoroughly cleaned and *all* moisture must be removed.

Chlorine Solutions

Chlorine solutions can be carried in PVC piping. All fittings should be plastic, glass, or Hastelloy™. Most metals, including 316 stainless steel, corrode rapidly in concentrated chlorine solutions.

In recent years, sodium hypochlorite has become more and more popular in lieu of gaseous chlorine and the associated chlorine solutions. Sodium hypochlorite, however, is corrosive to the "normal" solvent cements used in PVC and CPVC piping. There is some evidence that the sodium hypochlorite attacks the silica fillers in the conventional solvent cements. Thus, silica-free solvent cement or a solvent cement specifically recommended by the manufacturer for sodium hypochlorite service should be specified in this application.

Air

Use stainless steel or copper. Do not use PVC because heat destroys plastic pipe. Do not use galvanized steel because moisture in the air corrodes it, and although dryers can be added at the compressor, maintenance cannot always be ensured.

Design of Plumbing Systems

Plumbing work in a pumping station usually includes roof drainage, toilet fixtures, floor drains, and sump pumps in addition to the necessary water, waste, and vent piping and a water heater.

Storm Drainage

The sizing of roof drains, horizontal conductors, and vertical leaders or downspouts (usually covered in the applicable plumbing code) is based on roof area and historical rainfall intensity. Storm drainage, including roof and area drains, catch basins, and foundation drains, should (1) connect to a storm drain, (2) discharge through a trap into a combined sewer, or (3) be drained to grade if neither (1) nor (2) is available. In addition to any sump receiving subgrade sanitary waste, a separate sump may be required by code for foundation drainage.

Roof drainage piping often conveys water at a temperature below the dew-point temperature of air in the building, so condensation tends to occur on the pipe. The horizontal conductors (at least) should be

insulated to prevent dripping from such condensation, especially if the pipes pass over or near electrical equipment.

The materials usually used for storm drainage piping include galvanized steel pipe, galvanized cast-iron drainage fittings, cast-iron soil pipe and fittings, and polyvinyl chloride plastic pipe and fittings. Both storm and sanitary drain lines buried below floors should be encased in concrete when the floor is poured to protect them from corrosion and settlement.

Sanitary Drainage

Sanitary soil, waste, and vent piping sizes and arrangements must conform to the applicable plumbing code. Equipment drains receiving pump seal water can be enlarged hub drains raised above the floor or regular floor-mounted drains fitted with a funnel strainer. Cleanouts should be located on both sanitary and storm drainage lines at 15-m (50-ft) intervals to permit easy cleaning.

Cross Connections

Protection of the potable water supply against contamination is vital. Reduced-pressure-principle backflow preventers, or an equivalent approved means of protection, must be installed in the branch pipe that supplies equipment connections, hose bibbs, or hose valves. Backflow preventers should be located above the flood level with a proper air gap in their drain connections. They must be tested at least annually to ensure safe operation, and the care of backflow preventers should be included in the O&M manual.

Some codes do not permit the installation of backflow preventers, but require potable water to be discharged through an air break into a tank, from which it may be pumped to potentially contaminating uses. Interior hose bibbs should be conveniently located for easy wash-down of the station, and exterior hose bibbs should be frostproof in cold climates.

Sumps

To prevent frequent pump starts, sumps for subgrade drainage should be sized to hold at least two to three times the flow rate capacity of the installed sump pump below the lowest inlet. Install duplex pumps and a sump high-level alarm if damage to other equipment could follow failure of a single pump. Each pump should discharge individually through a check valve and a gate valve before joining in a common discharge riser to the gravity flow waste line. If the piping system of a sump pump can be subjected to freezing temperatures, it should either (1) be designed to be self-draining or (2) be protected by heat tracing to prevent the formation of ice.

So that occasional solids can be passed, sump pumps should be of the semi-open impeller type without an inlet screen. Submersible pumps have the advantage of being inherently floodproof, provided their control panel is located above the flood level.

Float switches are recommended in the sump. They are arranged (1) to alternate the operation of duplex pumps, (2) to start the second pump if the first does not handle the load for any reason, and (3) to energize local and remote alarms if the design HWL is exceeded. The pumps should be capable of continuous operation. Specify a manual "test" position on pump selector switches to permit periodic manual tests of pump operation, and state the recommended frequency of testing in the O&M manual. The pumps and their wiring, controls, and alarms must be explosion-proof if open to a space classified as hazardous by Articles 500 to 502 of the NEC.

4-10. References

Anyone responsible for designing or selecting piping should have a minimum library of reference materials that includes design handbooks, standard codes and specifications, and one or more manufacturer's catalogs for each of the piping materials in the chapter. Manufacturers' catalogs often contain most of the needed design methods and useful excerpts from standard specifications.

Because some of these publications are revised occasionally, obtain the latest edition.

1. Crocker, S., M. L. Nayyar, and R. C. King, *Piping Handbook,* 6th ed., McGraw-Hill, New York (1992).
2. Deutsch, D. J., *Process Piping Systems,* Chemical Engineering, McGraw-Hill, New York (1980).
3. Langelier, W. F., "The analytical control of anti-corrosion water treatment," *Journal of the American Water Works Association,* 28, 1500–1521 (October 1936).
4. Merrill, D. T., and R. L. Sanks, "Corrosion control by deposition of $CaCO_3$ films. A practical approach for plant operators," *Journal of the American Water Works Association,* Part I, 69, 592–599 (November 1977); Part II, 69, 634-640 (December 1977); Part III, 70, 12–18 (January 1978).

5. Merrill, D. T., and R. L. Sanks, *Corrosion Control by Deposition of $CaCO_3$ Films. A Handbook of Practical Application and Instruction,* American Water Works Association, Denver, CO (1978).
6. Pisigan, R. A., Jr., and J. E. Singley, "Calculating the pH of calcium carbonate saturation," *Journal of the American Water Works Association,* 77, 83–91 (October 1985).
7. Pisigan, R. A., Jr., and J. E. Singley, "Effects of water quality parameters on the corrosion of galvanized steel," *Journal of the American Water Works Association,* 77, 76–82 (November 1985).
8. Pigg, B. J., "Asbestos cement pipe," *ASTM Standardization News,* pp. 56–57 (April 1988).
9. AWWA M11, *Steel Pipe—A Guide for Design and Installation,* 3rd ed., American Water Works Association, Denver, CO (1989).
10. Sanks, R. L., *Water Treatment Plant Design for the Practicing Engineer,* Ann Arbor Science Publishers, Ann Arbor, MI (1989).
11. DIPRA, *Handbook of Ductile Iron Pipe,* 6th ed., Ductile Iron Pipe Research Association, Birmingham, AL (1984).
12. *Handbook of PVC Pipe, Design and Construction,* Uni-Bell PVC Pipe Association, Dallas, TX (1991; updated frequently).
13. AWWA M9, *Concrete Pressure Pipe,* American Water Works Association, Denver, CO (1995).
14. Spangler, M. G., *Soil Engineering,* 2nd ed., International Textbook Company, Scranton, PA (1960).
15. Spangler, M. G., "Underground conduits—An appraisal of modern research" (with discussions), *Transactions of the American Society of Civil Engineers,* 113, 316–374 (1948).
16. *Piping Engineering,* Tube Turns Division, Louisville, KY (1974).
17. *MSS, Pipe Hangers and Supports, Selection and Application, MSS SP-58*, Manufacturers Standardization Society, Inc., of the Valve and Fittings Industry, Arlington, VA (2002).
18. "Piping systems for dry chlorine," 14th ed., Pamphlet 6, The Chlorine Institute, Inc., Washington, DC (December 1998).
19. *The Chlorine Manual,* 6th ed., The Chlorine Institute, Inc., Washington, DC (1997).
20. "Non-refrigerated liquid chlorine storage," 6th ed., Pamphlet 5, The Chlorine Institute, Inc., Washington, DC (December 1998).
21. "Chlorine vaporizing systems," 5th ed., Pamphlet 9, The Chlorine Institute, Inc., Washington, DC (April 1997).
22. "Chlorine pipelines," 3rd ed., Pamphlet 60, The Chlorine Institute, Inc., Washington, DC (July 2001).
23. "Estimating the area affected by chlorine releases," 2nd ed., Pamphlet 74, The Chlorine Institute, Inc., Washington, DC (February 1991).
24. "Refrigerated liquid chlorine storage," 2nd ed., Pamphlet 78, The Chlorine Institute, Inc., Washington, DC (June 1994).
25. White, G. C., *Handbook of Chlorination and Alternative Disinfectants,* 4th ed., John Wiley & Sons, New York (1998).
26. "Chlorine pipelines," 3rd ed., Pamphlet 60, The Chlorine Institute, Inc., Washington, DC (July 2001).

4-11. Supplementary Reading

1. Steele, A., *Engineering Plumbing Design,* Miramar, Los Angeles, CA (1977).
2. Lindsey, F. R., *Pipefitters Handbook,* 3rd ed., Industrial Press, New York (1967).
3. *Procedures for Pipewelding,* National Association of Plumbing, Heating, Cooling Contractors, Washington, DC (1983).
4. Singley, J. E., et al., *Corrosion Prevention and Control in Water Treatment and Supply Systems,* Noyes, Park Ridge, NJ (1985).

Chapter 5
Valves

CARL N. ANDERSON
BAYARD E. BOSSERMAN II
CHARLES D. MORRIS

CONTRIBUTORS
Casi Cadrecha
Joseph E. Lescovich
Harvey W. Taylor
John Vuncannon

Engineers typically lavish much attention on pumps but little on valves, which are just as important for the proper functioning of a pumping station. The discussion of valves and actuators in this chapter applies mainly to the control of the pumped fluid. Small valves for auxiliary purposes (e.g., seal water, fuel, and plumbing) are only briefly mentioned.

Most valves in a pumping station are for isolation service and, as such, are either open or closed. Actuators are usually manual for valves smaller than 600 mm (24 in.), and power-driven actuators are usually used for valves larger than 900 mm (36 in.). Check valves respond to flow direction and open and close automatically. Pump control valves serve a dual function as check valves, and the powered actuators are programmed to open and close slowly enough to control transient pipeline pressures within acceptable limits. If used at all, control valves are the most important valves in a pumping station. Flow-control valves (or valves that modulate to control flow or pressure) are used in small sizes for cooling-water or seal-water piping. Pressure-control valves are sometimes used in distribution systems to separate regions of two different pressures. In pumping stations, surge relief or surge anticipation valves are occasionally used to relieve high-pressure surges.

The associated design considerations of cavitation, noise, actuator sizing, and vibration are specific for the brand and model of the valve used and, hence, are not discussed here. Designers should be aware, however, of the problems and explore them thoroughly with the manufacturer.

Additional information of value can be found in the literature and in the following references: Cook [1], O'Keefe [2], Deutsch et al. [3], AWWA M11 [4], Hutchison [5], Lyons [6], and others [7–11]. Photographs and drawings that depict the various valves and show how they work are so readily obtainable from manufacturers that few are reproduced here.

References to a specification or standard are given in abbreviated form (such as ANSI B16.34) because such designations are sufficient for identification. The dangers in referencing a standard without carefully reading the entire work are discussed in Section 1-4.

5-1. Designing for Quality

Choosing the right kind, style, and even make of valve in the right situation is vital to the proper functioning of the station. A valve proper for one installation may be improper for another. Style (and

even model and maker) has a profound effect on satisfactory service. The problem of selection is complicated by the following considerations: (1) a valve satisfactory in one location may not be satisfactory in another location even if conditions are only slightly different; (2) makers of several styles of valves may make some good ones and some poor ones; (3) models are changed from time to time and a valve, once poor, may now be good; and, finally, (4) it is extremely difficult to write specifications to comply with the law, allow competitive bidding, and still obtain a satisfactory valve.

Familiarity with the various manufactured products, which is the key to good selection, can be achieved by

- Interviewing many manufacturers' representatives (but with critical skepticism), and
- Discussing valves with expert consultants and with users—the operators and the utility managers.

Note that many makes or models of valves look alike but differ significantly in quality. A valve is probably of high quality if a competitor agrees. The best valves are expensive, so a misplaced emphasis on low initial cost makes procurement of satisfactory valves difficult at best. Valves are the heart of the hydraulic system; if they fail, the system fails. In the long run, a cheap valve will have proved to be the most expensive. Skimping on valves is the wrong way to try to save money.

Good quality can be obtained by incorporating into the specifications such items or criteria as listed below.

- *Materials:* Abrasion-, corrosion-, and cavitation-resistant materials of construction—especially for seats (see Table 5-1).
- *Headloss:* Specify a price penalty based on life-cycle energy costs for more headloss than a stated value.

Table 5-1. Typical Valve Seat Materials

Type specification	Life	Remarks
Resilient seats		
Buna N	Good	General purpose elastomer for water and wastewater. Economical, suitable for most water and wastewater uses.
Leather	Good	Usually impregnated with various waxes to improve qualities. Sometimes used for water, not wastewater.
UHMW[a]	Very good	Very abrasion- and chemical-resistant; not expensive.
Teflon™	Poor	Impervious to chemical attack; creeps too much for normal use; expensive.
Viton	Poor	Use only for aggressive liquids or high temperatures; creeps somewhat.
Natural rubber	Good	Suitable only for fresh water.
Rigid seats		
Bronze		
ASTM B 62, B 584 (34 alloys), B 16, B 371	Tremendous variation among alloys	Most common seat material; least resistant to erosion or corrosion.
Stellite		
SAE, J775, AMS 5373, 5375, 5378, 5380, 5385, 5387, 5788	Excellent	Expensive; best of all for resistance to both corrosion and erosion.

Stainless steel	Specification	Erosion[b]	Corrosion[b]	Remarks
440C	ASTM A276, alloy S44004	1	5	Most resistant to erosion; highest hardness.
420	ASTM A276, alloy S42000	2	5	
17-4 PH	ASTM A564, alloy S17400	3	3	
410	ASTM A276, alloy S41000	4	4	
405	ASTM A276, alloy S40500	5	4	
316	ASTM A276, alloy S31600	6	1	Highest resistance to corrosion.
304	ASTM A276, alloy S30400	7	2	Least resistant to erosion.

[a]Ultra-high-molecular-weight polyethylene.

[b]1 signifies the best resistance.

- *Proof-of-design tests:* Require certification of successful completion of proof-of-design testing conducted on a 6-in. (or larger) valve in accordance with AWWA C504, Section 5.2, altered as necessary to apply to the valve specified (e.g., "disc" in the standard means "plug" in a specification for a plug valve).
- *Massiveness of construction and conservatively designed bearings, shafts, and other moving parts:* Shafts, especially in cushioned-swing check valves, should be very large; compare the various makes and models to specify a high-quality product.
- *Service records:* Find a way to specify features that eliminate valves with poor records.
- *Complexity:* Specify valves and actuators that are simple, trouble-free, and require minimum maintenance or the kind of maintenance within the capability of the workforce.
- *Resilient seat material that will not cold flow under differential pressures:* Look for well-designed mechanisms to retain seats in place (see Table 5-1); for wastewater and sludge, seat materials must be resistant to oils and solvents.
- *Responsibility:* Include a clause that involves the manufacturer in the responsibility for valve (and valve actuator system) performance. Warning: In some instances, manufacturers have inserted additional rubber shims or seals to the seats during some tests to meet the AWWA C504 testing requirement that the valve be drop-tight in both directions. Such practices should not be accepted.

Life-Cycle Cost

Quality might be said to be an inverse function of life-cycle cost, which is a combination of capital, maintenance, and energy costs. Headlosses can result in mind-boggling energy costs, as demonstrated in Example 5-1.

Location

Quality is a continuing concern and, hence, also a function of maintenance. Maintenance costs are elusive, but, whatever they are, they can be reduced by placing valves and actuators in locations that are easily accessible for servicing. Make it convenient to isolate and drain separate parts of the system, and write maintenance and exercising programs into the O&M manual. Note that because of savings in parts and operation and maintenance labor, it may be more cost effective to install an expensive valve (such as a plug or a valve of corrosion-resistant construction) that works when needed than to install a cheap valve (such as a gate or a valve of lesser quality) that may not work until repaired.

High quality is achieved as much by good location and good piping layout as by specifying high-quality hardware. A good valve is cheapened by misapplication or poor location. Some considerations for proper location are:

- In all but clean water service, install valves (such as gate or swing check) with bonnets upright so that (1) the bonnet cannot become clogged with debris

Example 5-1
Energy Penalties for Three Valves

Problem: Compare cone, butterfly, and globe valves for life-cycle costs of energy. Assume (1) electric power at \$0.05/kW $\times$ h, (2) a flow velocity of 3.05 m/s (10 ft/s), (3) headlosses based on the *K* factors of Table B-7, (4) a wire-to-water power efficiency of 75% for the pump, (5) interest at 8%, (6) a life of 20 yr, and (7) 300-mm (12-in.) valves wide open.

Solution: Calculate the annual cost of electric power, using headloss data from Table B-7, Equation 10-7 for the relation between head, power, pump efficiency, and flow rate, and Equation 29-4 for the present worth of an annual expenditure. From the results, it is evident that headloss is an important cost factor.

	Headloss		Energy cost (in dollars)	
Valve	m	ft	Annual	Present worth
Cone	0.02	0.06	24	235
Butterfly	0.15	0.5	211	2070
Globe	2.3	7.5	3010	29,600

that can render the valve inoperative, and (2) the removal of the bonnet is easier and maintenance is less expensive (this means valves should not be located in a vertical pipe).

- Make sure that the connecting piping is large enough for projecting butterfly valve discs to rotate.
- Place a spool (at least two or, better, three pipe diameters long) between a butterfly valve and an elbow to prevent the diagonal streamlines from causing the vane to flutter, excessively wearing the bearings or even locking the vane; vane shafts should be horizontal so there is no bottom bearing to collect grit.
- Place all valves in locations such that operation, maintenance, and repairs can be comfortably accomplished.
- Locate sleeve, grooved-end, or flexible couplings nearby so that (1) valves can be easily removed for replacement or for factory repair, and (2) movements and strain in the piping are not carried through the valve body (unless the valve body is designed to resist such loads).

Check Valves

In comparison with choosing other valves, the judicious selection of the right valve for check service is by far the most difficult and frustrating. Because there are many kinds and makes of check valves, choose the type and style only after careful study of Sections 5-4 and 7-1 and the literature of various manufacturers. Location is particularly important for check valves; for example, check valves on manifolds usually behave quite differently than isolated check valves. Check valves should, if possible, be placed no closer than four or five pipe diameters from a pump; otherwise turbulence from the pump tends to cause the valve disc to flutter and wear the bearings. (However, flutter is of less concern if the spring in a spring-loaded lever is stiff enough to prevent slam.) Furthermore, such a separation of pump and check valve allows the discharge pressure gauge to be placed farther from the pump where pressure fluctuation is much less violent, gauge wear is reduced (this is not a concern if the gauge piping contains a normally closed spring-loaded shutoff valve), and accuracy is greatly improved. The cost of the several fixed pressure gauges needed for several pumps can be greatly reduced by using a pair (one for suction, one for discharge) of portable gauges.

Safety

No automatic control system for valves is complete if it does not incorporate safety features to prevent damage from malfunctioning equipment. These safety features can be arranged to prevent the operation of an unprimed pump, to prevent the operation of a pump against a closed control valve that does not open on schedule, or to limit excessive pumping when discharging to a broken pipe. Most safety features include pressure-sensing devices and relays with timers that operate valves only when pressures are normal. When abnormal conditions occur, the valve should remain closed (or should be closed if open), and pumps should stop (after a suitable delay) to prevent damaging pumps and motors.

Flange Interfaces with Piping

Some styles of butterfly valves and check valves cannot be used with mortar-lined steel pipe or ductile iron pipe. See Sections 4.3 and 4.4 for discussions.

5-2. Isolation Valves

Isolation valves are either fully closed or fully opened. Valves that remain in one position for extended periods become difficult—even impossible—to operate unless they are "exercised" from time to time. Valves should be exercised at least once each year (more often if the water is corrosive or dirty), and the required exercise routine should be emphasized in the O&M manual.

Isolation Valves for Water Service

Isolation valves likely to be used for water service include

- Ball. These are expensive, so in sizes larger than, say, 100 mm (4 in.) they are used rarely for isolation only.
- Butterfly. These are popular in all sizes.
- Cone. These are also expensive, so in sizes larger than, say, 300 mm (12 in.) they are also used rarely for isolation only.
- Eccentric plug. These are excellent and useful in all sizes.
- Gate. These are popular in all sizes.
- Plug. These are either lubricated or nonlubricated, both of which are rarely used for only isolation in

sizes larger than approximately 50 mm (2 in.); the lubricated version is frequently used if the valve is to be closed for extended periods of time.

The gate valves most likely to be used are double disc or resilient seat, but solid wedge, knife, or even sluice gates may be useful in some circumstances. The resilient seat type (per AWWA C509 or C515) has become increasingly popular for both raw water and finished water. The plug valve most likely to be used is the nonlubricated type with either a rectangular or a round port, but a lubricated plug valve would be used for higher pressures. Lubricants approved by the FDA are available for water service. Globe valves are not normally used (because of their high headloss) except in piping 50 mm (2 in.) and smaller in which an ability to regulate flow is desired. For clean water, the double disc gate and butterfly valves are the most frequently used. The more expensive ball or cone valves are used for flow control, pump control, or powered check service, usually in conjunction with another valve for isolation so that the control or check valve can be repaired.

Isolation Valves for Wastewater Service

Valves for wastewater service are more limited in type because stringy materials catch onto and build up on any obstructions to the flow and sticky grit tends to lodge in any pocket, such as a valve seat. Depending on the quality of treated wastewater, however, valves for water service might be used, albeit with some risk.

Isolation valves likely to be used for wastewater service include:

- Ball (in unusual circumstances only)
- Cone (in unusual circumstances only)
- Gate (popular in all sizes)
- Eccentric plug (popular in all sizes).

The specific styles most often used are eccentric plug, knife gate, and resilient seat gate. Solid wedge gate valves with nonresilient seats might be used, but grit that can collect in the seat is often troublesome. Lubricated and nonlubricated plug valves and ball valves are used, especially if flow control is needed.

Description of Isolation Valves

The following descriptions are for both water and wastewater valves. The types of valves that are recommended or could be used for isolation service are given in Table 5-2.

Ball Valves

The rotor (round plug) in a ball valve rotates 90 degrees from fully closed to fully open. A ball valve has very low headloss in the fully open position because the bore of the pipe is carried straight through the ball, which results in headloss nearly the same as in a straight piece of pipe of the same laying length as the valve. Ball valves are of two basic types: (1) seat-supported, usually for valves smaller than 150 mm (6 in.), and (2) trunnion-supported, usually for valves 150 mm (6 in.) in size or larger as described in AWWA C507.

Ball valves in water, wastewater, and sludge pumping stations are not ordinarily used as isolation valves. Their laying length, weight, and cost are much greater than those of gate and butterfly valves. Seat-supported ball valves are widely used in auxiliary piping in such services as seal water, fuel oil, and natural gas as well as in isolating pressure gauges and air and vacuum valves because such piping is smaller than 75 mm (3 in.). An advantage of a ball valve is that it offers a relatively leak-free seal.

Ball valves for severe duty service (wastewater, storm water, surge control, and pump control) are usually of the trunnion type and should be selected with a great deal of care, especially with respect to materials for seats and bearings, because ball valves vary widely in quality. Seats are subjected to wear and tear from grit and tramp iron (nuts, bolts, scraps) in wastewater and storm water service, particularly in check and surge control usage. Bearings and shafts must be designed to withstand unbalanced forces during rapid closure on pump failure and during surge control episodes. Ball valves are often reserved for severe service conditions, and some engineers have experienced difficulties with cold-flow of resilient seat materials under high differential pressures. In some seat designs or under some operating conditions, those materials tend to escape the mechanism intended to retain them in the plug or body castings. Stainless steel and stainless steel against Monel metal are excellent alternatives for seats in such service—although these are not a universal panacea, and problems have been reported with metal-seated ball valves as well.

There are two kinds of mechanisms for operating ball valves. In most valves, a shaft allows the ball to rotate about a fixed axis so that the seat is wiped by the turning of the ball. In some metal-seated valves (e.g., Figure 5-1a), a loose-fitted trunnion allows pressure from the force main to push the ball against its seat. When the pump starts against the closed valve, the pressure moves the ball away from its seat and allows the valve to open freely. If the valve is closed before the pump is stopped, the action is reversed, and when

Table 5-2. Recommendations for Use of Isolation Valves

	Service usage[a]						
	Water		Wastewater				
Type of valve	Raw	Clean	Raw	Treated	Slurry	Gas	Fuel oil
Angle	G	G	X	G	X	G	G
Ball	E	E	E	E	E	E	E
Butterfly	G	G	X	G	X	—	—
Cone	E	E	E	E	—	—	—
Diaphragm	—	—	—	—	G	G	—
Gate							
Double disc	G	E	X–F	G	X	G	G
Knife	F–G	F–G	F–G	F–G	F	X	X
Sluice	G	G	G	G	—	—	—
Resilient seat	G	G	F–G	F–G	F	X	X
Solid wedge	G	G	F	G	X	G	G
Globe	X	G	X	F	X	G	G
Pinch	G	G	G	G	G	G	G
Plug							
Eccentric	E	E	E	E	F	—	—
Lubricated	G	G	G	G	G	G	G
Nonlubricated	G	G	G	G	—	F	F

[a]E, excellent; G, good; F, fair; X, do not use; —, use is unlikely.

the pump stops, the manifold pressure again seats the ball tightly. In effect, the action is like that of a cone valve but without its complex mechanism. The seat is thus never wiped except on power failure.

An alternative mechanism for accomplishing the same purpose (avoiding wiping the seat when the valve is opened or closed) is a movable retainer ring for holding resilient seat material. The inside surface of the ring is always exposed to the pressure in the force main. When the pump is started and its pressure exceeds force main pressure, the retainer ring retracts away from the seat and allows the valve to open freely. When the valve is closed and the pump subsequently stopped, pressure from the force main closes the seat and makes it drip tight. The retainer ring mechanism can be adjusted for a wide range of differential pressures. Valves in service for 20 yr show no signs of seat wear.

Designers should carefully investigate the performance history of the ball valve under consideration and be satisfied that the valve will be satisfactory under the conditions that will prevail in service. In addition to considerations of cost and trouble-free operating life under imposed conditions, the cost, difficulty, down time, and probable frequency of repairs should be weighed. Some valves can be repaired in place, whereas others must be removed and dismantled for repairs.

Trunnion-supported ball valves should be installed with the shaft horizontal and should be located in horizontal—not vertical—pipes.

A ball valve for pump, check, and surge control in water or wastewater service is usually fitted with a worm gear and compound lever (or pantographic) operating mechanism, as shown in Figure 5-1. The operating mechanisms may be fitted with a variety of actuators. Such a valve is superior to all others in providing ideal opening and closing characteristics for minimizing surges caused by pump start-up and shutdown or loss of power. As shown in Figure 5-2, the last 10% of Cv (a measure of flow) requires about 50% of the stem travel, so the last portion of flow is choked off very slowly. The parameter Cv is more precisely defined by Equations 5-1 and 5-2.

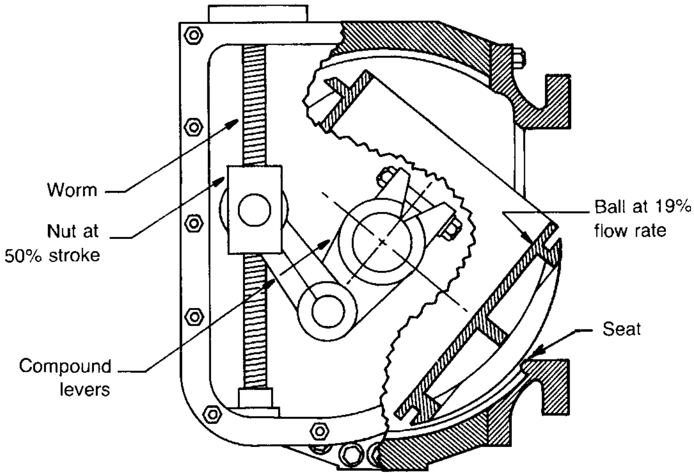
In SI units,

$$Cv = 2.919 \frac{d^2}{\sqrt{K}} \tag{5-1a}$$

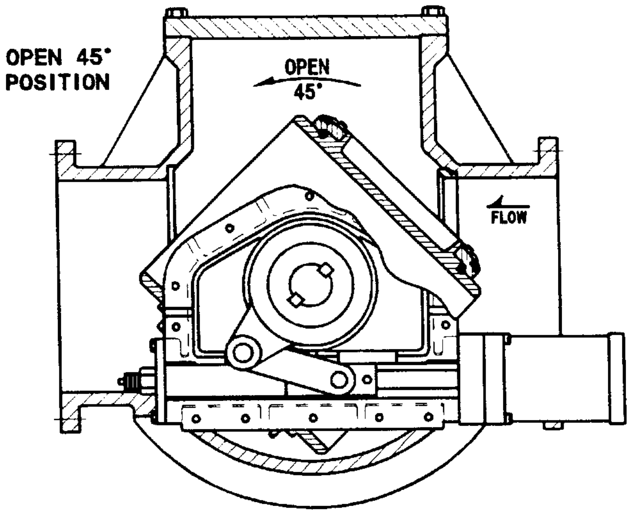
where d is the diameter of the valve (approach pipe, actually) in meters and K is the dimensionless valve coefficient in Table B-7.

In U.S. customary units,

$$Cv = 29.85 \frac{d^2}{\sqrt{K}} \tag{5-1b}$$



a



b

Figure 5-1. Ball valves with link and lever motion. (a) A metal-seated valve. Courtesy of APCO/Willamette Valve, Inc. (b) A resilient-seated valve. Note that removing the cover allows the seal ring to be replaced with the valve in situ. Courtesy of GA Industries, Inc.

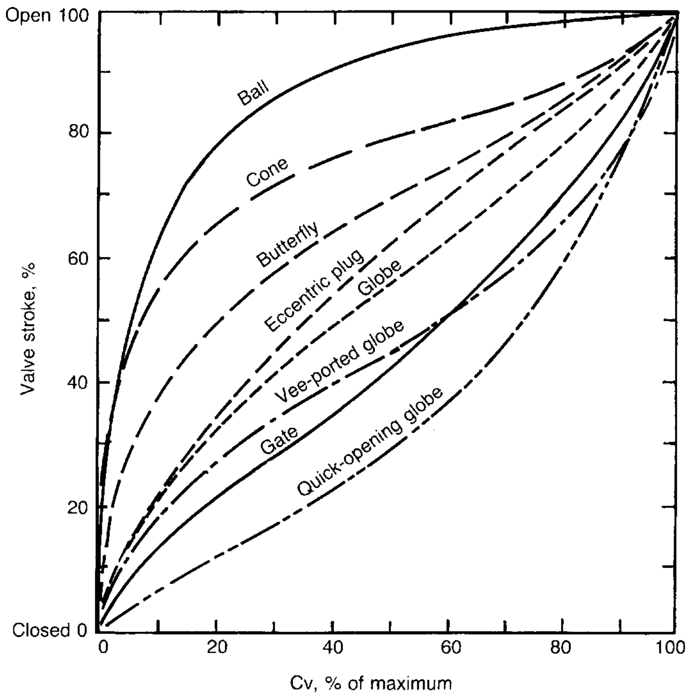
At any differential pressure across the valve, the flow rate in SI units is

$$Q = 0.3807Cv\sqrt{\Delta P} \quad (5\text{-}2a)$$

where Q is in cubic meters per second and ΔP is differential pressure in kilopascals. In U.S. customary units,

$$Q = Cv\sqrt{\Delta P} \quad (5\text{-}2b)$$

where Q is in gallons per minute and ΔP is in pounds per square inch. One can think of Cv as the gallons per minute of water at 60°F that flow through the valve at a pressure difference of 1 lb/in.^{2} An inconsequential correction for density at other temperatures is omitted from Equation 5-2 because the correction applicable in pumping stations would never reach 2%. As the valve is closed, Cv is reduced, as indicated in Figure 5-2. Historically, there has been no uniform or standard way of measuring the Cv or K factors in the waterworks valve industry.



LIVE GRAPH
Click here to view

Figure 5-2. Stroke versus *Cv* for various valves. After GA Industries, Inc., DeZurik Water Controls, Inc., and Willamette Valve, Inc.

Thus, it has not been possible to compare data among manufacturers and actually determine which valves truly have comparable headloss or *Cv* characteristics. In 2001, the AWWA published its manual M49 [12], which describes a uniform method of measuring and calculating such data.

Butterfly Valves

A butterfly valve (Figure 5-3) is a quarter-turn valve in which a disc (or rectangular vane for square channels) is rotated on a shaft so that the disc seats on a ring in the valve body. The seat usually is an elastomer bonded or fastened either to the vane or to the body.

Most, if not all, manufacturers have now standardized on the short body style (see AWWA C504). In the long body style, the vane is contained entirely within the body when the vane is in the fully open position. In the short body style, the vane protrudes into the adjacent piping when in the open position.

The wafer style, which is very thin, requires installation of the valve between pipeline flanges bolted across the valve body. When planning the use of short body or wafer-style butterfly valves, make sure the pipe on either side is large enough to accept the vane. Wafer valves are not recommended for isolating purposes where it may be necessary to remove the adjacent, connecting pipe spools. Some consultants refuse to use wafer valves under any circumstances.

Butterfly leakproof valves are used in both isolation and limited throttling service. These valves can be designed for shut-off, but leakage is significant without a resilient seat. Solids, wear, and scale buildup cause leakage even with resilient seats. In throttling service, the control range for the vane angle is about 15 or 20 degrees to 60 or 70 degrees. Because most butterfly valve designs are entirely unsuitable for throttling and some are prone to failure, select only valves recommended by the manufacturer for throttling. Then be suspicious. Investigate installations of valves proposed for throttling service. Pay particular attention to the seat design. The AWWA C504 standards alone do not ensure butterfly valve seats that are adequate for severe throttling (as in pump-control valves) where seats must be very rugged for longevity. Some valves have their rubber seats bonded into the body. Such valves are reliable, but when the seat needs to be replaced, the valve must be shipped to the manufacturer for repairs. Other valves have seats that are intended to be replaceable in the field, but after several years of service, corrosion may make it impossible to unscrew studs or nuts, and the

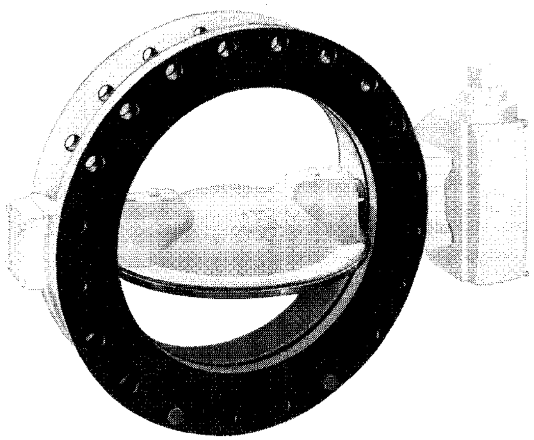


Figure 5-3. Butterfly valve. Courtesy of Henry Pratt Co.

valve must be sent to the shop for repairs anyway. So beware of sales claims concerning "easy replacement" in the field. Some (but by no means all) replaceable body seats are satisfactory.

Butterfly valves are used most frequently in water and air service. They are not suitable for wastewater, sludge, or grit service because the disc collects stringy solids and the seating edge is easily abraded. When used in a "dirty" service, such as raw water, install the valves with the vane axles horizontal to prevent grit from settling in the shaft bearings and oriented so that the bottom part of the vane moves in the direction of flow to wash solids through the valve when it first opens.

Butterfly valves should be located at least three (and preferably five) pipe diameters from an upstream bend and at least two pipe diameters from a downstream bend to place the valves in regions of approximately symmetrical velocities and coaxial streamlines. Valves that are closer to bends may chatter, and if they are very close, excessive forces may be required to operate them. Mounting valve shafts in the plane of the bend can eliminate the excessive forces, but it is best to orient shafts horizontally wherever possible.

Cone Valves

Cone valves, sometimes called "lift-action" plugs, are essentially plug valves that are positioned by first lifting the plug in the body before the operating mechanism moves the plug to its new position (see Figure 5-4). After reaching the new position, the plug is then reseated to provide a seal. The valve has excellent characteristics for surge-control and pump-control service. Cone valves have been used successfully in large water and wastewater applications. Compared with a ball valve, the axial motion in a cone valve reduces operating forces and seat wear and produces a tighter shutoff. However, because of the unusual stroking requirements, cone valve operating mechanisms require skilled maintenance personnel and can be complex and prone to failure.

A variant of the cone valve is the small taper-ground cock in which the plug is axially forced into tight contact by a spring. There is no mechanical lift, but a slight axial movement occurs anyway as the valve is turned. Cocks are useful for the isolation of taps, pressure gauges, and sampling ports.

Diaphragm Valves (Not Stem Guided)

A diaphragm valve contains an internal flexible elastomer diaphragm that presses down against the interior body wall (or, sometimes, against an internal weir that is part of the body). The diaphragm seals the valve body from the stem, so the valve is leak-proof and the stem requires no packing. This type of valve is useful in sludge and grit service because there is little obstruction through the body that could collect grit or solids. Cleaning tools such as pigs, however, cannot pass through them. These valves can be lined with various plastics such as PVC or polyethylene and are, therefore, often used in chemical piping (e.g., chlorine solution piping). They are not often encountered in raw wastewater, water, and sludge pumping stations because other types of valves are cheaper. Sometimes a diaphragm valve or its close relative, the pinch valve, is used as a safety device on a bypass pipe around a positive displacement pump to prevent destruction of the pump or main pipeline if a downstream valve were to be mistakenly closed. The diaphragm or pinch valve is set to open automatically, either by a preloaded spring or by air pressure, at a suitable overpressure. The lack of a visible position indicator is a shortcoming.

Another type of diaphragm valve is a stem-guided globe valve with a diaphragm in the bonnet to actuate the valve (see Section 5-5). The two types are in no way similar and should not be confused.

Eccentric Plug Valves

In an eccentric plug valve (Figure 5-5), both the body and the plug seat are offset from the center of rotation so that when the valve is opened the plug-seat surface rotates away from the body-seat surface; this movement minimizes the scraping and deterioration

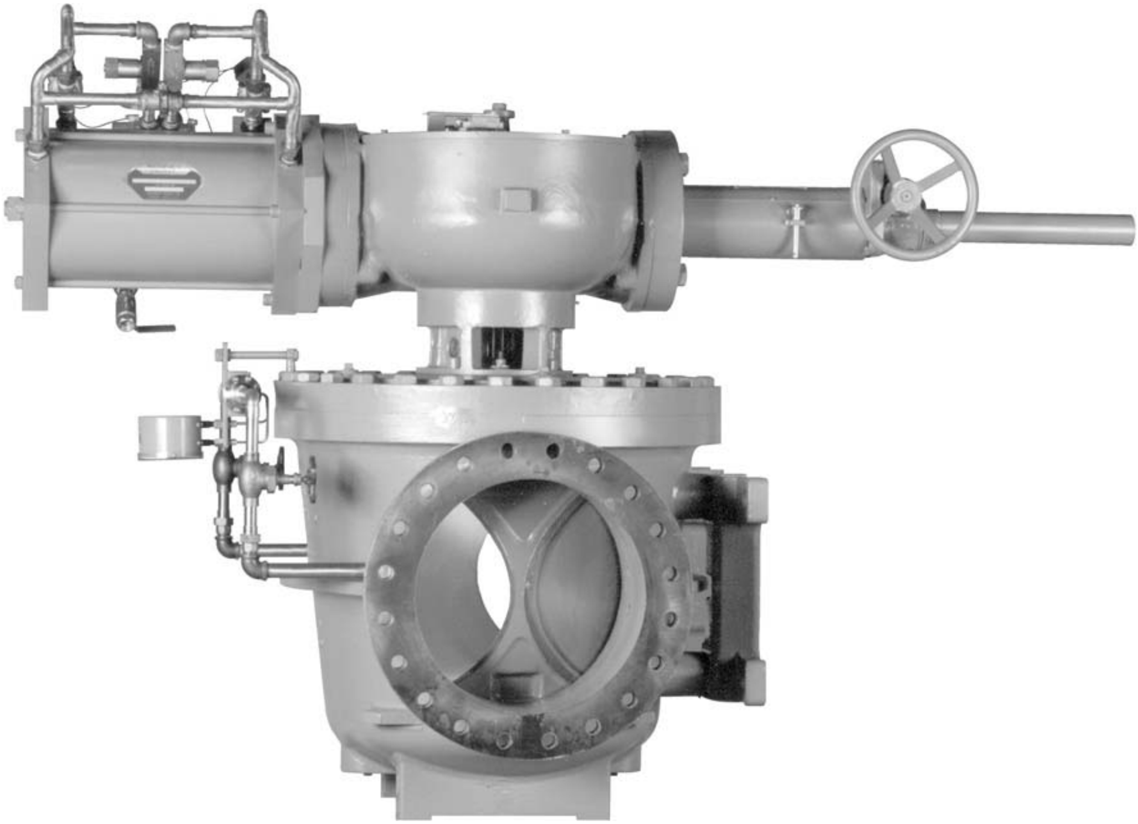


Figure 5-4. A 20-in. hydraulically operated Rotovalve® (cone valve) for pump discharge service. Courtesy of Rodney Hunt Co.

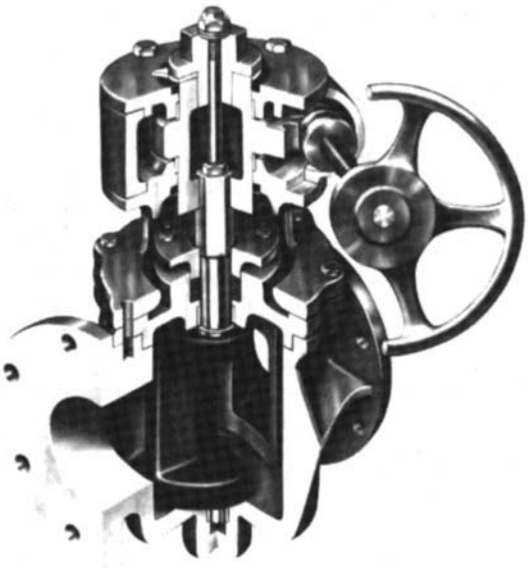


Figure 5-5. Eccentric plug valve. Courtesy of DeZurik Water Controls, Inc.

of seats common in other valves. In the open position, at a quarter-turn from closed position, the plug rests against the side of the valve body. In some makes of valves, the port is either rectangular or, at least, not full ported. In others, the port is circular and the same size as the pipe so that a pig can pass through the valve. Some styles or makes are difficult to service in the field. Plug valves are especially attractive in wastewater and sludge applications because there is nothing to become jammed or clogged with solids. The plug is usually coated with an elastomer, such as neoprene, to obtain a resilient seating.

Eccentric plug valves used in wastewater and sludge service, or for any fluid containing solids or grit, should be installed so that the plug rotates about a horizontal axis. The plug should be stored in the top when the valve is in the open position and should seat in the direction opposite the high-pressure side so that the pressure of the water forces the plug against the seat for a tighter seal. Because some designs hold pressure in only one direction, they must be installed to hold against the applicable pressure, which is often opposite to the direction of flow (as, for example, in the isolation valve on the discharge side of the pump). Debris problems are less severe if the body seat is upstream. The valve body may be marked to install the other way, so both the specifications and the O&M manual must clearly state both which way the valve is to be installed and the reason for the orientation.

When these valves are installed, tighten the flange bolts only enough to stop leaks. Excessive tightening squeezes the gasket material against the plug and causes it to bind. This cautionary note should also be placed in both the specifications and in the O&M manual. Add the warning that *all* valves (not just eccentric plug valves) should be exercised on a regular schedule.

Gate Valves

A gate valve has a disc sliding in a bonnet at a right angle to the direction of flow. Gate valves are further divided into several subtypes:

- Double disc (Figure 5-6)
- Solid wedge resilient seated (Figure 5-7) or metal seated
- Knife (Figure 5-8)
- Rising stem (shown in Figure 5-7)
- Outside screw and yoke (OS&Y), which also is often a rising stem design (as shown in Figure 5-7)
- Nonrising stem (NRS) (as shown in Figure 5-6)
- Inside screw and outside stem thread
- Bolted, screwed, or union bonnet

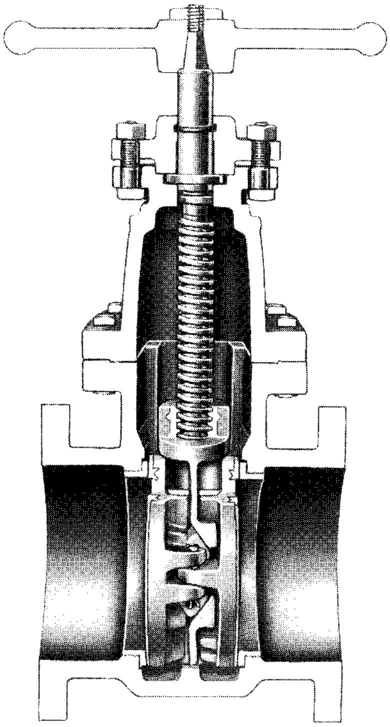


Figure 5-6. Double disc NRS (nonrising stem) gate valve. Courtesy of Mueller.

A rising stem (as shown in Figure 5-8) allows the operator or observer to easily determine if the valve is open or closed. An NRS (as shown in Figure 5-6) allows the valve to fit into cramped areas where a rising stem would strike a ceiling or wall. The NRS style should be avoided wherever possible because when it is used there is no indication of whether the disc is fully seated. If a hard object is caught on the seat, workers may assume the gate is closed. Dismantling a pressurized pipe, thought to be isolated, can lead to flooding and even loss of life. Hence, some indication of valve position is desirable for all valves. Furthermore, in the O&M manual, caution workers to always back off nuts by two or three threads and crack the joint to determine whether there is pressure. If there is, the nuts can be retightened until the trouble is corrected.

Gate valves are suitable only for isolation (open/closed) service. They should not be used for throttling service because a relatively small valve opening allows a high flow capacity [1]. Furthermore, gate valves are subject to damaging vibration when partly open.

Double Disc Gate Valves

The double disc gate valve equipped with a rising stem is one of the most popular types for clean water. After the discs drop into their seats, further movement of the stem wedges the discs outward to produce a leakproof shut-off even at pressures exceeding 1700 kPA (250 lb/in.^{2}). Opening the valve reverses the procedure. Hence, the discs do not slide until the wedging is relaxed, and sliding and grinding between the disc rings and body rings are thus minimized.

This type of valve should not be used if the water carries a heavy load of grit or solids that would fill the seats (pockets) and prevent the discs from first dropping into place. For such service, use resilient seated gate valves instead.

Solid Wedge Gate Valves

Solid wedge gate valves are suitable for water containing grit, solids, sludge, and other matter. These valves offer the advantages of full port opening and limited throttling service, and, strange though it may seem, they can be placed on vertical pipes or even upside down because the valve will work even when the bonnet is full of solids. Confer with the manufacturer, however, before installing the valve on a vertical pipe or in any position other than near upright. Consider, instead, plug or ball valves for such applications. Solid wedge valves are less expensive than the double disc type. The wedge minimizes sliding and scraping as the valve opens, and solids in the pocket can usually be displaced by opening and closing the valve several times. If the grit is sticky and the valve is to remain open a long time, however, consider a resilient seated gate instead.

Resilient Seated Gate Valves

The seat of a gate valve is a pocket that can entrap solids and prevent the valve from closing fully. The resilient seat type greatly reduces this problem because it has no pocket in the body in which the gate seats (see AWWA C509 and C515). Instead, the rubber edge of the disc seats directly on the valve body, as shown in Figure 5-7.

Because there is no pocket for the disc at the bottom of the valve to collect grit and debris, the resilient seated gate valve is suitable for grit-laden waters and wastewater as well as for clean water service. The disc is encapsulated with a resilient material (usually vulcanized rubber) that presses against the smooth, prismatic body of the valve. The valve is restricted to nearly horizontal pipelines and the bonnet must be oriented up or, at least, diagonally up. These valves can seal tight against working pressures up to 1380 kPa (200 lb/in.^{2}).

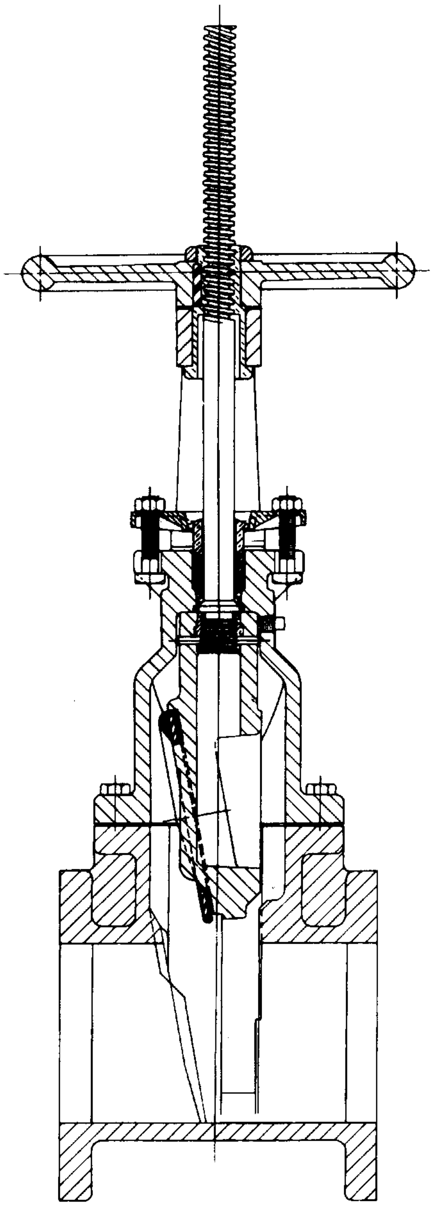


Figure 5-7. Solid wedge resilient seat OS&Y (outside screw & yoke) valve. Courtesy of Mueller.

Knife Gate Valves

The knife gate valve (Figure 5-8) is lighter and more suitable for water carrying debris than other gate

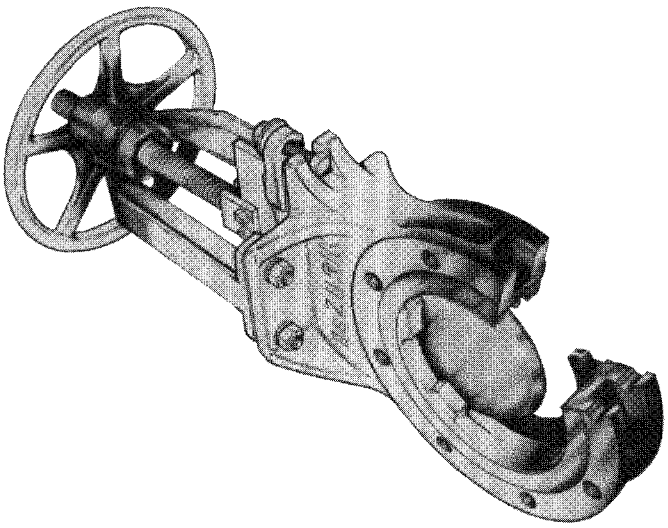


Figure 5-8. Knife gate valve. Courtesy of DeZurik Water Controls, Inc.

valve types, but it is more difficult to prevent leakage either through the closed valve or through the stem packing. Unless some leakage is acceptable and the head is low (say, 6 m or 20 ft), another type of valve should be selected. The knife gate is adapted for pressures below approximately 170 to 350 kPa (25 to 50 lb/in.^{2}).

Pinch Valves

A pinch valve is essentially a rubber or elastomer tube closed pneumatically or by a screw, wedge, or lever. The valve is leakproof and requires no packing, but it is not suitable for high pressures and the tube weakens where it is compressed. When closed pneumatically, it is an inexpensive and suitable valve for a bypass pipe around a positive displacement pump for guarding against damage due to a blockage or to an inadvertently closed valve. But it is limited in size, the tube is relatively expensive and difficult to replace, and the laying length is greater than for other valve types.

Plug Valves (Lubricated and Nonlubricated)

A plug valve contains a cylindrical or tapered plug with an opening cast or cut into it. A 90-degree turn of the plug fully opens or fully closes the valve. Hence, plug valves are considered to be quarter-turn valves. The design of the plug seat is such that solids do not accumulate and cause the plug to jam or bind. Plug valves can be obtained with full-ported plugs or with reduced port areas.

Lubricated plug valves (Figure 5-9) contain a lubricating system in which a nearly solid lubricant is forced into the top of the plug and through a series of grooves into the bottom so that the faces of the plug and seat are wiped with the lubricant, which functions as a deformable sealant each time the valve is open or closed. This feature is attractive in applications where

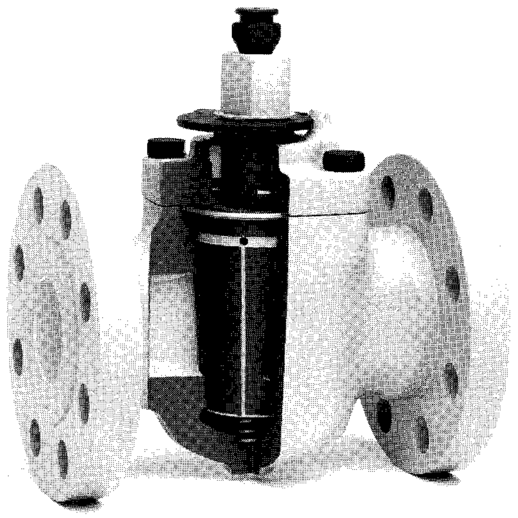


Figure 5-9. Lubricated plug valve. Courtesy of Nordstrom.

the valve may be fixed in the open or closed position for a long time. The valve is less likely to freeze in position and, if necessary, a small amount of lubricant can be forced into it to unfreeze it. Many different lubricants, which allow the valve to be used in different fluid services, are available. Lubricant can also be used in slurry service because there are no spaces that can become packed with solids. On the other hand, some fluids can dissolve the lubricant off the plug face, which would cause the valve to seize and gall, but this usually is no problem with the fluids encountered in water, wastewater, and sludge pumping stations. A plug valve provides a very tight seal and is especially useful in service pressures greater than 1000 kPa (150 lb/in.^{2}). In the cylindrical style, excess lubricant is forced outside of the valve body where it can be examined for contamination, less torque is required to turn it, and the rotor has a 100% bore. Both styles are excellent for pump control service. Lubricated plug valves require occasional but simple maintenance.

Nonlubricated plug valves have become unpopular, and several former manufacturers no longer make them. They seem to have no advantages over the lubricated types.

5-3. Sluice Gates, Shear Gates, Flap Valves, and Stop Plates

The devices in this section are not actually valves, but they are used as a means of flow isolation in channels and on ends of pipes such as those entering wet wells.

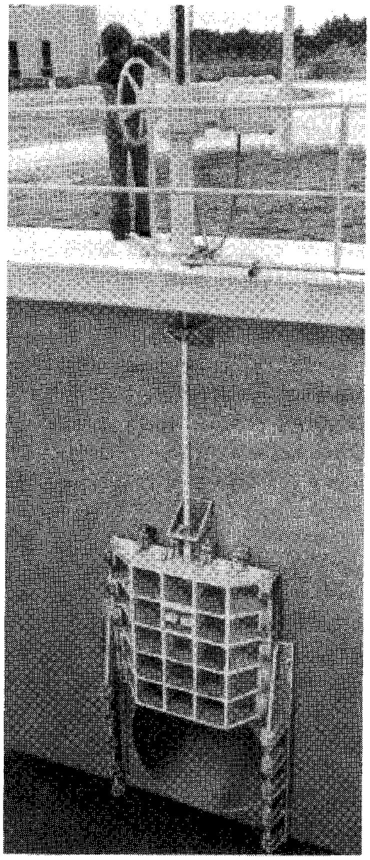


Figure 5-10. Sluice gate. Courtesy of Rodney Hunt Co.

Sluice Gate

Sluice gates (Figure 5-10) are used against a wall between two basins or between a pipeline and an open channel. Sluice gates are usually located where the influent channel or sewer enters the wet well and can be used if the pumping station

- is subject to flooding due to excessive influent flow or pumping failure;
- is critical, that is, if any major emergency repairs must be done as quickly as possible;
- has two wet wells; or
- has submersible pumps and the wet well must be dewatered when (or if) the pump lifting mechanism jams.

Stems and actuators must be sized to overcome large friction forces for the following reasons:

- Breakaway forces are sometimes greater than the manufacturer's typical recommendation, particularly if the gate remains closed for a protracted period.
- Power actuators can develop very large torques and thrusting forces. Therefore, it is prudent to size operating stems in a way that will more than adequately sustain those forces, especially when the fluid contains debris that can be trapped underneath the sluice gate leaf when closing.

Once the gate has been broken loose, the operating forces are reduced greatly.

Shear Gates

Shear gates are mounted on the ends of open pipes. Unlike conventional valves, they cannot be mounted in a pipeline. One side of the shear gate contains a

fixed bolt; on the other side is a lever that is used to lift and rotate the gate about the fixed bolt on the other side.

Flap Valves

Flap valves (see Figure 5-11) for pump discharge are substitutes for check and isolation valves and are an economical, reliable method of preventing backflow through out-of-service pumps. With this type of device, no isolating valve is installed. Rather, the flap valve is installed on the individual pump discharge piping at the point of discharge to the receiving sewer, channel, or discharge structure. A prudent engineer makes some provision (slide gate or bulkhead slots) for isolating the flap valve for maintenance purposes, but no expensive, heavy-duty isolating valves are required. Be sure to provide a vent just upstream from the flap valve to drain the pump discharge and prevent slam. Use flap valves with a cushion design specifically intended for pump discharge service.

Stop Plates

A stop plate is a thin, vertical, rectangular plate used to form a temporary dam in open channel flow. It is sometimes used in the wet wells of pumping stations to block flow to part of the wet well so that a pump and its suction piping can be dewatered for maintenance. The plate may have its own actuator or may be lifted by hand. Large plates can be lifted with a crane or hoist and stored on a rack or in a pit when not in service.

The plate is usually aluminum, but wood, fiberglass, stainless steel, and other materials are sometimes used. A local fabricating shop can make stop plates if supplied with detailed design information. Alternatively, a somewhat more sophisticated plate can be obtained from manufacturers, which means the engineer need not design such details as reinforcing.

Except for very small units, stop plates cannot be moved up or down when there is a substantial difference (more than about 0.2 m or 6 in.) in water level across the gate. If it is necessary to move the plate under such conditions, (1) a sluice gate may be used instead, or (2) a valve [typically a 100- to 200-mm (4- to 8-in.) gate or butterfly] or small stop plate can be mounted in a larger plate to allow equalization of water levels before the larger plate is moved. Stop plates are inexpensive, simple, suitable for local fabrication, and take up little of the valuable space in a wet well, but moving them is awkward and the leakage is high.

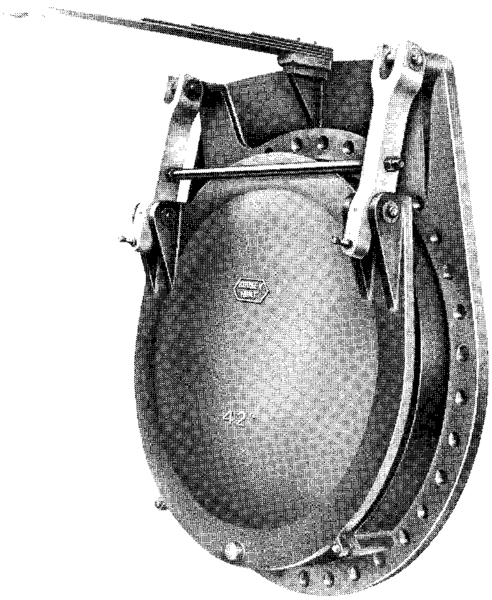


Figure 5-11. Flap valve designed for use with pump discharge. Courtesy of Rodney Hunt Co.

5-4. Check Valves

A check valve is usually (but not always) required to (1) prevent reverse flow and prevent runaway reverse pump speeds when the pump is shut off; (2) keep the pipeline full of water to prevent the entrance of air; and (3) minimize water hammer and surges for pump start-up and shut-down.

Vertical pipelines are poor locations for check valves if the water contains grit or solids. For vertically placed valves in clean water service, special springs or counterweights may be needed. Manufacturers that state that a check valve can be placed in a vertical pipeline are referring only to the springs or counterweights and ignoring the danger of deposited grit and solids, which can (and will) jam the valve.

The designer's responsibility is the selection of a valve that will give good service in keeping with the pump selection, hydraulics, and size of the system. The first decision is whether a check valve will serve, or whether a more sophisticated pump-control valve is necessary to limit surges. Some insights for this decision are contained in Chapters 6, 7, and 26 as well as in Parmakian [13], but only a sophisticated

mathematical model of the system solved by means of a computer can provide a rational analysis. Unfortunately, such modeling is time-consuming and expensive.

Valve Slam

Check valves can be divided into two broad classifications: (1) those that are closed by the static pressure of water above the valve (mechanical checks), and (2) those held shut by an external actuator (pump control or controlled check valves). The latter do not slam, but swing checks do if, before the valve is fully closed, any substantial reverse velocity catches the valve disc and accelerates it until it strikes the body seat abruptly. The sudden stop of disc, lever, and counterweight (if there is one) plus the violent impact of the disc on the body seat (especially if the contact is metal to metal) causes an explosive noise and vibrations that shake the pipe and may shake the whole building. The real problem is the water hammer that results if the water column is flowing backward at a significant velocity when the valve closes. However, valve slam can occur without water hammer and vice versa.

At worst, valve slam can rupture water lines and pump casings. At best, it is annoying. In between, it pounds the system, can overstress pipes and joints, and may well result in eventual leaks and greatly increased maintenance. It is difficult to give advice on the best kinds of valves to specify because valve slam depends on many interrelated factors in addition to valve design. Other factors that are just as important include static head, friction head, the inertia and spin-down characteristics of the impeller and motor, size of pipe, and velocity of flow. Generally, valve slam is caused or aggravated in the following ways:

Low flywheel effect. The principal cause of valve slam is quick deceleration of the pump due to low angular momentum of the impeller, the driver, and the water within the casing. With enough inertia, valve slam can be prevented, but the necessary flywheels may be large and costly.

High proportion of static head. If the headloss is 70% static and 30% dynamic (due to friction), valves slam worse than if the headloss is 50% static and 50% dynamic. A simple vertical lift (e.g., into an adjacent elevated tank) is especially prone to valve slam.

Frequency of valve slam. Valve slam may stress material beyond yield strengths and cause permanent deformations. A few deformations of a given intensity may be acceptable, but numerous deformations eventually cause leakage or rupture. Even a single slam, if severe, is dangerous.

Large pipe diameter. As valve size increases, the resulting time to close it increases, the disc velocity increases, and the energy in the system increases.

Parallel pumps. If two pumps are connected to a header and pump 1 shuts off while pump 2 is operating, pump 2 may cause the short water column between the two pumps to reverse very quickly and, thus, cause the check valve of pump 1 to slam.

Column separation. Water column separation can cause valve slam in two different ways: (1) rapid reversal of flow through the check valve, even though flow in most of the pipeline reverses slowly, and (2) a fast-rising positive pressure surge due to the collapse of a vapor cavity if the surge arrives at the valve when it is not closed fully (see Chapter 6).

Air chambers and surge tanks. These units can prevent column separation, but at the same time they can cause rapid flow reversal at the valve and thus aggravate valve slam (see Chapter 7).

Insufficient closing force. If the closing force due to the disc weight and spring or counterweight is low, the valve operates too slowly. But the closing force should not be so high that the valve does not open fully under steady-state pumping conditions. If the valve is not fully open, the headloss increases and debris is more likely to hang up in the valve. Also, excessive closing force can cause the disc to bounce off the seat so that valve slam recurs, sometimes two or three times.

Constant-speed pumps. Constant-speed pumps have two features that aggravate valve slam: (1) they must be turned on and off at full capacity (unlike variable-speed pumps), and (2) as they are turned off, their speed cannot be ramped down gradually unless soft starters equipped with soft deceleration features are specified in the motor control center. With variable-speed pumps, the speed can be ramped down during normal shutdown (although not when the power fails).

Friction in the hinge pin bearings. Friction is increased by dirt and corrosion. If the disc hesitates before moving, valve slam is almost certain to occur—mostly significant with tilting disc check valves.

Body shape. Details of check-valve design influence the closure operation. Because the disc must open wider in a valve with a straight body than in a valve with a bulbous body (Figure 5-12), the movement upon closing is correspondingly greater and the valve slam may be greater.

Inertia. Closing time increases with inertia of moving parts. A counterweight in a valve without a dashpot may therefore cause valve slam, and replacing such a counterweight with a spring sometimes min-

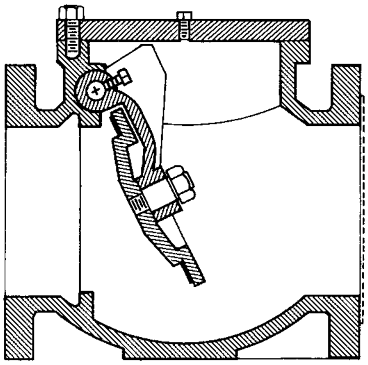


Figure 5-12. A swing check valve at full flow. After GA Industries, Inc.

imizes slam if the pumps have no significant spin-down time.

Preventing Valve Slam

Valve slam can be prevented, or at least kept within bounds, by

- using a valve that closes quickly—before the flow can reverse by adding a heavy counterweight or a stiff spring to the external lever;
- adding a dashpot or buffer to make the disc seat gently; or
- closing the valve with an external actuator so that the water column is gradually brought to rest without a significant increase in pressure.

The first two methods may prevent valve slam but do not necessarily prevent pressure surges.

Small Valves

For small valves, either confine swing checks to pipe less than, say, 250 mm (10 in.) in diameter or precede the valve with a reducer and follow it with an expander. Ordinary swing check valves are manufactured in large sizes, but there is a potential problem in using them in a low friction head system because they cannot close quickly enough.

Spring-Loaded Levers

Many engineers advocate the use of springs instead of counterweights to reduce the inertia of moving parts and thus speed the closure. However, as the valve closes, the spring tension relaxes and the torque on the disc shaft may decrease enough to be insufficient to prevent valve slam, so choose a design in which the combination of spring tension and the lever arm between the spring and shaft creates high torque at closure. A resilient seat aids in minimizing contact noise. These are the least expensive check valves.

Counterweight and Dashpot

A counterweight has the advantage of providing maximum torque on the disc shaft at closure, but it does not close the valve as quickly as a spring because of the inertia of moving parts. Some professionals (especially manufacturers) think the valve should be equipped with either (1) a side-mounted or top-mounted oil-filled dashpot to cushion the movement of the lever at shut-off, or (2) a bottom-mounted, piston-type shock absorber that engages the disc before it closes. Air-filled dashpots are difficult—occasionally impossible—to adjust to prevent valve slam. The required massiveness of construction needed to resist the high force of water against the disc and the dashpot mechanism makes this valve more expensive, but it is the recommended style when the spring-loaded lever type is inadequate. But note, however, that a heavy counterweight or a stiff spring, properly adjusted, is cushioned by the water in the valve.

Pressure-Regulated Bypass Dump

A spring-loaded, pressure-actuated surge relief valve (Chapter 7, Figure 7-8) with a pipeline returning the wasted water to the wet well can reduce the surge to an acceptable, preset level, but it does nothing to mitigate valve slam.

Actuator-Controlled Plug or Ball Valve

An actuator can be programmed to both open and close the valve slowly enough to prevent water hammer (see Chapter 7, Figure 7-7). A stored energy system is needed to operate the valve when power failures occur.

Summary

Selecting a proper type of check valve and control mechanism is more art than science. Experience, not analytical theory, is a key consideration. Of course, a simple method to determine whether a conventional

swing check valve can be used without excessive valve slam would be desirable, but unfortunately the complexity of the problem precludes a simple, accurate procedure. Complex computer programs can be used to predict with fair accuracy whether valve slam will occur. The cost of the analysis may be discouraging if the system is small, but large systems should always be so analyzed. If any general statement on check-valve selection can be made, it is probably this: use a swing check valve with an outside lever and spring. If that is inadequate, use a valve with a cushioned closure system such as a dashpot or bottom buffer. As a last resort, use a powered actuator. But note that even the experts disagree, and some prefer counterweights to springs.

Check Valves for Water Service

Check valves useful for water service include:

- Swing check valves
- Center-post guided (or silent) check valves
- Double leaf (or double door, double disc, or split disc) check valves
- Foot valves
- Ball lift valves
- Tilting (or slanting) disc check valves.

The several styles of swing check valves can be divided into those with and those without an outside lever. Outside levers can be equipped with either springs or counterweights, and the levers can be cushioned or noncushioned. Bottom buffers can be used instead of dashpots affixed to the outside lever.

Check Valves for Wastewater Service

Valves for wastewater service must be capable of passing large solids and, as with isolation valves, must have no obstructions to catch stringy material. Valves likely to be used for wastewater are essentially limited to:

- Swing check valves
- Flap valves, which might be used in special circumstances (for example, with combined sewers that contain storm water and wastewater)
- Ball lift valves, which are useful in positive displacement sludge pumps as the ball can be lifted completely out of the flow path.

The rubber clapper swing check valve has no outside lever, is not fully ported, and, hence, should not be used for raw wastewater or sludge.

Description of Check Valves

The following descriptions of check valves for water and wastewater offer some guidance and suggestions. A summary of recommendations for use is given in Table 5-3.

Ball Lift Check Valves

A ball lift check valve contains a ball in the flow path within the body. The body contains a short length or guide piece in which the ball moves away from the seat to allow the passage of fluid. Upon reverse fluid flow, the ball rests against an elastomeric seat.

Table 5-3. Recommendations for Use of Check Valves[a]

	Water		Wastewater		
Type of valve	Raw	Clean	Raw	Treated	Sludge
Ball	E	E	E	E	E
Ball lift	F–G	F–G	F–G	F–G	G
Center-post guided	P	F	X	X	X
Double door	X	G	X	X	X
Flap	—	—	G	—	—
Foot	F	G	—	—	—
Swing					
No outside lever	P	P	P	P	P
Outside lever and counterweight	F–G	F–G	F–G	F–G	G
Outside lever and spring	F–G	F–G	F–G	F–G	F
Outside lever and air cushioned	F	F	F	F	P
Outside lever and oil cushioned	G	G	G	G	G
Slanting disc	G	G	X	F	X

[a]E, excellent; G, good; F, fair; P, poor; X, do not use; —, use is unlikely.

These valves are often encountered in pumping stations in sizes of 50 to 150 mm (2 to 6 in.) or smaller for pump seal water or for wash water supply piping. They have also been successfully used by at least one major pump manufacturer for raw wastewater pump discharge piping up to 600 mm (24 in.). Except for small sizes, bodies are made of ductile iron. The ball is hollow with an external rubber coating resistant to grease and dilute concentrations of petroleum products, acids, and alkalies. The specific gravity of the balls can be adjusted to suit a wide range of operating conditions.

The valves are said to be self-cleaning, rugged, reliable, nonclogging, and to be able to withstand repeated cycling, because each time the ball is reseated, a different part of the surface rests on the seat. In larger sizes [100 mm (4 in.) or more] for sludge pumping service, ball lift checks are part of the mechanism in plunger (piston) sludge pumps.

As check valves for wastewater pump discharge piping, the valves have these advantages: (1) the headloss is lower than it is for other types; (2) there are no external penetrations and no leakage to the outside (although good swing check valves properly set up do not leak either); and (3) stringy materials have nothing to wrap around and do not foul the valve. On the other hand, the standard valve (unlike swing check valves) gives no indication of whether water is flowing—a serious disadvantage. Ball lift check valves are, however, available with ball position indicator-proximity switches.

Decisions to use a ball lift check valve instead of, say, the faster-closing swing check valve with a spring-loaded lever should be based on a computer-aided dynamic hydraulic analysis of the system.

Center-Post Guided Check Valves

Center-post guided check valves are low-cost and are called "silent check valves" by some manufacturers. They close more rapidly than any other check valve. As shown in Figure 5-13, the disc is held closed by a spring until the pump is started. The spring selection is very critical; it is the differential pressure across the valve (difference between static head and TDH) that must be specified and not the safety pressure rating of the system. An incorrect specification results in valve slam.

Three disadvantages of this valve type are that (1) the operating mechanism is enclosed so the valve must be removed for servicing; (2) there is no external indicator of the position of the disc; and (3) the headloss is high.

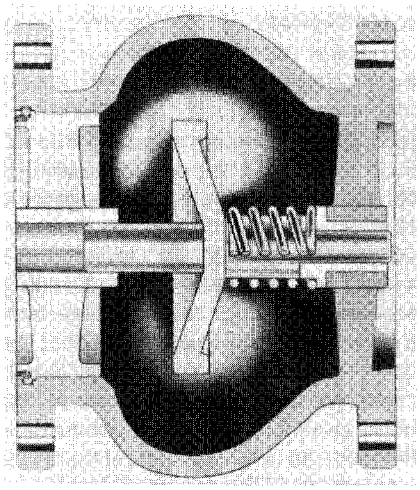


Figure 5-13. Center-post guided "silent" check valve. Courtesy of APCO Valve & Primer Corp.

Double Leaf Check Valves

Double leaf (also double door, double disc, or split disc) check valves contain two hinged half-discs in a short body. The two half-discs are hinged in the middle and contain a spring that forces them closed. This type of valve has no connecting flanges of its own. Instead, it is inserted between two adjacent pipe flanges. It can be installed in either the horizontal or vertical position.

These valves should never be used in wastewater or sludge service or in abrasive conditions because the hinge and discs can catch solids and the seat and discs would wear in abrasive service. Double leaf check valves are small, light, and inexpensive and have a short laying length. They close very quickly but they cannot be adjusted from the outside, nor can they be cushioned, so a sudden flow reversal can cause slam and water hammer. Shut-off is not leak proof. Other disadvantages are (1) the valve must be removed to service the mechanism; (2) there is no external indication of whether the valve is open or closed; and (3) they are subject to a fluttering motion caused by vortex shedding as the fluid moves past the valve plates. If the fluid velocity is less than 3.4 m/s (11 ft/s) and the valve is at least eight pipe diameters downstream from any source of flow disturbance such as a pump or a fitting, the problem is reduced [14].

Foot Valves

A foot valve is a special design of a lift check valve. It is used in the suction line of a sump pump to prevent

loss of prime. It is designed for upflow and is attached to the bottom of a pump suction pipe.

Foot valves are prone to leakage, especially when used in fluids containing abrasives and solids, and they are difficult to service. Foot valves decrease the net positive suction head available ($NPSH_A$, see Section 10-4). In a raw wastewater pumping station, a better choice would be to use a self-priming pump if a conventional wet well–dry well pumping station cannot be used or is not feasible.

Lift Check Valves

The body of a lift check valve is similar to that of a globe valve. A plug or stem moving within a guide lifts upward and allows fluid to pass through the valve. The plug seats when the flow reverses.

A lift check valve does not provide a tight shutoff. It cannot be used in fluids containing solids or abrasives, and gum-forming fluids can cause the stem to stick. Sudden flow reversal can cause water hammer.

This type of valve is normally encountered in pumping stations only in sizes 50 mm (2 in.) and smaller and in services such as utility water and compressed air.

Swing Check Valves

A swing check valve (Figure 5-12) contains a hinged clapper or disc that rests on a seat and prevents fluid from flowing backward through the body. A disadvantage of metal-to-metal design is the lack of a tight seal when the disc is seated, so a rubber seat is better. The disc is usually affixed to a hinge pin by means of an arm. The pin and arm allow the disc to move up and out of the flow path in the direction of fluid flow.

Swing check valves can be installed in both horizontal and vertical positions. In a horizontal position, the valve bonnet must be upright. In a vertical pipe, the valve must be installed so that fluid flow is in the upward direction, but never install swing check valves in vertical pipes in wastewater, sludge, or slurry service because rags, debris, and grit would settle against the disc and eventually prevent functioning. Clearing the valve in this position is a messy, disagreeable task.

Slamming when a pump stops and the fluid reverses direction is a significant problem when using swing check valves of some designs. In general, swing check valves larger than 150 mm (6 in.) should have an outside lever and spring to close the disc quickly before the fluid can reverse direction. Note, however, that a commonly used check valve standard, AWWA C508, does not cover the outside lever and spring design. A disadvantage of this type of valve is that the outside lever and spring or counterweight can prevent the disc from opening fully, especially at low flow velocities—less than about 3 m/s (10 ft/s)—with a consequent increase of headloss. The swing check valve in Figure 5-12 is fully ported when open 20 degrees, but most designs require a swing of 60 degrees to open fully. Headloss through the valve at low velocities is generally higher than the manufacturer's data, which are usually based on a fully open valve disc. If the valve is properly chosen for the specific application and the spring tension or the counterweight properly adjusted, however, the headloss should agree with the manufacturer's data. Headloss increase is often caused by increasing spring tension or the weight on the lever arm to reduce slam—a direct result of improperly selecting the valve. Swing check valves in pipes larger than 400 or 450 mm (16 or 18 in.) should be specified with caution, especially if the head exceeds about 15 m (50 ft), because the force on the disc is enormous.

Cushioned Swing Check Valves

Some check valve manufacturers offer pneumatic and/or hydraulic dashpots attached to the valve to regulate the speed of closure of the disc upon water column reversal. The cushioning system consists of:

- a weighted lever arm attached to the disc pin or axle
- a piston mounted outside of the valve body and contained in a cylinder (dashpot) attached to the weighted arm.

As the fluid velocity decreases, the weighted lever arm forces the disc to close, and the piston moves downward in the cylinder. The piston compresses the air (in a pneumatic system) or displaces oil through an orifice (in a hydraulic system). Adjusting the valves on the pneumatic or oil lines (or the orifices in the dashpot) controls the rate of closure.

The hydraulic system offers better control than the pneumatic system, which often does very little to reduce the slam. Sturdy valves can be closed quickly or slowly and can even be closed in two or three stages, such as quick closure to 50%, moderate speed of closure to 95%, and slow closure to shut-off.

Be very careful in selecting applications for these valves; close coordination with the manufacturer is necessary. In addition, field adjustment after installation is needed to set the closing controls properly. These valves are vulnerable to tampering.

Rubber Flapper Check Valves

The rubber flapper swing check valve is a swing check that is entirely enclosed. The seat is on a 45-degree angle and the steel-reinforced flapper need travel only about 35 degrees to reach the fully open position. The short stroke and light weight of the flapper make it capable of very fast shut-off, which, combined with the resilient seat, reduces slam. The construction of the valve is simple, as is maintenance. There is no outside lever, no way of adjusting the closing force, and no way of determining whether the valve is open or closed. This type of valve should not be used in raw wastewater service because debris can pack above the disc and prevent the disc from opening.

Slanting Disc Check Valves

A slanting disc check valve contains a disc balanced on a pivot. Instead of being perpendicular to the longitudinal axis as in conventional swing check valves, the seat is at an angle of 50 to 60 degrees from the valve longitudinal axis. Slanting disc check valves should only be used in water service; rags and solids present in raw wastewater and sludge would hang up on the disc.

The advantages of this type of valve are (1) headloss is low (although not as low as in a swing check valve) in the open position because the vane or flapper is designed as an air foil; (2) various pneumatic and oil-filled dashpots can be used to control the opening and closing speeds; and (3) the performance of these controls can be adjusted in the field. The disadvantages are (1) velocities less than 1.5 m/s (5 ft/s) do not fully open the vane; (2) the disc oscillates in the flow and the bearings wear on the bottom, so the valves begin to leak; and (3) the valve is not fully ported.

The two controls most frequently encountered are bottom buffers and top-mounted dashpots. These two systems are sometimes mounted together on one valve.

The bottom buffer consists of an oil-filled cylinder in which a piston is moved by the closing disc or vane. The disc moves freely for the first 90% of its closure, then strikes the buffer piston, which can be adjusted in the field to control the last 10% of disc travel.

The top-mounted oil dashpot system allows both the opening and closing speeds of the disc to be adjusted over the full range it travels. This adjustment can be especially valuable with pump start-up because the opening speed can be regulated to open the valve slowly, which greatly reduces hydraulic transient effects caused by pump start-up. A disadvantage is the high load exerted on the mechanical linkage when the pump reaches shut-off head. However, no electrical interconnections between the pump motor control center and check valve are needed.

5-5. Control Valves

Control valves are used to modulate flow or pressure by operating in a partly open position, thus creating a high headloss or pressure differential between upstream and downstream locations. Such operations may create cavitation and noise. If there is a large pressure differential and the limits of operation are approached or exceeded, the discs tend to flutter and bearings may wear quickly. Valve seats are especially vulnerable to wear because, if the pressure differential is high across the seat, small channels may be cut (called "wire drawing"), which prevents a tight seal, aggravates the wire drawing, and makes frequent replacement necessary.

To minimize wasting energy and to increase the life of the valve, it is desirable to minimize the time of operation at partly open positions. If the valve must throttle flow for extended periods, choose a style well adapted for the purpose and select hard materials for those parts that wear quickly.

Some control valves may be manually operated (for example, needle valves used to control the flow of a fluid in a valve actuator). Most control valves, however, are power-operated by programmed controllers. These valves are used for a variety of purposes: pump control, check valve control, control or anticipation of surges, or control of pressure or flow. The power source can be (1) hydraulic (usually oil), (2) pneumatic, (3) a combination of pneumatic and oil, (4) electric, or even (5) the pressure of the pumped water. All control methods feature some kind of adjustable-speed actuator, sometimes with three electric speeds that depend on the position of the valve mechanism. Whatever the power source, a backup is needed for power outages. The backup can be a pressure tank for pneumatic or hydraulic actuators or trickle-charged batteries for electric actuators (see Section 5-6).

Control valves are selected on the basis of the requirements of the hydraulic system and the characteristics of the pump. A major decision is whether to use a check valve that is controlled by the flow or a more sophisticated valve that itself controls the flow. The characteristics of the type—and even the brand—of pump-control or check valves are important. Every type of valve used as a check valve suffers some of the effects of cavitation, noise, and vibration while opening and closing, and some types are more

vulnerable than others. Cavitation occurs at regions of large pressure drops.

Pump-Control Valves

Pump-control valves can be any type—angle, ball, butterfly, cone, globe, or plug—suitable for the liquid being pumped. Use angle and globe valves where high headloss can be tolerated or is desirable (as in bypass pipelines); and use ball, butterfly, cone, or plug valves where energy costs are important (see Example 5-1). Controls and electrical interlocks are provided so that the valve is closed when the pump starts. After the pump starts, the main valve opens slowly at an adjustable rate. When the pump is signaled to shut off, the valve slowly closes at an adjustable rate. When the valve is 95 to 98% closed, a limit switch assembly shuts off the pump.

Surges induced by start-up and shut-down of constant-speed water pumps can be effectively controlled by diaphragm- or piston-operated globe-type valves utilizing differential pressure to open and close the valve. Operation is usually initiated by activating solenoid valves that act on the trim piping controlling pressure on the diaphragm. The initiation of solenoid operation is usually linked electrically to the pump motor control circuit, and the speed of operation is controlled by adjusting needle valves in the trim piping. Variations of this basic type of valve include straight-through or angle bodies, surge relief valves, and head sustaining valves. To provide some assurance of reliability, the trim piping to the power side of the diaphragm must be fitted with a fine strainer to remove particulate material that might otherwise interfere with valve operation.

Piston-operated globe valves have an advantage over diaphragm-operated valves in that leakage from the valve occurs long before failure. Diaphragm-operated valves are completely sealed and do not leak, but, on the other hand, they give no warning of impending diaphragm rupture, which puts the valve out of service. Both valves are very effective in reducing surges due to pump start-up and normal pump shutdown, but they cannot prevent surges caused by power failure.

Power-actuated ball, butterfly, cone, and plug valves are more expensive to install but, when fully open, cause less headloss than other valves.

Control Valves for Water Service

The control valves likely to be used for water service include angle, ball, butterfly, cone, globe, needle (for fine flow regulation in control piping), and eccentric, lubricated, or nonlubricated plug valves. See Figure 5-14.

Control Valves for Wastewater

The only valves suitable for control of wastewater are ball, cone, long radius elbow, and eccentric, lubricated or nonlubricated plug valves.

Description of Control Valves

Except for globe and needle valves, all of the valves that can be used for control are described in Section 5-2. Recommendations for their use are given in Table 5-4.

Angle Valves

Angle valves and globe valves are similar in construction and operation except that in an angle valve, the outlet is at 90 degrees to the inlet and the headloss is half as great as it is in the straight-through globe valve. An angle valve is useful if it can serve the dual purpose of a 90-degree elbow and a valve. Conversely, an angle valve should not be used in a straight piping run;

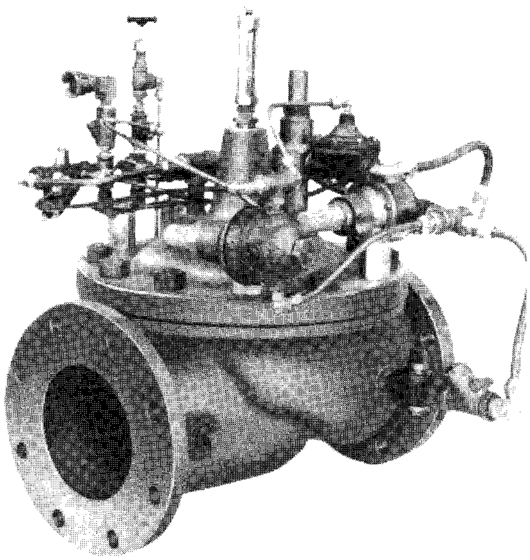


Figure 5-14. Control valve for water service with external piping arranged for surge anticipation. Courtesy of Cla-Val Co.

Table 5-4. Recommendations for Use of Control Valves[a]

Type of valve	Water		Wastewater	
	Raw	Clean	Raw	Treated
Angle	F	G	X	X
Ball	E	E	E	E
Butterfly	F	G	X	F
Cone	E	E	E	E
Globe				
Diaphragm	G	G	X	F
Differential piston	G	G	X	F
Surge relief				
Diaphragm or piston	G	G	X	X
Angle valve (for water)	G	G	—	—
Long radius elbow valve (designed for wastewater)	G	G	F	G
Surge anticipation				
Diaphragm or piston	G	G	X	X
Angle valve (for water)	G	G	—	—
Long radius elbow valve (designed for wastewater)	G	G	F	G

[a]E, excellent; G, good; F, fair; P, poor; —, use is unlikely; X, not recommended.

instead, use a globe valve. As with globe valves, angle valves are best used in clear liquid service because fluids containing grit or abrasives cause severe seat erosion. Globe valves must never be used in sludge or raw wastewater service because they are prone to becoming plugged with solids.

Globe Valves

As in the angle valve, a globe valve has a disc or plug that moves vertically in a bulbous body. Flow through a globe valve is directed through two 90 degree turns—upward and then outward—and is controlled or restricted by the disc or plug. The pressure drop or headloss is higher than in angle, gate, butterfly, or ball valves.

Because of this high pressure drop—even in the wide-open position—globe valves are not ordinarily used as isolation valves except in seal water, gas, and fuel oil pipelines. They are used in applications requiring throttling, such as pressure or flow control. Globe valves are suitable for clear liquid service, but not for fluids containing grit or abrasives, which cause severe seat erosion. Never use globe valves for wastewater or sludge service because they are prone to becoming plugged with solids.

Globe or Piston Valves with Vee-Ports

Globe or piston valves containing vee-ports are made to eliminate seat wear by cavitation and to allow the flow to start in a controlled manner. Throttling is by the vee-ports as shown, for example, in Figure 5-15. The first 50% of the stroke allows only about 20% of full flow—a feature that minimizes the effects of

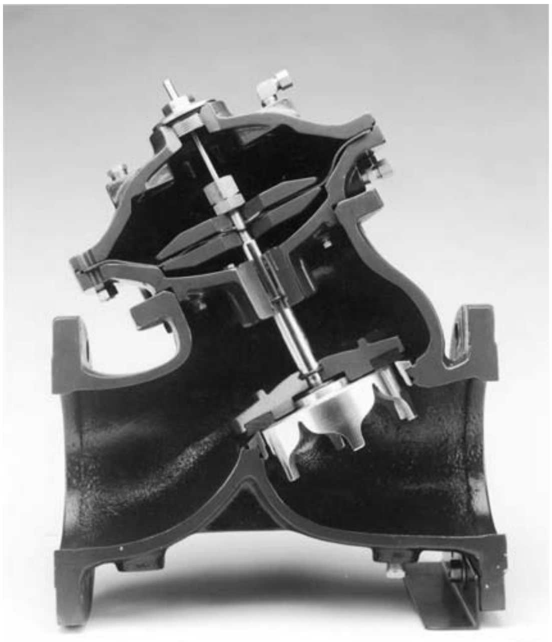


Figure 5-15. Vee-ported low headloss control valve in wye pattern fitted with vee-port throttling plug. Flow is from right to left. Courtesy of Bermad, Inc.

surge caused by opening and closing pump-control valves.

Needle Valves

A needle valve is a special variation of a globe valve. The plug is a slender tapered needle. The flow annulus is easily fouled by particulates; otherwise, its characteristics and advantages and disadvantages are much the same as those of a globe valve.

A needle valve offers very fine pressure and flow control, even in a low flow range with the valve almost completely closed. It is often used in seal-water piping and, to control the speed of operation, in the control piping of other valve types.

Special Control Valve Functions

Some control valves regulate parameters such as pressure rather than flow, and these valves may operate by being either fully open or fully closed. Most special-purpose control valves are built on a single body design. Only the exterior piping to the hydraulic actuator (diaphragm or piston) in the bonnet is changed to effect the type of control wanted, whether it be constant flow, constant pressure, or proportional flow.

Make it easy to service these valves by incorporating enough isolating valves to close off the water supply to them.

Altitude Control Valves

Altitude control valves are used to add water to reservoirs and to one-way tanks used in surge control (see Chapter 7). The body design is usually of the globe type. Altitude control valves are made in many variations of two functional designs:

- One design in which the valve closes on high water level in the tank and does not open again until the water leaves through a separate line and the water level in the tank falls.
- A second design in which the valve closes on high water level in the tank and opens to allow water to flow out of the tank when pressure on the valve inlet falls below the reservoir pressure on the downstream side of the valve.

Pressure Relief Valves

Pressure relief valves are often of the globe type in terms of body design. A control system is added to establish how the valve operates. In the angle type (Chapter 7, Figure 7-8), a direct-acting, adjustable spring is provided to open the valve and permit flow when the inlet pressure exceeds the spring setting. A common application is to place one of these valves on the branch of a tee on a pump discharge line. The valve then serves to release the fluid before a high pressure can develop and overstress the piping and valves. There is an inherent time lag in the opening of any valve, however, so the valve lags to some degree behind the actual pressure rise due to surges.

Surge Anticipation Valves

The entire surge anticipation system consists of a tee on the header, an isolation valve (to permit servicing the surge anticipation valve), the surge anticipation valve itself, a vent pipeline to waste, a pressure-sensing pipeline connected to the pump discharge, and a pilot system with an electronic timer. Following pump power failure, the pressure in the pump discharge drops, which opens the valve and vents water in anticipation of the subsequent high-pressure wave. The control system should be designed so that the valve does not open immediately after power failure but only after a timed delay (i.e., not until the pressure wave in the pipeline approaches the pumping station). The electronic timer keeps the valve open for a short period and then closes it slowly. If a second pressure wave follows, the valve reacts like a surge relief valve.

Although this type of valve can significantly reduce the return upsurge or high pressure after a pump power failure, it does nothing to control or reduce the effects of the initial downsurge or low-pressure wave (see Chapters 6 and 7).

5-6. Valve Actuators

Actuators (also called "operators") for valves can be manual or electrically, hydraulically, or pneumatically powered. Valve design (quarter-turn or lift type), valve size, operating pressures, and special requirements such as operation on loss of power or control of surge pressures determine the type and complexity of the valve actuating system. A comparison of valve actuators is given in Table 5-5.

Manual Actuators

Plug and butterfly valves 150 or 200 mm (6 to 8 in.) in diameter and smaller and ball valves 100 mm

Table 5-5. Comparison of Various Types of Valve Actuators

Type	Cost	Operating characteristics	Operation	Remarks
Manual, direct	Least	Relatively smooth	Operator must be in attendance.	Suitable for smaller valves.
Manual, geared	Low	Smooth, slow	As above	Suitable for valves up to 450 mm (18 in.) in diameter.
Electric	Moderate to expensive depending on options	Smooth; can position only in increments unless provided with electronic positioner.	Avoid unless stored-energy (batteries) reserve system is provided.	Can be expensive depending on size, functions, and number of valves.
Pneumatic	Moderate	Tends to be jerky, difficult to position.	Easily provided with local receiver.	Corrosion, caused by water in compressed air, can be troublesome. These systems can freeze up at exhaust ports.
Hydraulic, water	Moderately expensive	Very smooth	Easily provided with local hydropneumatic tank	Corrosion and freezing can be problems.
Hydraulic, oil	Expensive	Very smooth	Good, with precharged gas accumulators.	System can be complex. Reliability achieved only by using first-class components; recommend system pressures less than 14,000 kPa (2000 lb/in.^{2}).

(4 in.) in diameter and smaller can be actuated with a simple lever attached to the plug shaft. Some lever-type actuators can be fitted with adjustable stops for balancing or throttling service. This feature, however, is rarely necessary for isolating valve service. Quarter-turn valves larger than the sizes indicated here should be fitted with geared manual actuators for two reasons:

- The torque required for actuation is too great for direct operation.
- Valves equipped with geared-type operators close slowly and thereby reduce the potential for damaging surge pressures.

Gate valves up to 300 mm (12 in.) in diameter can be actuated manually by handwheels acting directly through threaded nuts bearing on threads cut into the valve stem. Larger, manually operated gate valves should be fitted with gear reducers to reduce the force required to move the valve disc or plug to within reasonable limits. This accommodation is always provided at the expense of the time required to operate the valve. Thus, powered actuators are recommended for valves larger than 450 mm (18 in.) in diameter.

Manually actuated valves installed more than 2.1 m (6 ft, 9 in.) above the operating floor should be equipped with chain actuators to permit operation without the need for a ladder, which is both inconvenient and dangerous. Small [100 mm (4 in.) in diameter and less] plug and butterfly valves can be equipped with extension arms and chains. Larger valves should be equipped with chain-wheel actuators. This type of actuator can be obtained with a hammer-blow feature to break loose valves that are hard to start.

The following are miscellaneous but important notes on specifying valves and designing installations:

- To avoid safety problems, valve stems and actuators should not protrude into walkways.
- If a valve or gate is below floor level and needs frequent operation, a floor stand is appropriate. A less expensive square nut, to be turned by a wrench, is sometimes a suitable substitute. Nuts should be standardized and, in the United States, the dimensions are given in AWWA standards.
- Ordinary chain wheels should not be used in a wet well because of the possibility of sparking. Nonsparking chain-wheel designs are available.
- Specifications and purchase orders for valves should state the direction of opening. The usual standard is for valve stems to turn counterclockwise to open. However, some valves open

clockwise; if these must be used, paint handwheels, nuts, and levers red and/or plainly mark the directions for operating them.

- Valves are made with both rising and nonrising stems. The rising style is advantageous for indicating the valve opening. Geared actuators should incorporate position indicator dials.
- In some valves (especially butterfly valves), the flow through the valve tends to move the disc and may cause flutter. Such valves require locks. Worm gearing can be designed to be self-locking.
- Butterfly, plug, and ball valves 200 mm (8 in.) and larger in nominal diameter should be equipped with some type of actuator employing a mechanical advantage. Some engineers and pumping station workers prefer mechanical actuators for 150-mm (6-in.) valves as well. Worm gearing is the best because it is usually manufactured with greater precision than other types of actuators. Pantograph and traveling sleeve type actuators are often troublesome.

Powered Actuators

Powered actuators can operate directly on the valve shaft or stem or through gear reducers and special drive linkages. For rotary motion valves, such as ball or cone designs intended for surge control service and some butterfly valve actuating mechanisms, these special linkages serve two purposes: (1) conversion of linear motion to rotary motion, and (2) special closing and opening characteristics to control pressure transients on pump start-up, shut-down, and power failure.

Electric Actuators

Electric actuators generally consist of electric motors driving through a gear train to power the valve stem or shaft. In general, the speed of operation and differential pressure at specified conditions determine motor power requirements. Motor operators have a hammer-blow feature to start hard-to-open valves (e.g., gate, nonlubricated plug). As with other actuators, a hammer-blow feature is important to start the valve in operation in either direction. Usually, this type of actuator is equipped with a handwheel for manual operation should the motor be disabled. It is important to specify a declutching mechanism that disengages the motor from the power train whenever the handwheel is being used—it can be motor preference or handwheel preference. Electric motor actuators can be specified, however, to accept remote commands, to telemeter position to remote locations, and to function with remote reversing starters. Electronic, modulating positioners are available, but these rarely are used in pumping station designs. To provide safeguards against potential damage, specify (1) torque-limiting switches for both open and closed positions, and (2) four train limit switches to position the valve for seating. Specify integral, independent safety overrides.

Direct current power with battery support is recommended for all system control and monitoring functions where the actuating system must function during power failure. Batteries should be constantly trickle charged at low input with automatic switching to fast charging if the battery charge is low.

Hydraulic Actuators

Hydraulic actuators use fluid under pressure as a source of power, and both linear and rotary actuators are available. Hydraulic actuators (fluid power actuators) can be designed to use either oil under pressure from a self-contained system or water from the local potable supply, wherein the water is usually run to waste. However, because potable water supply systems must be designed to resist corrosion, specify stainless steel, bronze, or chrome-plated construction. Hydraulic actuators should be selected to provide sufficient power to break the valve loose. Once the valve is in motion, depending on the actuator linkage, a lower pressure differential may be required to move it from one position to another. One of the advantages of fluid power actuators is that fluid can be stored in pressure-charged accumulators or hydropneumatic tanks to provide a source of power under emergency conditions, such as commercial power failures. Another advantage is the ease of changing the speed of opening or closing the valve. Pressure should not exceed 14,000 kPa (2000 $lb/in.^2$) in all fluid power systems to limit leaks and joint failures, and a limit of 75% of that pressure is better. Specify premium components for fluid power systems. If operation of the equipment during emergencies is a prime concern, retain a specialist to design the system.

Self-contained hydraulic power packs, such as those manufactured by Trident and Rodney Hunt Co., should be considered as a reliable and lower cost means of providing emergency and normal operation for critical valves and gates. The power packs can be mounted on the valve body and include a complete hydraulic power system: reservoir, pump, actuator, uninterruptible power supply, accumulator,

and all controls in a hermetically sealed, corrosion-resistant and submersible or explosion-proof enclosure.

Pneumatic Actuators

Pneumatic actuators are available for both linear and rotary motions. The disadvantages of pneumatic operators include (1) noise; (2) poor operating characteristics because the powering fluid, a gas, expands on change of pressure; (3) a tendency to freeze because of expansion on release to atmospheric pressure; and (4) corrosion (with compressed air systems) because of water entrained in the gas.

A pneumatic actuator system generally has a lower initial installed cost than a motorized actuator system. However, the maintenance costs for the pneumatic actuators and associated equipment (compressors, receivers, traps, separators, filters, and piping) are usually much higher than they are for a motorized actuator system.

Pneumatic actuator systems are especially attractive for pumping stations because they can actuate valves when a power failure occurs. A receiver (tank) provides the compressed air to operate the actuator. A solenoid valve, energized to close (deenergized to open), is placed in the air line connecting the receiver to the pneumatically actuated valve. Upon power failure, the solenoid valve opens and the pneumatic actuator causes the valve to close. This system allows some control over the time of closure of the valve so that excessive surge pressures can be avoided. Size the receiver to hold twice as much air as needed to operate all of the valves through one cycle.

In most pumping stations requiring only a few powered valves (no more than three or four), an electric actuator system generally has the lowest installed cost. However, electric actuators are not usually considered fail-safe devices. Hydraulic systems are usually the most expensive, with pneumatic systems in the middle. The cost of the hydraulic and pneumatic actuators themselves may be cheaper than the electric actuators, but the cost of the necessary auxiliary equipment—such as receivers, compressors, dryers, filters, and relief valves—rapidly increases the cost of small pneumatic systems. However, self-contained actuators that use the pumped water for power (so that auxiliary equipment is not required) are relatively inexpensive and low in maintenance labor.

Similarly, electric actuators require less maintenance than pneumatic and hydraulic actuators. Again, it is the maintenance associated with the auxiliary equipment that usually causes electric systems to be selected.

5-7. Air and Vacuum Valves

Air release and vacuum relief valves are often needed along transmission mains and may sometimes be unavoidable in wastewater force mains. Air must be expelled when the pipeline is being filled. During operation, air (or sewer gas) accumulates along flat pipelines and especially at high points, and must be bled slowly to prevent (1) "air binding" due to the reduction of the cross section of the pipe at high points; and (2) corrosion at the soffit of the pipe. Vacuum conditions must be prevented when the pump head drops quickly (as in power failures) to prevent the shock of colliding water masses in column separation. Vacuum relief valve openings must be large—as much as one-sixth of the diameter of the transmission main, whereas air release valves must be small—as small as one-fiftieth of the diameter of the pipe. Although such valves are outside of the pumping station, their presence in the transmission main has a profound effect on surge and, hence, on the whole system.

A pipeline designed for fluid velocities high enough to scour air to the exit is an alternative approach that does not require the use of air release valves. Such velocities are within the normal design range for pipes 300 to 375 mm (12 to 15 in.) in diameter or smaller, as shown in Table B-9. Elimination of air and vacuum valves in favor of high velocity is not objectionable and may be of some benefit (by eliminating air bubbles altogether) *but only if there is assurance that catastrophic failure is precluded by air-scouring velocities* in pipes on flat or downward slopes. Excessive headloss can be prevented by the use of larger pipe for upward slopes. Note, however, that air-scouring velocities must be reached frequently enough to prevent large air bubbles from forming. Also, note that large pockets of air may greatly increase the head on the pumps. Design such combination systems on the assumption that the air release valves will sometimes fail to operate.

Air Release Valves

Air release valves slowly release the pockets of air that accumulate at high points in piping systems. In pumping stations, they are recommended on the discharge of vertical turbine pumps, especially when

pumping from wells and sumps. A conventional valve has a float that drops to vent the air that accumulates in the body. Valves smaller than 19 mm ($^3/_4$ in.) usually have a float-activated compound lever with a linkage mechanism to provide a tight closure.

The valve body contains an orifice, usually 5 mm ($^3/_{16}$ in.) or smaller, through which the air escapes.

Combination Air Valves

A combination valve combines the functions of air release and vacuum relief. It allows the use of one valve and one connection to the piping instead of two connections. Conventional valves contain linkage mechanisms for float assemblies as shown in Figure 5-16. The large venting orifice exhausts large quantities of air from a pipeline being filled and admits large quantities of air into a pipeline being drained. Some valves include a perforated water diffuser on the inlet to prevent the water column from rapidly entering the valve and slamming the float shut, thereby possibly causing a severe water hammer problem.

Air and Vacuum Valves in Water Service

Wherever possible, select a profile that minimizes the number of air valves because they constitute an onerous maintenance problem. In water service, the short valve body (Figure 5-16) is appropriate for combination air valves. Lescovich [15] has discussed the use of air valves in transmission mains.

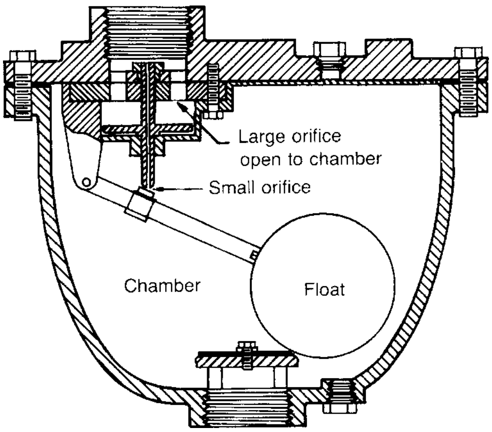


Figure 5-16. Conventional combination air valve for water service. After APCO Valve & Primer Corp.

Air and Vacuum Valves in Wastewater Service

Historically, conventional air release and vacuum relief valves for wastewater service have been expensive to maintain and are prone to jamming either open or closed due to grease and debris. In side-by-side tests in two systems [16, 17] known for considerable difficulty with conventional combination air-vacuum valves, a new design, Vent-O-Mat® series RGX [18], was found to be both reliable and to require little maintenance. The period between cleanings of conventional air-vacuum valves is typically one to three months, whereas the permissible period between cleanings of the RGX valve was found to be at least six times longer. In one of the tests, a Vent-O-Mat® series RGX valve was still functional after 18 months of service without cleaning. A somewhat different style, Series RBX, is intended for water service. Product information comes with software to help engineers select the proper size valve for a specific application.

The operation of the valve is illustrated in Figure 5-17. When a pipeline is being filled, the floats are at the bottom of the cylinder, and (as shown in Figure 5-17a) air is exhausted freely through passageways that are unchanging in cross-sectional area. If the column of water moves rapidly, however, the higher air velocity lifts the anti-surge float and reduces the passageway to a small orifice (Figure 5-17b) that releases air slowly and keeps the surge pressure to the level shown in Figure 5-17c. Conventional combination air-vacuum valves do not have such features. Small amounts of air or gas reduce the buoyancy of the lower float and allow the nozzle to separate slightly from the nozzle seat (Figure 5-17d) and release small quantities of air or gas. When a pump stops and part of the pipeline empties, or if column separation occurs, the floats fall and air rapidly enters the pipeline (Figure 5-17e).

Regardless of the proven reliability of air release and vacuum relief valves (or combination valves), if the equipment is to be used to protect pipelines, prudence dictates that two valves, connected to the pipeline with a trans-flow-type three-way valve (that can block either air valve but not both) be installed for the added insurance that the surge control system will function 100% of the time. The valves must be installed in a flood-protected location to assure that air, not water, will enter the valve when a vacuum occurs. See Figure 7-2 in Section 7-1 for a proper installation schematic.

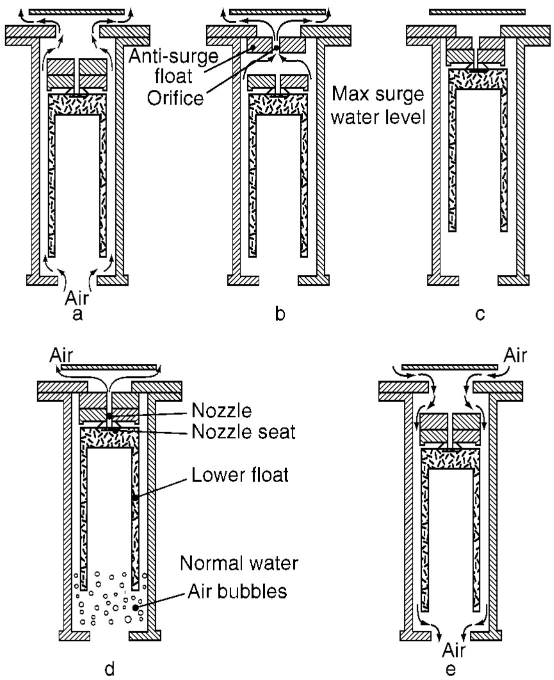


Figure 5-17. Vent-O-Mat® combination air release and vacuum relief valve, Series RGX. (a) Pipeline filling at subcritical wastewater approach velocity; (b) pipeline filling at supercritical wastewater approach velocity; (c) pipeline fully charged; (d) bleeding small amounts of air under pressure; (e) vacuum relief. Adapted from Vent-O-Mat® literature.

5-8. Materials of Construction

Bodies

Most valves in water, wastewater, and sludge pumping stations are not exposed to severely corrosive conditions. Bodies are usually cast iron (ASTM A48 or A126), cast steel (ASTM A216), or ductile iron (ASTM A395 or A536) for valves 100 mm (4 in.) or larger, and bronze (ASTM B62 or B584, for which several alloys are available) for valves 75 mm (3 in.) and smaller. Fabricated steel (ASTM A36 or A516) is sometimes used in valves larger than 1800 mm (72 in.), especially in butterfly valves. In many locales, however, water and wastewater are indeed corrosive to iron and steel. For such liquids, the iron and steel bodies should be lined with epoxy (such as a product complying with AWWA C550) or other products. Also, some waters attack bronzes that contain high percentages of zinc and cause dezincification [19]. There is no universal agreement on an acceptable level of zinc in bronzes and other copper alloys. For example, some valve standards or specifications allow bronzes with zinc contents of up to 16%, but many engineers believe that is too high and allow no more than 5 to 7% zinc [20]. Some copper alloys (bronzes) frequently used in valves and having zinc contents of no more than 7% are alloys C83600, C87600, C90500, and C93700, defined in ASTM B584 and B763.

A common phrase encountered in specifying valves is "iron body, bronze mounted (IBBM)." There is no universally accepted definition of this phrase. Some manufacturers provide a bronze disc and seat, while others provide only a bronze seat. Others furnish a bronze disc and seat up to a certain size, and then provide only a bronze seat in larger sizes. Be careful to define exactly what is meant when using a phrase such as "iron body, bronze mounted."

Seats

Seats in isolation or control valves are more subject to erosion and corrosion than bodies because the fluid velocity impinges most noticeably there.

General

Bronze-on-bronze is the cheapest, and seats in metal valves 75 mm (3 in.) and smaller are frequently bronze (note the previous discussion of the zinc content in bronzes for valves used for some waters). Most seats or seat retention devices in valves 100 mm (4 in.) and larger are some grade of stainless steel. Stellite facing is much harder and more erosion resistant than stainless steel, but it is also much more expensive. Do not specify an exotic material unless there is a clear need for it. The most common materials for resilient seats are listed in Table 5-1, but there are numerous plastics available with special qualities. Buna N can be attacked by industrial chemicals, but unless there is an excess of illegal dumping of such solvents, it is a nearly ideal seat material. Teflon™ is more resistant to attack, but it creeps.

Teflon™

Teflon™ (polytetrafluoroethylene, or PTFE) is suitable for both water and wastewater. It is hard, strong, and impervious to attack by nearly all chemicals, but it has shortcomings such as cold flow and creep. Other materials are usually more suitable.

Elastomers

Elastomers (natural or gum rubber and the synthetic rubbers, such as neoprene, Viton, and Buna N) are used for resilient seats, O-rings, and a few other parts. Natural rubber is suitable for clean water, but wastewater contains oil and grease products and organic solvents that attack it. The synthetic rubbers usually give good service in both water and wastewater systems. Elastomers are subject to wear in grit slurry service and where they rub against iron tubercles, hard scale deposits, or corroded surfaces. They are, however, suitable if the grit has been removed.

Engineers should be aware that rubber compounds—both natural and synthetic—are not uniformly and consistently resistant to some disinfecting agents commonly used in the water works industry, such as chloramines. There appear to be wide variations in resistance to attack on various elastomers and on differing formulations of the same elastomer. It should also be noted that various standards pertaining to valves (such as AWWA C500, C504, and C509) do not address the issue of resistance of the rubber seating material to either chloramines or free chlorine. The resistance to attack by disinfecting agents is addressed by Reiber [21]. As of 2004, the issue of developing a testing standard to specify the resistance of elastomers to disinfecting agents was being discussed and investigated by several AWWA standards committees.

Packing

Valve packing is as important in a valve as the packing gland is in a pump. It prevents leakage past the valve stem and damage to the valve housing. The seal is normally made by placing packing material around the valve stem and compressing it with a follower or gland, which is tightened by a packing nut. Although asbestos has historically been used as a packing material, more and more valve manufacturers have discontinued its use and have switched to nonasbestos materials such as Teflon™, aramid fiber, acrylic fiber bound with nitrile, and Buna N. Note that some common standards specify packing material that is no longer even made. For example, AWWA C500 for gate valves specifies flax conforming to Federal Specification HH-P-106d. However, the flax specification HH-P-106d was discontinued in 1978 by the federal government because of insufficient usage.

Stems

Gate, globe, and needle valves have stems that rise in the body to seat or unseat. In both water and wastewater service, these stems are usually bronze, and the problem of dezincification is especially acute (see the discussion on zinc content in “Bodies” of this section).

5-9. Installation of Valves

Warping of the valve body due to pressure and thermal stresses in the connecting pipelines or lack of proper valve support can damage the valve enough to prevent it from functioning. The valve body should not be supported by the adjacent piping, nor should it support the piping. The following are some suggestions for valve support:

- For piping and valves supported on floors, provide separate bases or supports for valves 100 mm (4 in.) or larger.
- For piping and valves suspended from ceilings, provide separate hangers at valves—one hanger or support at each end of the valve body or on the connecting pipe within one pipe diameter of the valve end.
- Provide enough flexibility in connected piping so that thermal strains in the piping do not stress the valve.
- Install piping without springing, forcing, or stressing the pipe or any connecting valves.
- Some valves (such as AWWA C507 ball valves) are intended to be supported in a certain manner. Be sure to read the relevant standard before designing the support system.
- Always consult the valve manufacturer about proper support.

Buried valves with flanged or other rigid end connections should be installed with adjacent flexible couplings or other means to accommodate issues such as differential settlement, thermal expansion and contraction of the adjacent piping, capability to remove the valve for maintenance, access to the valve interior for inspection, support of the valve, and controlling the shear loading on both the valve flanges and the adjacent pipe flanges.

End Connections

End connections for valves can be

- screwed (ANSI B1.20.1);

- flanged (ANSI B16.1 for cast-iron valves, ANSI B16.5 for steel valves);
- grooved end (AWWA C606);
- butt welded (ANSI B16.34); or
- socket welded (most commonly found in plastic valves).

In general, valves 75 mm (3 in.) and smaller should have screwed ends, whereas larger valves have flanges or grooved ends because, in larger sizes, assembling pipes or valves with threaded ends becomes very laborious. Furthermore, bolted connections are much easier to disassemble than screwed connections, even if unions are installed at moderate spacing.

Butterfly, gate, and eccentric plug valves with grooved-end or flanged connections are probably the most commonly encountered valves in a pumping station. Butterfly valves must be of long body (not short body) style per AWWA C504 to accommodate grooved ends. The use of grooved-end connections provides greater ease than do flanges for removing valves from a piping manifold. Sleeve (Dresser®) couplings need not be provided adjacent to grooved-end valves, although such couplings may be needed for other reasons, such as thermal expansion, alignment, and differential settlement.

Locate valves and design the surrounding piping to prevent clogging with grit. Except for clean water service, bonnets must be within 45 degrees of upright to keep out grit that can build up and prevent operation. If possible, avoid installing valves on risers, especially if there is a long section above the valve. If a valve must be placed on a riser, use a ball or plug valve that is wiped clean by operation and has no crevices to collect grit. Design risers to enter a header horizontally from an elbow to the tee so that grit and solids cannot block the riser. Even if valves are installed on horizontal pipes, locate them at least three (or, better, five) pipe diameters from the riser. Even in clean water service, butterfly valves should be at least two (three is better) pipe diameters from an elbow so that streamlines, entering from an angle, do not make the valve difficult to open or close and do not cause the vane to flutter. If the valve must be close to an elbow, orient the valve so the vane is not subjected to dynamic loading from the flow through the elbow.

5-10. Corrosion Protection

When used in water lines, sewers, sludge piping, or any other service in which the fluid is particularly corrosive, metal valves can be lined with epoxy. This epoxy lining can only be applied to valves 100 mm (4 in.) and larger.

In areas of corrosive soil, buried valves should be coated with coal tar, coal-tar epoxy, or a high-solid (usually 70% solids by volume or higher) epoxy.

Pay particular attention to the need for stainless-steel bolts in buried and submerged valves and valves exposed to H_2S atmospheres. In some soils and waters, galvanized steel bolts and nuts (ASTM A307) corrode readily. Therefore, type 304 or 316 stainless bolts (ASTM A193, Grade B8 or B8M) and nuts (ASTM A194, Grade 8 or 8M) may be required.

Low-solids and low-pH waters are very corrosive to brass [19], iron, and, to some degree, cement liners. Such waters can be rendered benign by chemical treatment [22, 23].

Seats must neither corrode nor erode. The corrosion and erosion resistance of several seat materials are compared in Table 5-1 (see also Lyons [6]).

5-11. References

1. Cook, D. T., "Selecting hand-operated valves for process plants," *Chemical Engineering,* 89, 126–140 (1982).
2. O'Keefe, W., "Pump controls and valves," in *Pump Handbook,* I. J. Karassik, W. C. Krutzsch, W. H. Fraser, and J. P. Messina, Eds., McGraw-Hill, New York (1976).
3. Deutsch, D. J., et al., *Process Piping Systems*, McGraw-Hill, New York (1980), pp. 193–357.
4. AWWA M11, *Steel Pipe—A Guide for Design and Installation,* 2nd ed., American Water Works Association, Denver, CO (1985).
5. Hutchison, I. W., Ed., *ISA Handbook of Control Valves,* 2nd ed., Instrument Society of America, Research Triangle Park, NC (1976).
6. Lyons, J. L., *The Valve Designer's Handbook,* Van Nostrand Reinhold, New York (1982).
7. Lyons, J. L., and C. L. Askland, Jr., *Lyon's Encyclopedia of Valves,* Van Nostrand Reinhold, New York (1975).
8. British Valve Manufacturer's Association, *Technical Reference Book on Valves for the Control of Fluids,* 2nd ed., Pergamon, London (1966).
9. Schweitzer, P. A., *Handbook of Valves,* Industrial Press, New York (1972).
10. Pearson, G. H., *Applications of Valves and Fittings,* Pitman, London (1968).
11. Zappe, R. W., *Valve Selection Handbook,* Gulf Publishing, Houston, TX (1981).
12. AWWA Manual M49, *Butterfly Valves: Torque, Head Loss, and Cavitation Analysis*, 1st ed., American Water Works Association, Denver, CO (2001).

13. Parmakian, J., "Water hammer," Section 9.4 in *Pump Handbook,* I. J. Karassik, W. C. Krutzsch, W. H. Fraser, and J. P. Messina, Eds., McGraw-Hill, New York (1976).
14. Collier, S. L., et al., "Behavior and wear of check valves," *Journal of Energy Resources Management, Transactions of the American Society of Mechanical Engineers,* 105, 58 (March 1983).
15. Lescovich, J. E., "Locating and sizing valves in transmission mains," *Journal of the American Water Works Association,* 64, 457–461 (July 1972).
16. Wichita, Kansas.
17. Salina, Kansas.
18. Vent-O-Mat®. International Valve Marketing, Inc., P.O. Box 56, Plainfield, IL 60544. Email limeym@internationalvalve.com or fivev@mindspring.com.
19. Jester, T. C., "Dezincification update," *Journal of the American Water Works Association,* 77, 67–69 (October 1985).
20. "Valve dezincification prevented," *Heating/Piping/Air Conditioning,* 53(11), 30 (November 1981).
21. Reiber, S. H., *Chloramine Effects on Distribution System Materials*, American Water Works Association, Denver, CO (1993).
22. Singley, J. E., et al., *Corrosion Prevention and Control in Water Treatment and Supply Systems,* Noyes Data, Park Ridge, NJ (1985).
23. Merrill, D. T., and R. L. Sanks, "Corrosion prevention by the deposition of $CaCO_3$ films," *Journal of the American Water Works Association,* Part 1, 69, 592–599 (November 1977); Part 2, 69, 634–643 (December 1977); Part 3, 70, 12–18 (January 1978).

Chapter 6

Fundamentals of Hydraulic Transients

BAYARD E. BOSSERMAN II
WILLIAM A. HUNT

CONTRIBUTORS
Robert C. Glover
Joseph R. Kroon
M. Steve Merrill
Gary Z. Watters

The purpose of this chapter is to provide an overview of the problems caused by hydraulic transients and an insight into the circumstances that make a more thorough analysis necessary. The fundamental theory of hydraulic transient analysis is described simply, with no attempt to present rigorous mathematical or analytical methods. Simple numerical examples include surge pressure calculations, attenuation of surge pressure by programmed valve closure, and the design of pipe to resist upsurge and downsurge pressures.

For more complete discussions of hydraulic transients, see Parmakian [1], Rich [2], Wylie and Streeter [3], Watters [4], and Chaudhry [5].

6-1. Introduction

Hydraulic transients are the time-varying phenomena that follow when the equilibrium of steady flow in a system is disturbed by a change of flow that occurs over a relatively short time period. Transients are important in hydraulic systems because they can cause (1) rupture of pipe and pump casings; (2) pipe collapse; (3) vibration; (4) excessive pipe displacements, pipe-fitting, and support deformation and/or failure; and (5) vapor cavity formation (cavitation, column separation).

Some of the primary causes (and frequency of occurrence) of transients are (1) valve movements—closure or opening (often), (2) flow demand changes (rarely), (3) controlled pump shutdown (rarely), (4) pump failure (often), (5) pump start-up (rarely), (6) air venting from lines (often), (7) failure of flow or pressure regulators (rarely), and (8) pipe rupture (rarely).

The identification and calculation of pressures, velocities, and other abnormal behavior resulting from hydraulic transients make possible the effective use of various control strategies, such as the

- selection of pipes and fittings to withstand the anticipated pressures;
- selection and location of the proper control devices to alleviate the adverse effects of transients; and
- identification of proper start-up, operation, and shutdown procedures for the system.

The analysis of unsteady flow in pipe systems is generally divided into two major categories.

Rigid water column theory (surge theory). The fluid and pipe are inelastic, and pressure changes propagate instantaneously. These flow conditions are described by ordinary differential equations.

- *Solution:* closed-form integration or finite difference numerical integration.

- *Advantages:* the analysis can be applied by a person with little numerical analysis skill and with limited computational facilities.
- *Disadvantages:* the solutions, which are always approximations, are applicable only to simple pipelines. Considerable experience is required to know whether the results are applicable.

Elastic theory (water hammer). The elasticity of both fluid and pipe affect the pressure changes. Pressure changes propagate with wave speed, a, which varies from about 300 to 1400 m/s (1000 to 4700 ft/s). Flow conditions are described by nonlinear partial differential equations.

- *Solution:* arithmetic, graphical method or method of characteristics using finite difference techniques.
- *Advantages:* the theory accurately represents system behavior and is, therefore, applicable to a wide range of problems. Pipe friction, minor losses, and varying valve closure procedures can be incorporated.
- *Disadvantages:* applying the theory requires a substantial initial effort on the part of the user to learn it, a digital computer programmed for the method of characteristics, and a knowledge of the operational characteristics of system components to set up the solution for the computer.

6-2. Nomenclature

In Chapters 6 and 7, "velocity" always means velocity of water and "speed" means the velocity of pressure waves. The symbols used in Chapters 6 and 7 are defined as follows.

a Elastic wave speed in water contained in a pipe [in meters per second (feet per second)]
C Coefficient whose value depends on pipe restraint
D Inside diameter of a pipe [in meters (inches or feet)]
e Wall thickness of a pipe [in meters (inches or feet)]
E_j Longitudinal joint efficiency in welded pipes (dimensionless)
E Modulus of elasticity of pipe material [in newtons per square meter (pounds per square inch or pounds per square foot); see Table 6-1].
g Acceleration due to gravity [in meters per second squared (feet per second squared)]
h Head due only to surge [in meters (feet)]
Δh Change of head due to surge [in meters (feet)]
h_a Allowable head due to surge [in meters (feet)]
K Bulk modulus of elasticity of the liquid [in newtons per square meter (pounds per square inch or pounds per square foot); see Table A-8 or A-9]
L Length of pipeline [in meters (feet)]
ΔP Change of pressure due to surge [in newtons per square meter (pounds per square inch)]
ΔP_a Allowable pressure change due to surge [in newtons per square meter (pounds per square inch)]
P_{atm} Atmospheric pressure [in newtons per square meter (pounds per square inch); see Table A-6 or A-7]
ΔP_c Difference between external and internal pressure on a pipe [in newtons per square meter (pounds per square inch)]
P_v Vapor pressure of water [in newtons per square meters (pounds per square inch); see Table A-8 or A-9]
SF Safety factor (dimensionless)
s_y Yield stress [in newtons per square meter (pounds per square inch)]
t Time (in seconds)
t_c Critical time ($2L/a$; in seconds)
V Volume [in cubic meters (cubic feet)]
v Velocity [in meters per second (feet per second)]; in this chapter, "velocity" is average velocity of fluid flow
Δv Change in velocity [in meters per second (feet per second)]
Δv_a Allowable change in velocity [in meters per second (feet per second)]
ρ Density [in kilograms per cubic meter (slugs per foot; see Table A-8 or A-9)]
μ Poisson's ratio (dimensionless; see Table 6-1)
γ Specific weight of water [in newtons per cubic meter (pounds per cubic feet; see Table A-8 or A-9)]

6-3. Methods of Analysis

Methods of analyzing pipelines for the effects of hydraulic transients, with or without various means of controlling them or reducing the severity, can be summarized as follows:

- Graphical [1]
- Arithmetic [2]
- Algebraic [3]
- Method of characteristics [3,4,5]
- Finite element
- Implicit differentiation

These are methods of *analysis*, not methods of *design*. In analysis, the system is described mathematically, and the behavior of the system is predicted by the analysis. In design, the desired physical results are

Table 6-1. Physical Properties of Pipe Materials

		Modulus of elasticity		
			U.S. customary units	
Material	Poisson's ratio	SI units (N/m^2)	lb/in.^{2}	lb/ft^2
Aluminum	0.33	7.30 E + 10	1.05 E + 7	1.51 E + 9
Asbestos-cement	0.30	2.30 E + 10	3.40 E + 6	4.90 E + 8
Brass	0.34	1.03 E + 11	1.50 E + 7	2.16 E + 9
Copper	0.30	1.10 E + 11	1.60 E + 7	2.30 E + 9
Ductile iron	0.28	1.66 E + 11	2.40 E + 7	3.46 E + 9
Gray cast iron	0.28	1.03 E + 11	1.50 E + 7	2.16 E + 9
HDPE	0.45	1.0 E + 9[a]	1.5 E + 5[a]	2.2 E + 7[a]
PVC	0.45	2.70 E + 9	4.00 E + 5	5.76 E + 7
Steel	0.30	2.07 E + 11	3.00 E + 7	4.32 E + 0
Concrete	—	$4.73 \times 10^6 \sqrt{f'c}$[b]	$57{,}000\sqrt{f'c}$[c]	

[a]At 16°C (60°F). Increases greatly with decreasing temperature and vice versa.

[b]$f'c$ is ultimate strength in newtons per square meter.

[c]$f'c$ is ultimate strength in pounds per square inch.

described and alternative means for attaining these results are compared, which leads to a selection of one or more control measures. Analysis should be performed as a part of the design process.

All of the above methods involve the equations of motion and continuity used to describe the velocity and pressure variation in the pipeline. For computer modeling, the most widely used is the method of characteristics, in which the partial differential equations of motion and continuity are converted into four first-order equations represented in finite difference form and solved simultaneously with a computer. The method provides for

- the inclusion of many possible pipeline or system features, such as junctions, pumping stations, air chambers, air release valves, reservoirs, and line valves;
- the inclusion of fluid friction;
- the retention of small or secondary terms in the original equations so that accuracy is retained; and
- the computation of pressure and velocity as a function of time at various points throughout the entire pipeline system.

6-4. Surge Concepts in Frictionless Flow

Water hammer can occur in a pipeline flowing full when the flow is increased or decreased, such as when the setting on a valve in the line is changed. When a valve in a pipeline is closed rapidly, the pressure on the upstream side of the valve increases, and the pulse of increased pressure travels upstream at the elastic wave speed, a. This pulse (called an "upsurge") decreases the velocity of flow. Downstream from an in-line valve, the pressure is reduced and the wave of decreased pressure travels downstream, also at the elastic wave speed, a. This pulse (called a "downsurge") decreases the velocity of flow. If the velocity is reduced too rapidly and the steady-state pressure is low enough, the downstream pressure can be reduced to vapor pressure, which creates a vapor pocket. A large vapor pocket (called "column separation") can collapse with a dangerous explosive force produced by the impact between solid water columns and can cause the pipe to burst. This phenomenon can also occur upstream of the valve when the reflected positive wave returns to the valve.

The events following a sudden closure of a valve located a distance (L) downstream from a reservoir is described in Figure 6-1. Friction is neglected, and the energy gradeline (EGL) and hydraulic gradeline (HGL) are assumed to coincide because velocity heads are small compared with water hammer pressure heads. The steady-state EGL is called H, and the added-pressure head pulse is called h. The velocity of the fluid under steady-state conditions is v_o just before the valve is closed (at $t = 0$).

The sequence of events between the valve and the reservoir occurs in a four-phase cycle; the duration of each phase is the time for the pressure wave to travel between the valve and the reservoir (the length of the pipeline divided by the elastic wave speed, L/a). The sequence occurs as follows:

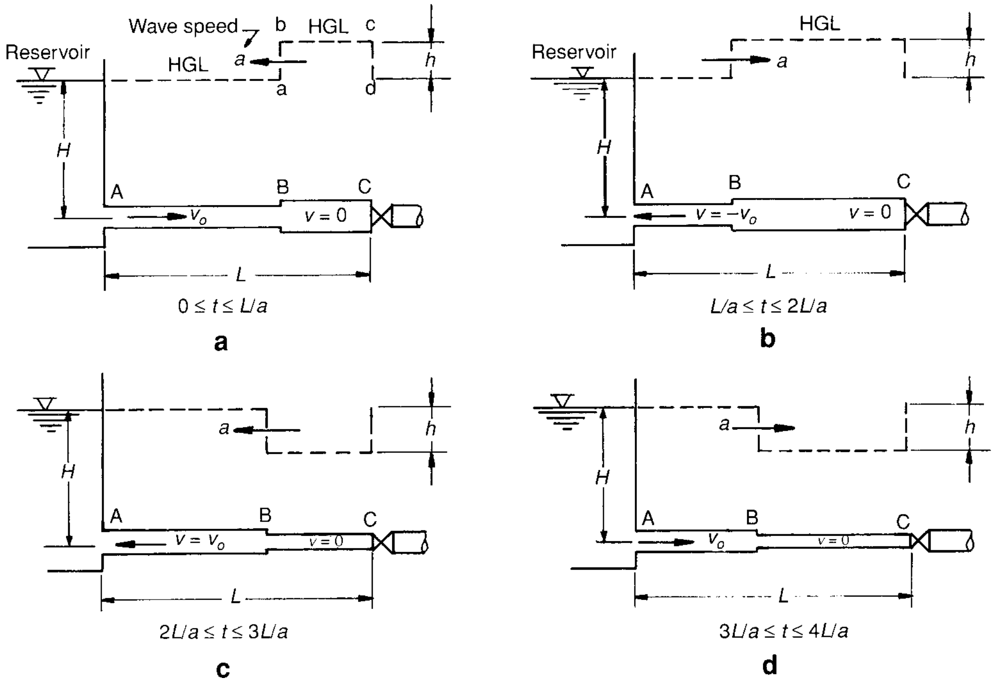


Figure 6-1. Sequence of events for one cycle after sudden valve closure.

1. $0 \le t \le L/a$. At $t = 0$, the fluid just upstream from the valve is compressed and brought to rest. Part of the pipe (section BC) is expanded and stretched, as shown in Figure 6-1a. This process is repeated for each successive increment of fluid as the pressure wave travels upstream. The fluid upstream from the wave front continues to flow downstream until it is stopped by the advancing pressure wave front. When the pressure wave reaches the reservoir at $t = L/a$, the fluid (at rest in the pipe) is under a total pressure head $H + h$, or h greater than the static head in the reservoir.

2. $L/a \le t \le 2L/a$. The pressure head difference at $t = L/a$ at the reservoir causes the fluid to flow from the pipe back into the reservoir with a velocity $-v_o$. The pressure along AB is reduced to the original steady-state level, H, and a negative wave producing normal pressure propagates back to the valve, as shown in Figure 6-1b. At $t = 2L/a$, the pressure is normal (equal to H) along the pipe but the velocity throughout the pipe is negative; that is, water is flowing away from the valve.

3. $2L/a \le t \le 3L/a$. At $t = 2L/a$, there is no fluid available to maintain the upstream flow at the valve, and the normal pressure head, H, is reduced by h to bring the fluid in section BC to rest (Figure 6-1c). This wave of reduced pressure propagates back toward the reservoir, all fluid comes to rest, and the reduced pressure allows the fluid to expand and the pipe walls to contract. At $t = 3L/a$, the reduced pressure, $H - h$, exists all along the pipe, the velocity is zero throughout the pipe, and the static pressure head in the pipe is less than the pressure head in the reservoir.

4. $3L/a \le t \le 4L/a$. At $t = 3L/a$, the unbalanced pressure head at the reservoir causes the fluid to flow back into the pipe at a velocity $+v_o$. A wave of pressure at the original static pressure head level propagates downstream toward the valve, but in section BC, the pressure is still reduced to $H - h$ (Figure 6-1d). At $t = 4L/a$, the pressure throughout the pipe is normal (equal to H), and the velocity is the same as the original $+v_o$ prior to the valve closure. But when the velocity, v_o, reaches the valve, the four-phase cycle repeats and continues to repeat periodically every $4L/a$ time period.

The variation of pressure with time during a $4L/a$ interval is shown at several points along the pipeline in Figure 6-2. Friction in a real pipe system eventually dampens the increased water hammer pressure head, h.

The principal concepts indicated by the events following a sudden valve closure are as follows:

- The time L/a is a significant parameter for water hammer analysis.

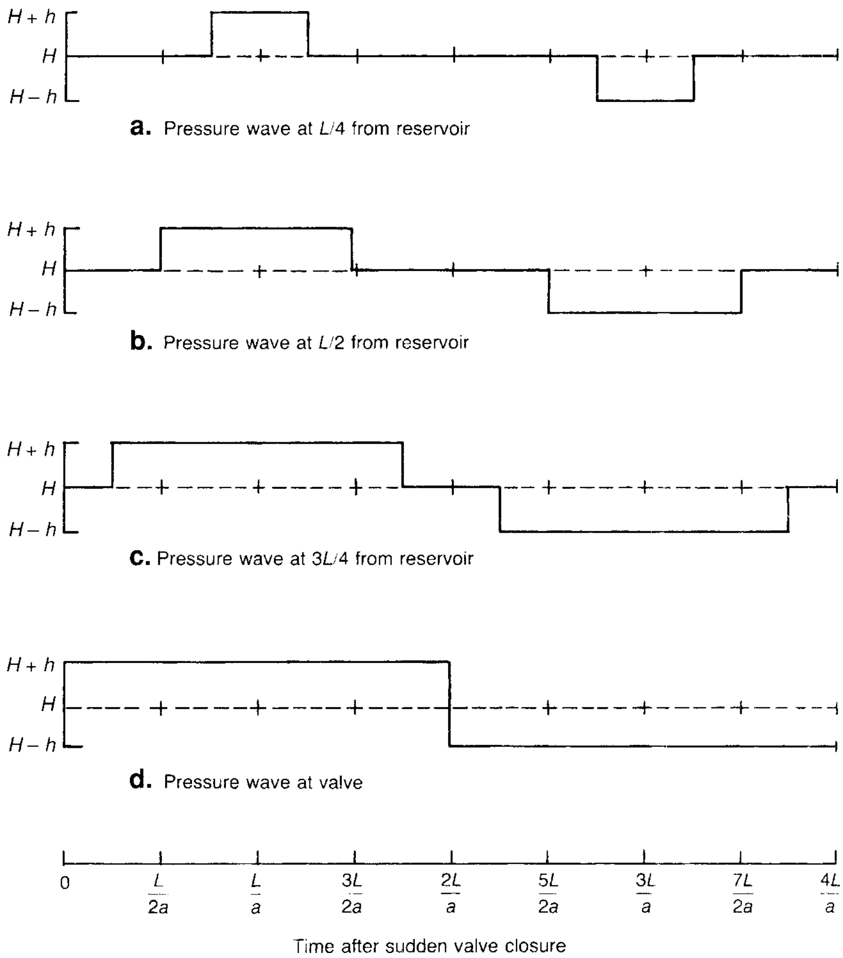


Figure 6-2. Head fluctuations at selected points in the pipeline of Figure 6-1 after valve closure.

- The time $2L/a$ is critical because the pressure head at the valve reaches a maximum at $2L/a$. A valve closed in any shorter time produces the same maximum pressure head rise at the valve, whereas the pressure rise is reduced if the valve is closed in a longer time interval. Hence,

$$t_c = \frac{2L}{a} \tag{6-1}$$

where t_c is the critical time, L is the length of the pipe, and a is the elastic wave speed.

Pressure Head Change

A momentum analysis of the flow conditions for the valve closure shows the pressure head change, Δh, is a function of the change in flow, Δv.

$$\Delta h = \frac{-a\Delta v}{g} \tag{6-2}$$

where Δh is the change in pressure head in meters (feet), a is elastic wave speed in meters per second (feet per second), Δv is the change in velocity (of water) caused by the event in meters per second (feet per second), and g is the acceleration due to gravity in meters per second squared (feet per second squared). Use the negative sign for waves traveling upstream, use the positive sign for waves traveling downstream, and note that $\Delta v = v_2 - v_1$, where v_1 is the velocity prior to the change in flow rate and v_2 is the velocity following the change. If the flow is suddenly stopped, $\Delta v = v_1$ and $\Delta h = av_1/g$. Note, too, that Δh is positive if Δv is negative. If the valve on the downstream end of a pipe is closed incrementally, Equation 6-2 becomes

$$\Sigma\Delta h = -\frac{a}{g}\Sigma\Delta v \text{ (for } t < t_c) \qquad (6\text{-}3)$$

Transient pressure heads due to valve closure can be reduced by slowly closing the valve over a time interval greater than t_c, as discussed in Section 6-5.

Elastic Wave Speed

The pressure head change due to a change in the flow rate requires the calculation of the elastic wave speed in the pipe. The wave speed depends on both the fluid properties and the pipe characteristics.

$$a = \sqrt{\frac{K/\rho}{1 + C\left(\frac{K}{E}\right)\left(\frac{D}{e}\right)}} \qquad (6\text{-}4)$$

where K is the bulk modulus of elasticity of the liquid in newtons per square meter (pounds per square foot), E is the modulus of elasticity of pipe material in newtons per square meter (pounds per square foot), D is inside pipe diameter in meters (feet), e is the pipe wall thickness in meters (feet), C is a correction factor for type of pipe restraint, and ρ is the density of the fluid in kilograms per cubic meter (slugs per cubic foot). The bulk modulus given in Table A-8 must be changed from kilopascals to newtons per square meter, and in Table A-9 it must be converted from pounds per square inch to pounds per square foot.

For thin-walled pipes (those with $D/e > 40$), the correction factor, C, varies according to pipe restraints. Values for three cases are

- $C_1 = 1.25 - \mu$ for pipes anchored at the upstream end only (Wylie and Streeter [3] use $1.0 - \mu/2$; the difference has little practical significance);
- $C_2 = 1.0 - \mu^2$ for pipes anchored against axial movement; and
- $C_3 = 1.0$ for pipes with expansion joints throughout.

The symbol μ is Poisson's ratio (see Table 6-1 for properties of pipe materials). Expressions for C_3 for thick-walled pipes ($D/e < 40$), which are more complex, are given by Wylie and Streeter [3]. Buried pipes are best represented by the factor C_2. Typical values for wave speed for water in pipes are given in Table 6-2. Because the elastic wave speed, a, for steel and ductile iron pipe is often close to 980 m/s (3200 ft/s), the change in pressure head from Equation 6-2 is, roughly, $\pm 100v$. For PVC, $h = \pm 34v$.

The speed of the pressure wave in a pipeline carrying liquids is greatly reduced if bubbles of free gas are entrained, as shown in Table 6-3. A detailed discussion of this phenomenon is given by Wylie and

Table 6-2. Typical Wave Velocities in Pipe for Water Containing Dissolved Air

	Wave velocities	
Pipe material	m/s	ft/s
Asbestos cement	820–1200	2700–3900
Copper	1000–1300	3400–4400
Ductile iron	980–1400	3200–4500
HDPE[a]	180–370	600–1200
PVC	300–600	1000–2000
Steel	600–1200	2000–4000

[a] For a modulus of elasticity of 1.03E + 9 N/m^2(150,000 lb/in.^{2}).

Table 6-3. Effect of Air Entrainment on Wave Speed[a]

	Wave speed	
V_{air}/V_{total}	m/s	ft/s
0	1200	4000
0.001	610	2000
0.002	460	1500
0.004	300	1000
0.008	210	700

[a]After Wylie and Streeter [3].

Example 6-1
Effect of Pipe on Wave Speed and Pressure

Problem: For 300-mm (12-in.) pipes of ductile iron, steel, and PVC, find the effects of pipe material, bedding, and joint conditions on wave speed and water hammer pressure. Assume the pipes are 3000 m (9840 ft) long and carry 0.15 m^3/s (5.3 ft^3/s) of water at a temperature of 15°C (59°F).
Solution: The wave speeds are based on Equation 6-4. The calculation of wave speed to the nearest 30 m/s (100 ft/s) is usually sufficient, because the accuracy of the data does not justify greater precision. The critical valve closure times, t_c, are based on Equation 6-1. Valve closure

in any time interval less than t_c subjects the pipe to the maximum pressure, Δh, given by Equation 6-2. Note that $K = 2.15 \times 10^9$ N/m^2(4.49×10^7 lb/ft^2) for water.

	SI Units			**U.S. Customary Units**		
	Ductile iron	Steel	PVC	Ductile iron	Steel	PVC
OD [m (ft)]	0.335	0.324	0.324	1.10	1.063	1.063
e [m (ft)]	0.0094	0.0048	0.0149	0.0308	0.0157	0.0259
E [N/m^2 (16/ft^2)]	1.66×10^{11}	2.70×10^{11}	2.70×10^9	3.46×10^9	4.32×10^9	5.76×10^7
μ	0.28	0.30	0.45	0.28	0.30	0.45
$C_1 = 1.25 - \mu$	0.97	0.95	0.80	0.97	0.95	0.80
$C_2 = 1 - \mu^2$	0.92	0.91	0.80	0.92	0.91	0.80
$C_3 = 1.0$	1.00	1.00	1.00	1.00	1.00	1.00
Wave speed, a [m/s (ft/s)], from Equation 6-4:						
$C_1 = 1.25 - \mu$, no air	1220	1150	280	4000	3770	930
$C_2 = 1 - \mu^2$, no air	1210	1130	280	3970	3710	930
$C_3 = 1.0$, no air	1190	1110	260	3920	3640	840
$C_3 = 1.0$, 0.2% air	490	490	230	1620	1600	760
For $C_3 = 1.0$ and no air:						
$t_c = 2L/a$ (in s)	5.0	5.4	23	5.0	5.4	23
Flow [m/s (ft/s)]	1.8	1.8	1.9	5.9	6.0	6.2
h [in m (ft)]	219	206	49.4	718	676	162
P [in kPa (lb/in.^{2})]	2150	2020	480	310	290	70

Streeter [3]. There seems to be no way of predicting air volume entrainment. Neglecting the effect of air entrainment provides a conservative analysis. However, as the wave speed decreases, L/a increases and the time for closure of valves must be increased.

6-5. Slow Closure of Valves

To limit the pressure rise, the maximum deceleration of the water during the critical time period, t_c, must be limited. The maximum allowable deceleration can be calculated by using the ratio

$$\frac{\Delta P_a}{\Delta P} = \frac{\Delta h_a}{\Delta h} = \frac{\Delta v_a}{\Delta v} \tag{6-5}$$

where ΔP is pressure rise, Δh is the head rise, Δv is the change in velocity, and the subscript, a, means allowable. The terms in the denominator are for instantaneous valve closure.

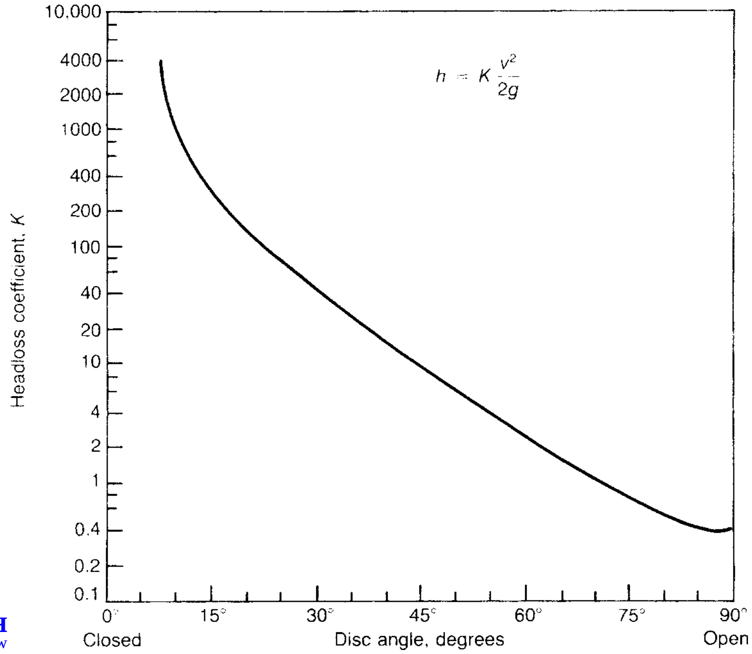
Manufacturers can provide either (1) curves of the valve headloss coefficient, K, in Equation 3-15 wherein $h = Kv^2/2g$ (shown for a butterfly valve in Figure 6-3) or (2) values of flow coefficients, C_v, wherein C_v equals water flow in gallons/minute at 70°F and 1 lb/in.^{2} pressure differential shown for ball and eccentric plug valves in Figure 6-4 (see Appendix A for conversion to SI units). Both K and C_v vary greatly with valve opening, so either is a convenient aid for solving valve problems.

Equation 6-5 can lead either to simplified equations or to computer solutions for determining the allowable rate of valve closure so that a given pressure is not exceeded. To be valid, a method of solution must include

- the change of valve coefficient, K, in the formula for headloss (Equation 3-15), or
- the shape of the valve closure curve, C_v/C_{vo}, and the nonlinear action of some valve stems, and
- the effect of the increasing pressure, which tends to sustain the original flow through the valve. The pressure, for example, depends on the shape of an associated pump curve and the location of the valve as well as on other characteristics of the system.

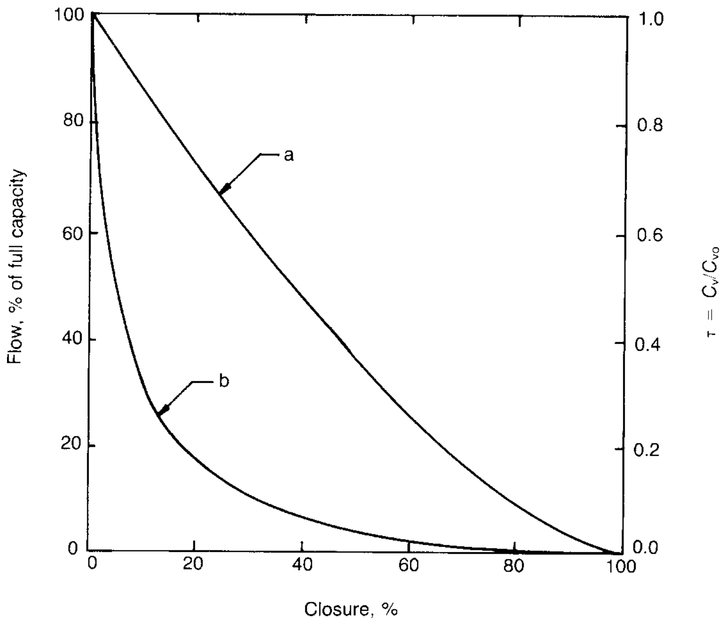
Because resistance to flow is relatively small in the initial stages of closure, a valve can be quickly closed to near shut-off if the final closure is slow. Of the three gate valve closure programs occurring in the same interval, $3t_c$, as shown in Figure 6-5, the least pressure is generated by a 95% closure in $0.5t_c$ with the remainder of the closure occurring in $2.5t_c$.

A valve poorly selected with respect to C_v closure characteristics can often be the difference between a



LIVE GRAPH
Click here to view

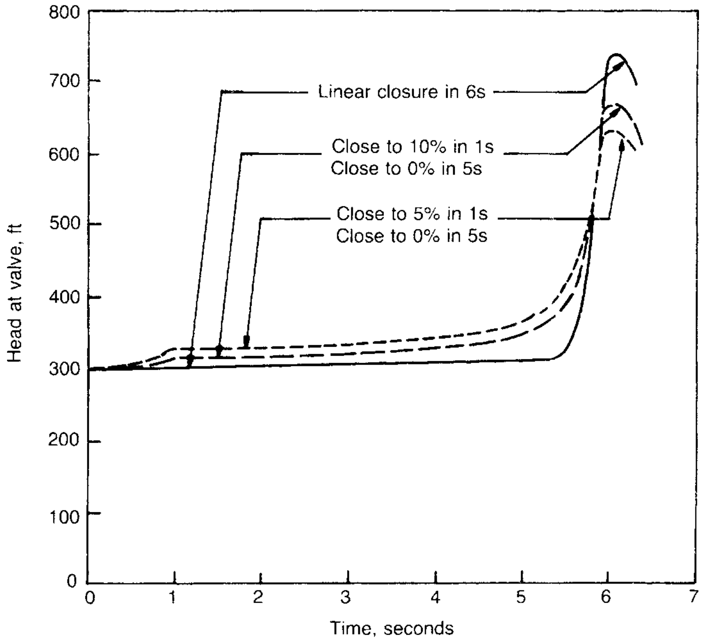
Figure 6-3. Headloss coefficient for a butterfly valve.



LIVE GRAPH
Click here to view

Figure 6-4. Valve closure functions. (a) Plug valve (DeZurik Series 100), the plug angle is linear with the time of closure; (b) ball valve (Williamette List 36), actuator travel is linear with the time of closure.

problem situation and no problem at all. Ball valves (with their worm and compound lever actuators) have the ideal characteristic of closing rapidly at first followed by very slow closure without any complicated control gear. The capacity of the valve is reduced 80% at about 20% closure (see Figure 5-2).



LIVE GRAPH
Click here to view

Figure 6-5. Water hammer caused by various gate valve closure programs. After Watters [4, p. 195].

In contrast, the eccentric plug with linear valve stem angle (or percentage of closure) has a near linear characteristic and would require either a much longer time for closure or a programmed closure to equal the ball valve's control of surge pressure.

As a rule, valves should not be closed in less than 2 to 10 times t_c. Insist on a computer solution wherever more certainty than this is needed, or when the problem is important because of the size of the pipe, the amount of flow, or the dynamic pressure involved.

Example 6-2
Seating a Valve

Problem: A 200-mm (actually, a 203-mm or 8-in.) butterfly valve discharges a flow of 12.6 L/s (200 gal/min) to atmosphere from a level pipe 4.83 km (3 mi) long with a gauge pressure of 690 kPa (100 lb/in.^{2}) just upstream of the valve. The valve design is such that it will close from its fully open position in 50 s.

Find the initial valve position (percentage of opening) and estimate the surge pressure when the valve closes.

Solution: First, find the percentage of valve opening from Equation 3-15 and a manufacturer's curve of K as shown in Figure 6-3. Then find the approach velocity in the pipe.

SI Units

$$v = \frac{Q}{A} = \frac{0.0126\,\mathrm{m^3/s}}{(\pi/4)(0.200)^2} = 0.401\,\mathrm{m/s}$$

U.S. Customary Units

$$v = \frac{Q}{A} = \left[\frac{2.00\,\mathrm{gal/min}}{(\pi/4)(8/12)^2}\right]\left[\frac{0.00223\,\mathrm{ft^3/s}}{\mathrm{gal/min}}\right] = 1.28\,\mathrm{ft/s}$$

The differential head across the valve is (see Tables A-8 and A-9):

$$h = \frac{P}{\gamma} = \frac{690{,}000\,\mathrm{N/m^2}}{9789\,\mathrm{N/m^3}} = 70.5\,\mathrm{m}$$

$$h = \frac{P}{\gamma} = \frac{100\,\mathrm{lb/in.^2}}{62.3\,\mathrm{lb/ft^3}} \times \frac{144\,\mathrm{in.^2}}{\mathrm{ft^2}} = 231\,\mathrm{ft}$$

SI Units	**U.S. Customary Units**
From Equation 3-16	
$K = \frac{2gh}{v^2} = \frac{2 \times 9.81 \times 70.5}{(0.401)^2} = 8600$	$K = \frac{2gh}{v^2} = \frac{2 \times 32.2 \times 231}{(1.28)^2} = 9080$

The discrepancy between SI and U.S. units is due only to rounding off the difference between 200 mm and 8 in. From Figure 6-3, the valve is open about 7° (nearly closed).

The wave speed can be calculated from Equation 6-4 or simply estimated as about 100 g. For linear valve closure, the last 7° of valve movement would require $(7/90)50 = 3.9\,\text{s}$.

$a = 100 \times 9.81 \approx (980)$	$a = 100 \times 32.2 \approx (3200)$
$t_c = \frac{2L}{a} = \frac{2 \times 4830}{980} \approx 10\,\text{s}$	$t_c = \frac{2L}{a} = \frac{2 \times 3 \times 5280}{3200} \approx 10\,\text{s}$

Because the valve will close in 3.9 s (less than the critical time, 10 s), it closes quickly enough to generate the full transient pressure given by Equation 6-2.

$\Delta h = \frac{a\Delta v}{g} = \frac{980 \times 0.401}{9.81} \approx 40\,\text{m}$	$\Delta h = \frac{a\Delta v}{g} = \frac{3200 \times 1.28}{32.2} \approx 130\,\text{ft}$

And the total pressure in the pipeline is

$P = 690\,\text{kPa} + 40 \times 9.81 = 1080\,\text{kPa}$	$P = 100\,\text{lb/in.}^2 + 130 \times 0.433 \frac{\text{lb/in.}^2}{\text{ft}} = 156\,\text{lb/in.}^2$

The assumption of a steady pressure upstream from the valve is often erroneous. The pressure usually changes with a change in flow as the system follows the characteristic pump H-Q curve. For pumps with type numbers greater than about 85 (specific speeds greater than about 4400), the head developed by a pump increases substantially as the flow decreases. If the type number is about 125 (specific speeds about 6600) or more, the increase in head is dramatic, as shown in Figure 10-16.

If the time of closure in Example 6-2 were greater than t_c, the resulting pressure surge would be less than the 40 m (130 ft) calculated, and a computer analysis would be used to reveal the pressure rise. Note that the shape of the valve closure curve would be very important in such an analysis.

6-6. Surge Concepts in Flow with Friction

The explanation of water hammer in Section 6-4 is simplified by omitting friction. Such simplification can be justified in some systems, such as those with short force mains in which friction head is a small part of the total design head. In most systems, however, friction contributes substantially to the total head, and it is important to understand its effect.

Figure 6-6 is similar to Figure 6-1a except for the hydraulic grade line, which slopes during steady-state flow. Shortly after sudden closure of the valve at point O, the wave front would arrive at point A. Consider the anomalies that would ensue by following the construction shown for frictionless flow in Figure 6-1a in which the pressure rise for the wave front at B forms rectangle abcd. Its counterpart is trapezoid oabc in Figure 6-6, but because the hydraulic grade line bc is sloping, water continues to flow past point A (corresponding to point B in Figure 6-1); therefore, v (and, hence, h) would be less in Figure 6-6 than in Figure 6-1a for the same original velocity. But flow continues only until the HGL becomes level. Meanwhile the wave front progresses to point B (Figure 6-6) and the same phenomenon occurs again, with more water flowing past point A, the

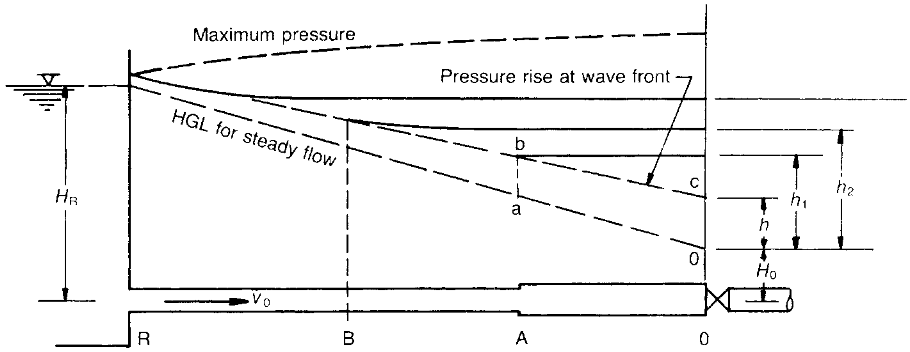


Figure 6-6. Packing and attenuation in a long pipe with friction.

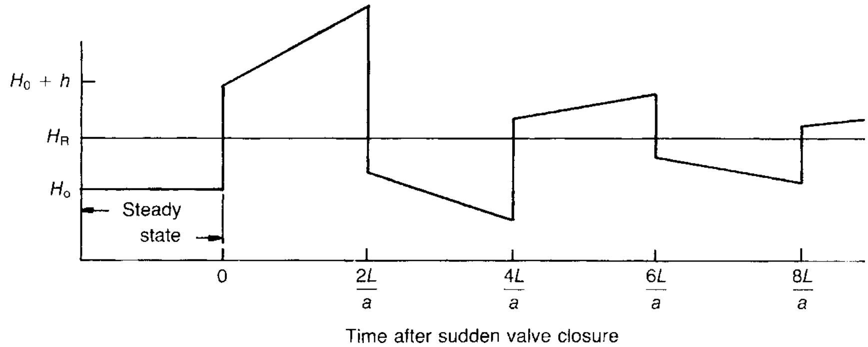


Figure 6-7. Surges at a valve in a pipe with friction.

pressure rising, the pipe expanding, and the water compressing. Wylie and Streeter [3] call this occurrence "packing." As the wave front approaches the reservoir, friction decreases v and h, a phenomenon called "attenuation." Eventually, the pressure builds to a maximum above the level of the reservoir and finally dies to the reservoir level.

Depending on the length of the pipeline, the pressure surges at the valve might appear, as shown in Figure 6-7. With each reversal of the pressure wave, friction decreases the pressure changes, so pressure eventually coincides with the reservoir level.

6-7. Column Separation

In the preceding sections, the pipelines are assumed to be level, but in real systems pipes may slope and downsurges result when power fails or when valves at the upstream end (the usual configuration) close quickly. Under some conditions, the downsurges can cause column separation—a condition to be avoided at any cost by a proper control strategy. As shown in Figure 6-8, a knee (a reduction in gradient or a change from a positive to a negative gradient) makes a pipeline especially vulnerable. If the power fails, the pumps stop quickly with an effect like closing a valve. The upstream flow (near the pumping station) stops whereas, due to inertia, flow continues at the downstream end (near the discharge). The static hydraulic gradeline begins to decay as shown by successive curves labeled $t = 1$ s, $t = 2$ s, $t = 3$ s, until, at $t = 4$ s, a slight negative pressure occurs between the pump and the knee. At $t = 4.5$ s, vapor pressure exists over a considerable length of the pipeline, and the water is boiling and forming large pockets of vapor. Column separation has occurred. On the upsurge of pressure that follows, the vapor pockets collapse and the two liquid columns can come together at literally express-train speed. Since the water is almost incompressible, the forces at impact can be enormous.

The knee makes the situation in Figure 6-8 worse, but note that column separation would occur with or without the knee and would occur even if the pipeline had a uniform gradient. However, if the pipeline profile were flat near the pump and the steep gradient occurred near the reservoir, column separation might be avoided —a useful control strategy that is maintenance free.

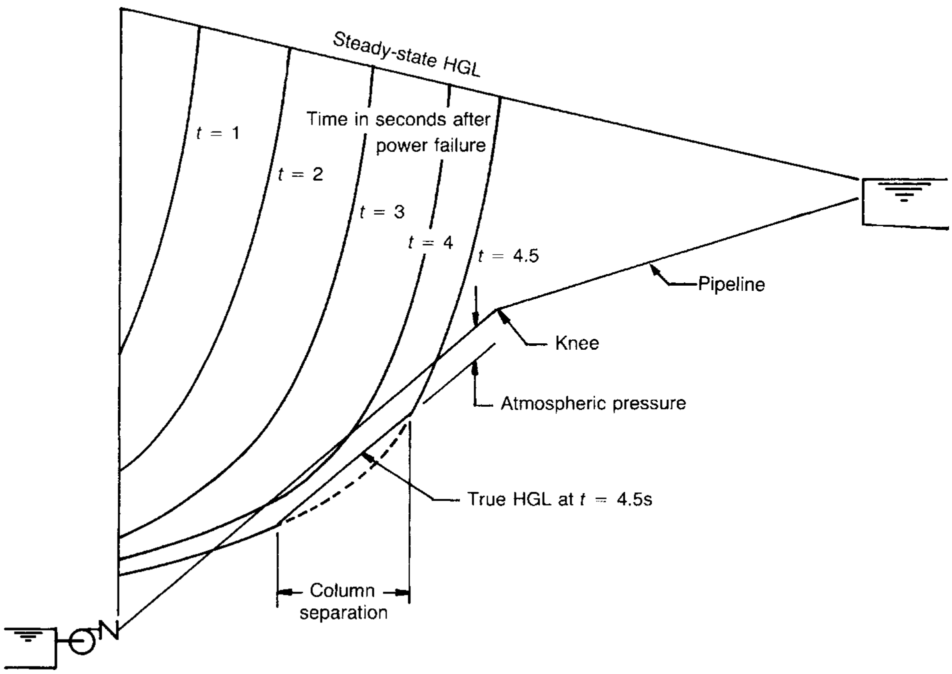


Figure 6-8. Successive hydraulic gradelines following power failure. Adapted from Watters [4, p. 271].

A computer analysis of a similar problem is given by Watters [4, pp. 235 ff], and the results of still another somewhat similar problem with and without surge control devices are shown in Figures 7-12 and 7-13.

The hydraulic gradelines after power failure can sometimes be crudely estimated at several time intervals if the deceleration time of the pump is known or can be estimated. Flow through centrifugal pumps after power failure is a function of many variables, including

- inertia and speed of the pump, driver, and water within the casing;
- length and profile of the pipe;
- steady-state hydraulic gradeline;
- velocity of flow; and
- suction conditions in the wet well.

The true shape of the hydraulic gradelines requires solution by a computer.

6-8. Criteria for Conducting Transient Analysis

Every pump and pipeline system is subject to transient pressures, but in practice, it is impractical to spend the time and expense necessary to analyze all of them. The following empirical guidelines, which seem to be satisfactory in most (though certainly not all) situations, can be used to decide whether a complete transient analysis is required.

Do Not Analyze

- Pumping systems with flow less than $23\,m^3/h$ (100 gal/min). Discharge piping is usually such that velocity is low and transient pressures are low. Even if transient pressures are high, small diameter (100-mm or 4-in.) piping has a high pressure rating and can usually withstand the pressures.
- Pipelines in which the velocity is less than 0.6 m/s (2 ft/s).
- Distribution systems or pipe networks (as in community potable water systems). The many junctions significantly dissipate the pressure waves.
- Reciprocating pumps, because virtually every reciprocating pump should have a pulsation dampener on the discharge (see Ekstrum [7] for methods of sizing such dampeners).
- Pumping systems with a static differential pressure between suction and discharge of less than about 9 m (30 ft).

Warning: it is possible that a very low static head coupled with a relatively high dynamic head could result in a column separation problem.

Do Analyze

- Pumping systems with a total dynamic head greater than 14 m (50 ft) if the flow is greater than about $115\,m^3/h$ (500 gal/min).
- High-lift pumping systems with a check valve, because high surge pressures may result if the check valve slams shut upon flow reversal.
- Any system in which column separation can occur: (1) systems with "knees" (high points), (2) a force main that needs automatic air venting or air vacuum valves, or (3) a pipeline with a long [more than 100 m (300 ft)], steep gradient followed by a long, shallow gradient.
- Some consultants analyze any force main larger than 200 mm (8 in.) when longer than 300 m (1000 ft).

Checklist

Some additional insight can be gained from the following conditions, which tend to indicate the seriousness of surge in systems with motor-driven centrifugal pumps. A serious surge may well occur if any one of these conditions exists. If two or more conditions exist, a surge will probably occur with a severity proportional to the number of conditions met [8–11].

- There are high spots in pipe profile
- There is a steep gradient: length of force main is less than 20 TDH
- Flow velocity is in excess of 1.2 m/s (4 ft/s)
- Factor of safety (based on ultimate strength) of pipe (and valve and pump casings) is less than 3.5 for normal operating pressure
- There can be slowdown and reversal of flow in less than t_c
- There is check valve closure in less than t_c
- There is any valve closure (or opening) in less than 10 s
- There can be damage to pump and motor if allowed to run backward at full speed
- Pump can stop or speed can be reduced to the point where the shut-off head is less than static head before the discharge valve is fully closed
- Pump can be started with discharge valve open
- There are booster stations that depend on operation of main pumping station
- There are quick-closing automatic valves that become inoperative if power fails or pumping system pressure fails.

Criteria for determining whether to use simple hand calculations or a more detailed computer program are also given in *Pipeline Design for Water and Wastewater* [8, p. 65].

Shut-downs will occur, so plan for them. They can result in low pressures and column separation at knees in steep pipelines. Air venting valves that close too rapidly while an empty pipe is being filled also cause destructive hydraulic transients. Even on low-lift pumping stations, depending on the pipe profile, column separation can occur in the vicinity of the discharge header or farther downstream.

Computers

There is no simple, easy way to perform reliable transient analyses. Computer modeling is the most effective means available, but there are practical constraints on time and cost. Both computer time and the labor needed to analyze and review are expensive, so the extent of the analysis should be related to the size and cost of the project. For example, spending \$1000 to \$5000 to analyze transients for a \$1,000,000 project is probably worthwhile even if no problem is found.

6-9. References

1. Parmakian, I., *Waterhammer Analysis*, Dover, New York (1963).
2. Rich, G. R., *Hydraulic Transients*, Dover, New York (1963).
3. Wylie, E. B., and V. L. Streeter, *Fluid Transients*, Feb Press, Ann Arbor, MI (1982, corrected copy 1983).
4. Watters, G. Z., *Analysis and Control of Unsteady Flow in Pipelines*, 2nd ed., Butterworths, Stoneham, MA (1984).
5. Chaudhry, M. H., *Applied Hydraulic Transients*, Van Nostrand Reinhold, New York (1979).
6. Wood, D. J., and S. E. Jones, "Water hammer charts for various types of valves," *Journal of the Hydraulics Division, Proceedings of the American Society of Civil Engineers*, 167–178 (January 1973).
7. Ekstrum, J. D., "Sizing pulsation dampeners for reciprocating pumps," *Chemical Engineering* (January 12, 1981).
8. *Pipeline Design for Water and Wastewater*, American Society of Civil Engineers, New York (1975).
9. AWWA M11, *Steel Pipe—A Guide for Design and Installation*, p. 54, American Water Works Association, Denver, CO (1985).
10. Kerr, S. L., "Minimizing service interruptions due to transmission line failures: Discussion," *Journal of the American Water Works Association*, 41, 634 (July 1949).
11. Kerr, S. L., "Water hammer control," *Journal of the American Water Works Association*, 43, 985–999 (December 1951).

Chapter 7

Control of Hydraulic Transients

BAYARD E. BOSSERMAN II

CONTRIBUTORS
Robert C. Glover
William A. Hunt
Joseph R. Kroon
M. Steve Merrill
John Vuncannon
Gary Z. Watters

The advantages, disadvantages, and typical uses of various kinds of control devices (such as tanks and valves) and control strategies (such as timing of valve operations, slow filling of empty pipelines, and adequate maintenance) used to limit transients are discussed in this chapter. Coping with transients is site-specific, so the discussion is limited to examples that illustrate only a few of the many different problems designers may encounter. Chapter 6 is prerequisite to this chapter.

The approach to design is the same for all strategies. If surges are expected to be severe (see Section 6-8), the piping system is modeled for a computer solution and the magnitudes of surges for the critical conditions are determined. If surges are excessive, a promising control scheme is selected and the system is then analyzed again by computer. The process of selecting an appropriate candidate solution and analyzing the results is continued until satisfactory control and reliability are achieved for minimum (or at least supportable) cost—a decision that usually requires considerable judgment.

Although the details of computer analyses are beyond the scope of this book, some of the factors involved in choosing a solution are included.

7-1. Overview of Hydraulic Transient Control Strategies

Every piping system for water and wastewater should be evaluated for water hammer. From a review of the plan and profile of the pumping station and pipeline system as well as the operating scheme, it is frequently possible to determine where potential hydraulic transient problems may exist and what means might be taken to control them. For a quick check to determine the potential for column separation, draw a mirror image of the hydraulic profile, as in Figure 7-1. If the pipe lies well above the mirror image, column separation might occur or even be likely. If column separation can occur, or if the criteria for required analysis in Section 6-8 are met, use (or contract for) a trustworthy computer analysis of the system [1].

Make every effort to eliminate (or, at least, to mitigate) transients by avoiding knees, high spots, steep gradients near the pump, and air (or, much worse, vacuum) in pipes. When any of these conditions cannot be avoided, use a combination of pipe strength and control strategies to provide adequate protection at reasonable cost.

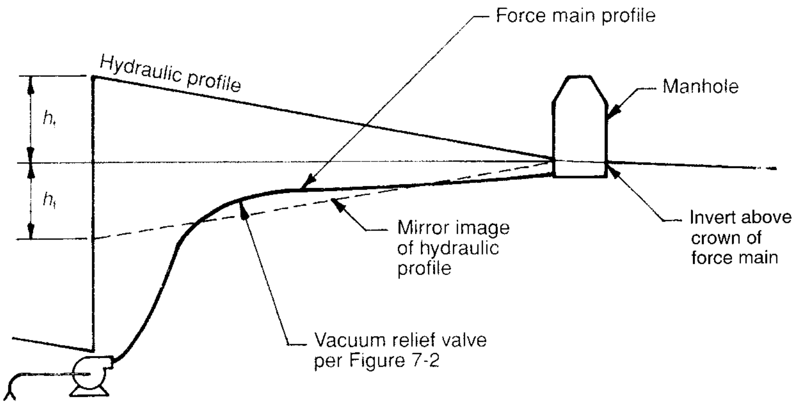


Figure 7-1. Construction for indicating the probability of water column separation in pipelines.

Recommended Options for Wastewater Systems

Because wastewater contains (1) microorganisms that can form hydrogen sulfide and sulfuric acid; (2) grease (sometimes in large quantities) that floats, tends to stick to walls, and can eventually develop into a hard blanket; (3) stringy material (rags, paper) that tend to wrap around any protrusion; and (4) grit that sinks into any depression, only a few strategies can be used to control water hammer in wastewater pumping. See Section 7-7 for additional details. Briefly, surge control strategies are limited to the following, more or less in order of preference:

- Rerouting the force main to avoid knees or negative gradients so that air traps are eliminated and, potentially, the need for special surge protection can be avoided.
- Using an alternative source of power, such as an engine on some pumps, so that the entire system cannot fail at once. Properly designed, engine drives offer superior protection against power failure. Engines rarely fail or shut down suddenly even when starved for fuel, and—unlike motors in constant speed (C/S) operation—in normal operation, speed can be increased or decreased gradually.
- Using pump control valves that open and close gradually, or (on very long pipelines) using variable speed (V/S) drives for ramping speed up and down slowly. Either prevents the continual pounding of the system. The latter, however, offers no protection when power failure occurs in a system where a column separation is likely.
- Increasing the rotating moment of inertia of the pump and driver by adding a flywheel.
- Using high-strength pipe. This option is limited to small systems, but if the force main is 200 mm (8 in.) in diameter or smaller, it may be the cheapest solution. But be aware that the continual pounding of surges may cause leaks and eventual failure. Note: All thrust restraints must be designed for the surge pressures.
- Combination air release-vacuum relief (air-vac) valves. The valves, however, must be very reliable and installed in a manner that ensures reliability (see Figure 7-2). The type of mechanism shown in Figure 5-16 is prone to clogging with grease even when installed in a tall form body. The editors recommend only Vent-O-Mat® valves for combination vacuum relief and air release (see Figure 5-17) or valves similar to the Golden–Anderson Model 911 valve (essentially a swing-check valve) for vacuum relief only.

The vents in Figure 7-2a operate due to the temperature differential between top and bottom of the vault. Each vent should be as large as the air-vac valve size. Insulate the vault if necessary to keep the valves from freezing in the worst weather. The rock floor is alright for water, but for wastewater, the floor should be concrete sloping to the sump to make cleanup possible. For 100% reliability, install two sump pumps and connect a high-level alarm to a central station. The air intakes of the air-vac valves must *always* be above any possible high water level. If necessary, attach a breather pipe to the air inlet, extend it above the vault cover, and terminate it in a 180-degree bend. (If the valve leaks and fills the breather with water, the breather is quickly emptied.) The air accumulator intercepts all air flow at the station, and, although desirable for water transmis-

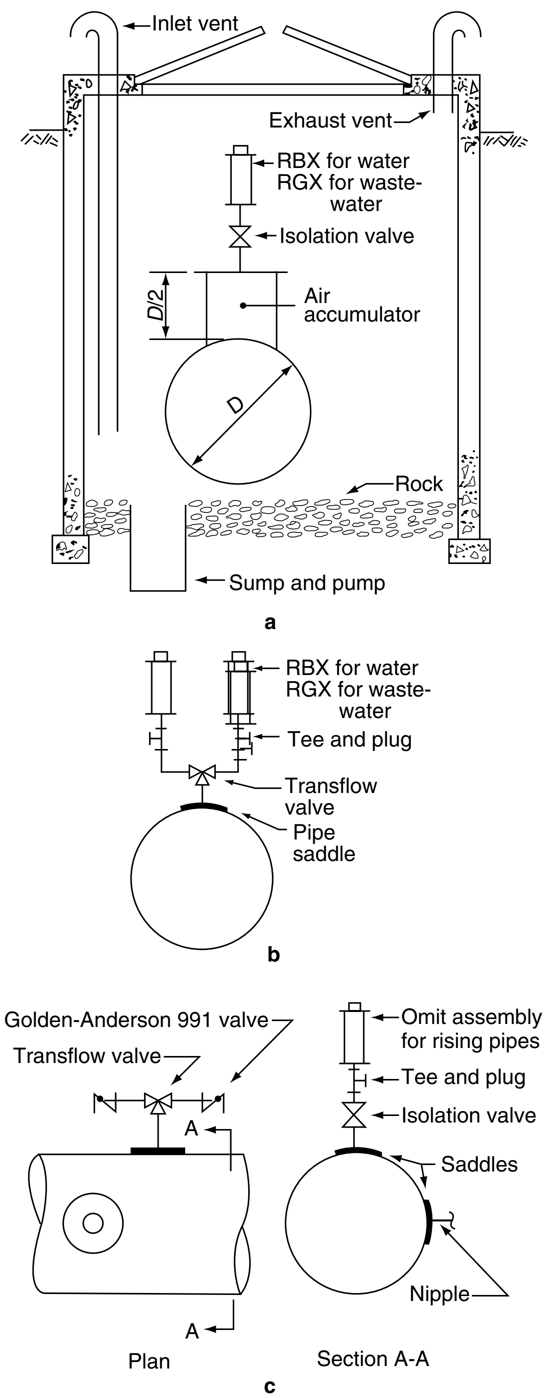


Figure 7-2. An air-vac station with Vent-o-Mat valves (see Figure 5-17). (a) Recommendation of International Valve Marketing, Inc. [2]; (b) a more conservative design (with vault not shown); (c) an alternative, conservative design (with vault not shown).

sion mains with (RBX instead of RGX) Vent-O-Mat® valves, it is a grease trap for wastewater and probably should not be used if the grease load is expected to be high. While the wastewater is flowing, turbulence keeps grease mixed, but when the pumps are off (as happens many times per day with constant-speed pumps) the grease rises into the accumulator and then into the air-vac valve. As the isolation valve is always wide open except during maintenance of the air-vac valve, almost any type of valve can be used, ranging from ball valves (in small sizes) to butterfly or resilient seated gate valves (for larger sizes). The ban against gate valves on vertical pipes is unimportant here because grit does not pass the valve and cannot clog the valve bonnet. The single isolation valve is, however, a problem. Murphy's Law ("Anything that *can* go wrong *will* go wrong . . . ") and its corollary (" . . . at the worst possible time.") dictates that someday a maintenance worker will leave the valve closed, the pipeline will be unprotected during a power failure, and the pipeline may rupture.

In the more conservative design of Figure 7-2b, the single isolation valve is replaced by a Nordstrom Transflow™ multiport valve or PBM T-port valve. Either valve can be modified so that both exits cannot be closed at the same time, but either or both can be open at the same time. Hence, Murphy's Law does not apply. The tee makes testing the valve (by hooking pressurized or vacuum tanks or air and vacuum pumps to it) easy and convenient. The air accumulator is omitted thus furnishing some protection for the air-vac valves. Although the valve can remove large pockets of air, small air bubbles moving at speed along the soffit of the pipe are likely to pass the station. The bubbles are of no concern hydraulically, but air encourages corrosion, and iron or steel pipes lined with cement mortar (or concrete pipes) may not last long. Consequently, use plastic force mains or line metallic pipe with plastic.

The design of Figure 7-2c, with a pipe saddle at the springline is even more appropriate in force mains subject to heavy grease loads. There is almost no tendency for grease to enter a horizontal pipe at the springline, and the swing check valves are not affected by grease. In pipelines with a rising gradient (as in Figure 7-1) the side-mounted system alone is entirely adequate and has been used with success for many years. A vacuum of 7 to 14 kPa (1 to 2 lb/in.^{2}) is sufficient to open the spring-loaded check valves to prevent any greater vacuum in the force main. Air is expelled by both velocity (see Table B-9) and gravity. If the knee is followed by a flat or descending gradient, an RGX (or RBX) valve can be added at the top of the pipe for expelling large pockets of air. Here, a

single isolation valve is permissible, because a closed valve during a power outage is only objectionable—not disastrous.

All air-vac valves should be sized to match the volumetric flow of water with an equal volumetric flow of air at a pressure drop of about 14 kPa (2 lb/in.^{2}) or less. Note that the use of air-vac valves requires an aggressive maintenance program to assure they will be operational when needed. Make that clear to the owner when adopting this type of transient protection. Both Vent-O-Mat® and Transflow™ valves are expensive and some client resistance can be expected, but a proper air-vac valve station costs only about 1 to 3% of the cost of the pipeline—inexpensive insurance against disaster.

Air scouring velocities (per Table B-9) can be reached by decreasing the diameter of the pipe at knees or in any pipe laid flat or at downward gradients. To minimize friction losses, only the pipe on a flat or downward gradient need be made small to obtain the high velocities required. (Note that the custom of never decreasing the diameter of a sewer line does not apply to force mains. Reducing pipeline diameter may, however, preclude the use of pigs for cleaning.) Air-scouring velocities must be reached frequently—at least once per day. It is wise to include pigging facilities because pigging removes the bacterial slimes that produce hydrogen sulfide.

Options Undesirable for Wastewater Systems

Protection strategies that are undesirable or impractical in varying degrees for wastewater systems include:

- Hydropneumatic tanks. The use of hydropneumatic tanks (or air chambers) is highly controversial, but in any situation it is always a means of last resort. Opponents state that grease will enter the tank with each surge event, and eventually the grease may harden and might clog piping, thus eliminating the supposed protection. Some operators have stated that they immediately close valves to remove surge tanks from the system because of grief with them in the past. Other operators have not experienced the above difficulties and have not been upset over surge tanks in their pumping stations. A few have been enthusiastic, because the tanks effectively controlled surges. On the other hand, many have complained about the maintenance required for the compressors, level controls, pressure relief valves, and grease accumulations on sight gauges. Proponents also claim that running a stream of fresh water (with, of course, proper backflow prevention) into the tank keeps out the wastewater, although opponents reiterate that, as grease floats and surges introduce wastewater anyway, grease will accumulate and may eventually prevent proper functioning and expose the designer to lawsuits. In a water-short situation, a steady flow of fresh water into a surge tank at a rate needed to purge the tank effectively may be unacceptable.
- Conventional air-vacuum valves. The protection these valves afford is most uncertain, and their lifecycle cost is exorbitant. But because their first cost is the lowest of all, they may sometimes be demanded by the owner. If they are to be used despite these objections, the owner should be warned in the preliminary design report and in the O&M manual that even the best conventional valves will require a high level of maintenance—a minimum of servicing monthly—wherein the best maintenance is replacement of the valve's interior mechanism and cleaning the replaced parts in the shop. However, see Section 5-7 for a satisfactory alternative.

All air-vacuum valves used to prevent pipe failure should be installed in pairs so that the surge control system will function even while one valve is being serviced. Always install the inlets for air-vacuum valves above any possible high water level to ensure air, not water, enters the valve when a vacuum occurs.

Recommended Options for Water Systems

All of the recommended strategies for wastewater also apply to water systems. Additional options for water systems include:

- The use of hydropneumatic tanks (air chambers) or standpipes.
- The use of one or more air release and vacuum relief valves—an inexpensive, first-cost solution but one in which inspection and perhaps cleaning is required twice per year as a minimum One major pipeline failed because in 20 years of service, the air release and vacuum relief valves had never been maintained. Recognize that the failure of any means for protecting a pipe can lead to utter disaster.

Means for coping with transients are discussed in detail in Section 7-6.

Power Failure

Power failure will occur several to perhaps many times, so plan carefully for it. Power failure at a

pumping station results in an initial rapid downsurge in the discharge header and piping close to the pumping station. The change in head that occurs in the pump discharge piping after power failure (or pump shut-off) is shown in Figure 6-8. The following methods can be used to reduce the magnitude of a downsurge.

- Increase the inertia of pump and motor by adding a flywheel. Engines have higher moments of inertia than do electric motors, and it is easy to increase the size of the flywheel. Flywheels control downsurge and can prevent column separation (which is especially important in wastewater systems), whereas control valves (which control upsurge) cannot.
- Install a hydropneumatic tank (air chamber) near the pumping station and, perhaps, at other points as well (see Figure 7-3). Check valves near air chambers must be fast-acting but slamproof because the short water column between the air chamber and the check valve can quickly accelerate.

Column Separation

The most serious consequence of downsurge is column separation, which must always be avoided. It is more likely to occur if there are knees in the pipeline (as in Figure 6-8) and is almost certain to occur at a high point. Column separation can also occur at other places along the pipe. The location and extent depends on the relative relationship between the dynamic head, the total head, and the profile of the pipeline (review Section 6-7). A quick way to determine whether column separation is likely is shown in Figure 7-1. If the mirror image of the hydraulic gradient intersects the pipeline profile, column separation will occur. Column separation may also occur as depicted in Figure 6-8. Various means for preventing column separation damage in raw wastewater force mains include (but are not limited to) the following:

- Relocate the pumping station and the associated pipeline to avoid knees or high points.
- Reroute the pipeline or bury the pipe deeper to achieve a flat gradient near the pump so that successive hydraulic grade lines (as in Figure 6-8) do not intersect the pipe.
- Install one-way air valves or, preferably, a check valve to admit air into the pipeline on the downsurge and to trap the air on the upsurge.
- Add a flywheel sized to prevent the column separation.

In addition to these four methods, column separation in water transmission pipelines can be prevented by the following means:

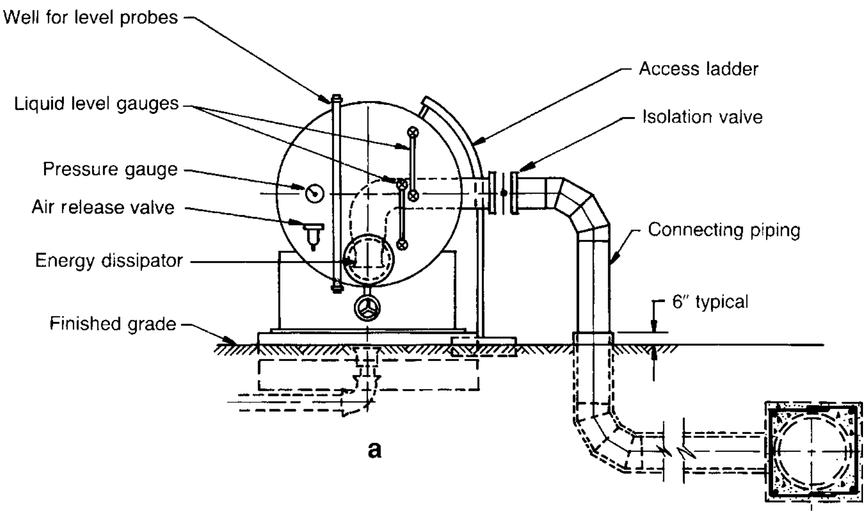
- Install open surge tanks (standpipes) at knees or high points if the required height is not excessive. The stored water can control both high and low pressures (see Figure 7-4 and refer to Chapter 10 in Watters [3]).
- If the hydraulic gradeline (HGL) is too high for an open surge tank, a one-way (Figure 7-5) or a two-way (Figure 7-6) surge tank can be used to keep the HGL above the pipe. A check valve allows flow from the tank to the pipe but not from the pipe to the tank.
- Install air valves, more specifically called "vacuum relief and air release valves" or "combination air valves" (Figure 5-16), that admit air into the pipe and thereby prevent a vacuum. Upon the following upsurge, the air must be exhausted slowly enough to prevent overpressure when the two water columns meet.
- Install a hydropneumatic tank (air chamber). More than one might be required for very long pipelines to ensure reliability.

Except for pipeline rerouting and the flywheel, all of these methods require routine maintenance, which cannot be guaranteed and is not always provided.

Start-Up

Pump start-up can cause a rapid increase in fluid velocity that may result in an undesirable surge, but usually it is not a problem unless the type number (specific speed in SI units) of the pump exceeds about 135 (specific speed in U.S. customary units exceeds approximately 7000). (Refer to Figure 10-16 for the effect of specific speed on pressure at start-up.) Methods for controlling such surges, in order of increasing cost, are as follows:

- If there are several pumps, start them one at a time at intervals from 4 to 10 times the critical period ($t_c = 2L/a$).
- Program a pump-control valve to open slowly (again, from 4 to 10 times t_c) after the motor starts. Pump motors typically reach full speed within 1 to 2 s after energization (see Figure 7-7).
- Use a variable-speed drive for each pump, ramped up to full speed slowly enough (from 4 to 10 times t_c) to avoid high surges.
- Add a hydropneumatic tank (air chamber), but only if quick-acting, slam-resistant check valves are used.



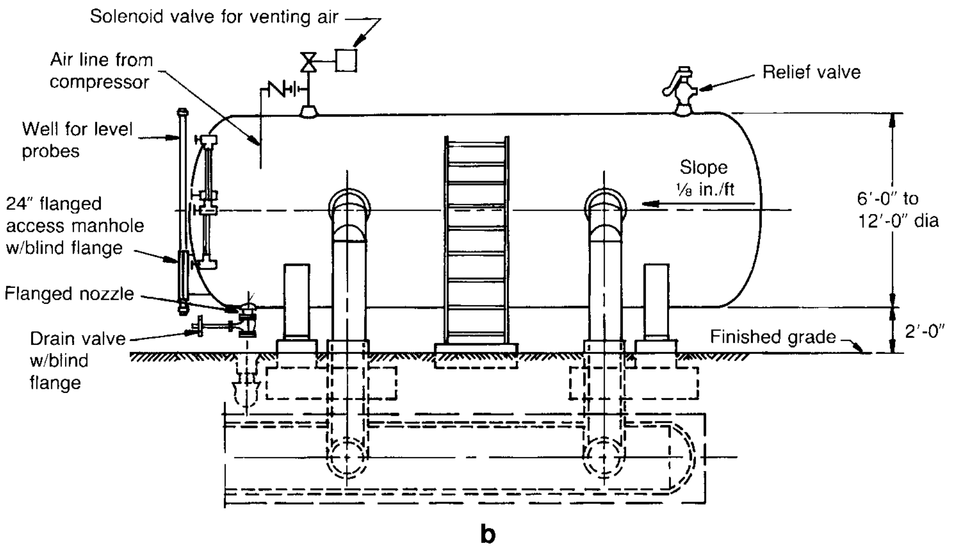


Figure 7-3. Horizontal hydropneumatic tank (air chamber) for clean water service. (a) End elevation; (b) side elevation.

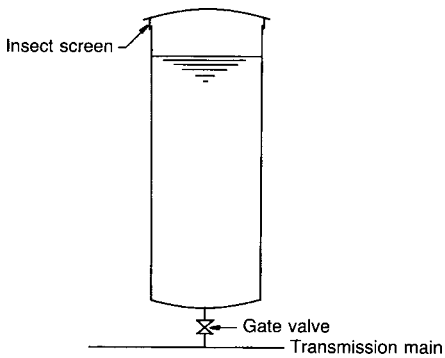


Figure 7-4. Open-end surge tank or standpipe.

Shut-Down

Normal pump shut-down may also cause surges. They can be controlled to remain within acceptable bounds by the following methods:

- Turn pumps off one at a time at intervals from 4 to 10 times t_c.
- Program a pump-control valve to close slowly (4 to 10 times t_c) before the motor is stopped.
- If pumps are equipped with variable-speed drives, ramp down slowly.
- Increase the inertia of the motor and pump unit so that it coasts to a stop over a longer interval.

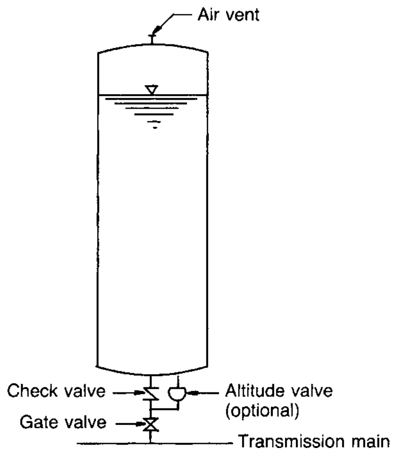


Figure 7-5. One-way surge tank.

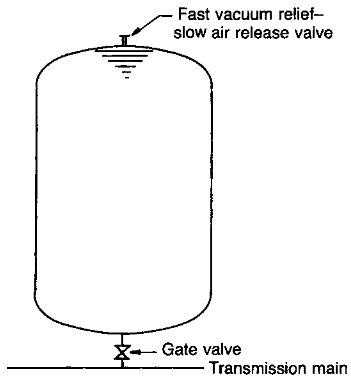


Figure 7-6. Two-way surge tank.

- Add a hydropneumatic tank (air chamber). The tank should not be needed for normal pump shut-down, but if one is installed for other reasons, it would be an effective control method.

Check Valve Slam

If the pumping station is small or if there are several duty pumps, the sequencing of the pump shut-down is usually adequate to prevent valve slam. However, when the last pump shuts down, or when power fails, the liquid comes to rest and then reverses. An unrestrained check valve disc, seating after the fluid flow reverses, closes with a resounding and disconcerting slam that may shake the building. The slam may or may not be accompanied by a significant hydraulic surge. If slam does occur, there are several remedies that may, depending on circumstances, mitigate the problem.

- Ensure quick closure of the valve *before* the flow can reverse to cause a slam by adding a stiff spring (or a counterweight) to an outside lever attached to the disc shaft.
- Ensure slow, gentle seating of the disc (especially during the last 5% or so of closure) by adding an oil-filled dashpot to an outside lever attached to the disc shaft.
- Substitute a pump-control valve that cannot slam. It must be equipped with a stored energy closure mechanism to ensure operation when power fails.

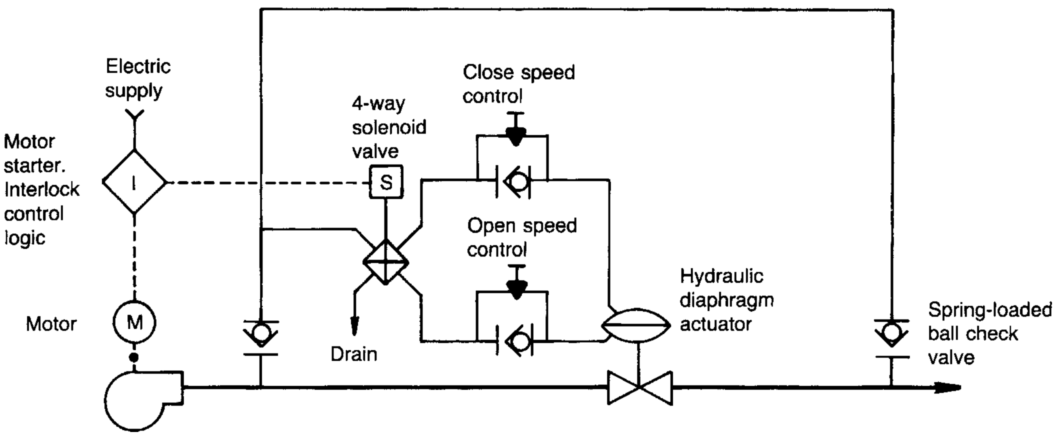


Figure 7-7. Pump control valve system. To ensure the highest available operating pressure, take the control water supply from both sides of the check valve. Provide separate speed-control needle valves on each side of the double-sided diaphragm to control the water flow rate out of the diaphragm actuator.

- Use rubber-cushioned flapper seats to prevent metal-to-metal contact and to cushion the closure.
- Install a pressure-actuated relief valve (see Figure 7-8).

For a more extensive discussion, refer to "Slam" in Section 5-4.

Choosing Check Valves

Choosing the right check valve is vital. Often there are profound differences in the same kind and style of valve offered by different manufacturers. Headlosses in some makes of swing check valves are twice as great as in others, and the massiveness of stressed parts varies widely.

Swing check valves should always be equipped with external levers, which are useful in several ways. The position of the lever indicates whether flow is occurring, and the lever can be equipped with an inexpensive switch that shuts off power to the motor (after a timed delay) if flow does not occur. The lever can be equipped with either springs or counterweights, which can be adjusted to mitigate slam or disc flutter. Dashpots can also be added to control slam. But whether counterweights, springs, dashpots, or pump-control valves are used, it is mandatory that the valves be properly adjusted in the start-up procedure; it is equally important that operators understand the valve operation and practice the necessary preventive maintenance.

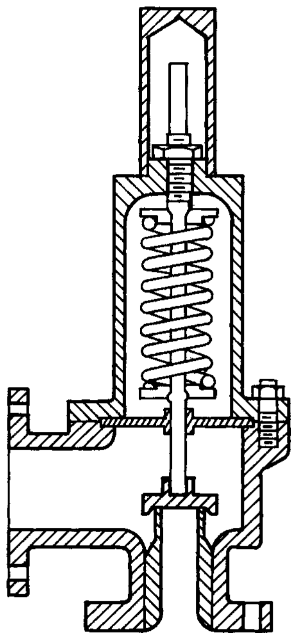


Figure 7-8. Pressure-actuated surge relief valve.

Because many contrary opinions prevail, designers should be extremely careful in specifying check valves. Investigate them thoroughly by obtaining and analyzing advice from several sources.

Filling Empty Pipelines

Small air bubbles, which collect at summits of pipelines, can be bled off with air release valves without creating hydraulic transients. However, the initial filling of pipelines must be done cautiously with velocities kept below 0.3 m/s (1 ft/s) and with air release valves open to exhaust the air slowly. Avoid full-capacity start-ups until all of the large air bubbles are exhausted. Provide properly sized air and vacuum valves or slow-acting pump-control valves for pumps with long discharge columns. These measures complete air evacuation without sudden slamming of valves, which creates high transient pressures. Always include the start-up procedure for empty pipelines in the O&M manual.

Pipelines that are nearly flat may require air release valves spaced at intervals of approximately 400 to 800 m ($\frac{1}{4}$ to $\frac{1}{2}$ mi) to vent air properly during filling. Knees and high points require both air and vacuum valves. At best, air pockets in pipelines increase flow resistance by as much as 10% or even more. At worst, air pockets can generate pressures as high as 10 times normal operating pressures, and air trapped at high points reduces the water cross-sectional area and, thus, acts like a restriction in the pipe.

Air release cocks or valves must be installed in pump casings (or at high points of pump manifolds) to prevent air binding. If the HGL can fall much below any part of the system on a downsurge, an air and vacuum valve is needed to prevent vapor cavities. Limit the vacuum to about half of an atmosphere, and exhaust the air slowly so that it acts as a cushion. Alternatively, especially for raw wastewater, reroute the pipeline to produce a uniform or, better, an increasing gradient with no knees.

7-2. Control of Pumps

None of the methods for controlling water hammer is universally applicable. Some methods might control one cause of water hammer but leave the system unprotected from other causes. Some methods may be unacceptable for a variety of reasons, such as excessive maintenance or unreliability. Because a

single device is often inadequate, several must be used to provide full protection. Several schemes for controlling pumps can be used to limit surges during start-up and shut-down. Some methods offer limited control of surges due to power failure.

Pump Sequencing

By controlling the sequence of pump start-up and shut-down so that the starting and stopping of several pumps are staggered, the magnitude of change in flow at any one time is reduced—often to acceptable levels for normal operation. Sequencing is automatic, reliable, and inexpensive.

Pump-Control Valves

By interlocking the pump with control valves in its discharge piping, transient problems caused by pump start-up and shut-down can be greatly reduced. The control valves are set to open and close slowly (4 to 10 times t_c). Upon start-up, the pump operates against a closed valve. As the valve opens, the flow into the pipeline gradually increases to the full pump capacity. Upon shut-down, the control valve slowly closes to decelerate the flow, after which power to the pump is shut off (but not until the valve is fully closed).

To circumvent power failure, the valves should be operated by a stored-energy auxiliary power source such as trickle-charged batteries for electric systems or a compressed air tank (with enough capacity to operate every valve through two cycles) and either a water or an oil valve actuator. Ball valves are excellent for either water or wastewater service. For water service, butterfly valves can be used, but they have poor throttling characteristics when slow closing is required to control head rise following a power failure. Diaphragm-actuated globe valves operating off pipeline pressure with a check valve feature can also be used. For wastewater service, eccentric plug valves can be used. Pump-control valves, however, cannot prevent downsurge on power failure.

Increasing the Rotational Inertia

The moment of inertia of the pumping system has an important effect upon hydraulic transients in the piping downstream from a pumping station. Upon pump power failure, the pump speed and head rapidly decrease and a negative pressure wave, or downsurge, is propagated down the pipeline. The greater the moment of inertia, the slower this decrease in head and pump speed and the lower the magnitude of the resulting downsurge becomes. Thus, adding a flywheel increases the moment of inertia of the system and slows the decrease in pump speed and head. A flywheel can be added to an electric motor by extending the housing or frame (usually toward the pump) to enclose the flywheel and to support bearings above and below the flywheel. When the conditions are suitable, high inertia is the most reliable method of all for control of surge in both normal operations and power failure, and, except for maintaining the bearings, it is maintenance-free. An alternative is to use an engine. It is easy to enlarge the flywheel, and engines inherently take longer to come to a stop (even when starved for fuel) than do motors.

Variable-Speed Drives

A variable-speed drive that can be ramped up or down slowly has several advantages for controlling transients during normal operation: (1) enhancement of motor life due to infrequent starts and, for adjustable frequency drives, low inrush current; (2) flickering of lights (annoying to nearby residents) caused by inrush is avoided; and (3) the change of flow is gradual and does not upset primary sedimentation tanks or other processes in a wastewater treatment plant.

The disadvantages of a variable-speed drive are that (1) it does not provide protection from power failure; (2) such a drive for a motor is expensive (it costs more than the motor); (3) it may require special training for maintenance workers; (4) it adds complexity; and (5) it reduces reliability somewhat. If surge control is the only objective, variable-speed drives for electric motors are seldom the best answer.

Engines can be easily used for variable-speed drives unless there is prolonged operation at low loads. Note that engines are more reliable than electric power, although the maintenance required is high.

7-3. Control Tanks

Control tanks range from the simple, reliable standpipe to hydropneumatic tanks (air chambers). If all else fails, they are, *when properly maintained,* a positive and reliable means of control.

Avoid using surge tanks for wastewater if possible. If they are, nevertheless, used with wastewater, they should be flushed intermittently or continuously. Some installations are successful if well maintained, but some experienced engineers will not use air cham-

bers for wastewater under any circumstances. Air chambers are worse than useless if not maintained, because they become inoperable yet nevertheless give a false sense of security. The consequences of inadequate maintenance should be dramatized in the O&M manual.

Standpipes

Standpipes or open-end surge tanks (Figure 7-4) provide a free water surface at atmospheric pressure and act as small reservoirs to accumulate or supply water temporarily to control pressure variations. They are useful at high points where the hydraulic grade line is within 20 or 30 ft of the ground. They are simple to construct, easy to maintain, require no power or other utilities, and are completely automatic in operation.

Overflow drain, piping, and a water disposal area must be provided because upsurges often cause overflow. The height of the tanks may be aesthetically unacceptable. They cannot be used for pipelines with a large variation in HGL, and they must be protected from freezing in cold climates.

One-Way Surge Tanks

One-way surge tanks (Figure 7-5) are somewhat similar to standpipes and, because they can be pressure vessels as well as gravity tanks, they are especially useful if the original HGL is too high to allow the use of a standpipe. If used as a pressure vessel, the tank must be closed with an airtight cover containing air release and vacuum relief valves. But unlike standpipes, they permit water to flow only from the tank into the pipeline. Therefore, a water-disposal system or storm drain (which is needed for a standpipe) is not required. One-way tanks are sometimes used at pumping station discharge headers in low-head (15-m or 50-ft) applications. They are simpler in design and operation than hydropneumatic tanks, but are more complex than standpipes.

Electrical power is required to operate the solenoid valve on the fill line. Special attention to the selection and specification for the connecting check valve is required. The valves are usually in low-pressure service and require a soft seat material to prevent water from leaking past the disc.

Two-Way Surge Tanks

Two-way surge tanks are similar to one-way surge tanks except that they have no check valve and no external fill line (see Figure 7-6). Ordinarily, the tank is full of water. In this condition it is ineffective for an upsurge, but on a downsurge it releases water into the pipeline while a large vacuum relief valve allows air to flow into the plenum to avoid a vacuum. If there is a return upsurge, water flows into the tank against the increasing pressure of air in the plenum, which can escape (but only slowly) through a small air release valve. In this operational sequence, the two-way tank behaves much like an air chamber. Eventually, the surges dissipate and the tank gradually fills with water again.

Air Chambers

An air chamber (or hydropneumatic tank) is a pressure vessel about half-filled with air and half-filled with water and connected to the piping system on the discharge side of the pumps (see Figure 7-3). The air in the tank is compressed, which stores energy available to sustain flow after power failure. Pipe inlets to these tanks are usually fitted with differential orifices that allow water to exit the tank with low loss but cause high energy dissipation for flow into the tank. The operation during a downsurge–upsurge sequence is as follows:

- Upon pump shut-down after power failure, the air pressure forces water out of the tank and into the pipeline. As water leaves the tank, the volume of air increases and the pressure decreases.
- The decreasing pressure causes the flow into the pipeline to decrease gradually. Eventually, the flow in the pipeline stops and then reverses, and the downsurge gives way to an upsurge.
- As water enters the tank, the volume of air decreases and the pressure increases.
- The increasing pressure causes the flow to decrease. Eventually, the flow in the pipeline stops and then reverses. The cycle then repeats until friction gradually halts it.

Because the tank must be large enough for a reserve of water to remain at the end of the downsurge and for a reserve of compressed air to remain in the plenum at the height of the upsurge, the computer modeling must be accurate. Air chambers are very effective and reliable for controlling both upsurge and downsurge and so versatile that they can be used in almost any water pumping system, although they are unusual in systems with force mains smaller than 250 mm (10 in.). They are commonly used on the pumping station headers and at high points. Air chambers require a fair amount of complex auxiliary

equipment and controls and, hence, need frequent maintenance. For long, large transmission mains, the site area may need to be enlarged.

The relationship between air pressure and air volume within the air chamber can be described by the equation

$$P_1 V_1^{1.2} = P_2 V_2^{1.2} \tag{7-1}$$

The use of the empirical exponent 1.2 is standard practice and describes a gaseous expansion between isothermal (exponent = 1.0) and adiabatic (exponent = 1.43). Some analysts use an exponent of 1.3.

The following are the recommended design criteria and accessory equipment for air chamber systems:

- An air compressor to supply air automatically to the chamber. Oil-free air compressors are required by some health departments for potable water systems.
- A solenoid valve to vent excessive air from the chamber.
- Level probes, float switches, or capacitance probes in the chamber or in a separate probe well to start and stop the compressor and open and close the solenoid valve.
- Liquid-level gauges or sight glass to observe water level in the chamber. Equip the sight glass with stopcocks and cleanout plugs so that the glass can be cleaned while the tank is in service.
- A flanged access opening for inspection and maintenance.
- A flanged connection to the main pipeline.
- Design the air chamber in accordance with the *ASME Boiler and Pressure Vessel Code, Section VIII* [4]. The air chamber should bear the ASME Code stamp. Code vessels are, however, very expensive, especially in large sizes.
- Provide a poppet safety-relief valve on the air chamber.
- Specify slosh plates in horizontal air chambers. Use horizontal air chambers only for clean water—never for wastewater.
- The ratio of air:water at normal operating (steady-state) pressure should be about 1:1.
- If an air chamber is used for wastewater or dirty water service, use only a vertical tank with a hopper bottom (conical or elliptical) and a fresh water supply for flushing the tank pumped through a 25- or 38-mm (1- or $1\frac{1}{2}$-in.) pipe. The pump can run either constantly or intermittently. However, read the warning in Section 7-1 before deciding to use an air chamber.

7-4. Valves for Transient Control

Valves can be used effectively to control both upsurge and downsurge at pump start-up and shut-down, but they cannot prevent downstream column separation if that is the problem (as in Figure 6-8), nor can they prevent downsurge upon power failure.

Air and Vacuum Control

Vacuum Relief Valves

The only single-purpose vacuum relief valve unequivocally recommended for wastewater service is shown at the side of the pipe in Figure 7-2c. Although it is reasonably well protected from scum, grease, and stringy material, it must nevertheless be serviced on a regular schedule—at least once every 4 months. It is useful in force mains that do not have negative slopes, because negative slopes require either air-scouring velocities or combination vacuum relief-air release valves. In wastewater force mains with flat or negative slopes, add an RGX Vent-O-Mat® valve at the crown as shown in Section A-A in Figure 7-2c. See the subsection “Recommended Options for Wastewater Systems” in Section 7-1 for a more detailed discussion.

Vacuum relief valves for water service are usually accompanied by air release valves, as described in the next subsection.

Air Release Valves

For bleeding air from pump casings or air pockets from other high points within a pumping station, a simple hand-operated cock is sufficient in small stations. For large stations and for long, flat pipelines, use air release valves that close automatically when air is expelled.

Air Release and Vacuum Relief Valves in Water Mains

Air release and vacuum relief valves (or “air-vac valves”) are needed to remove air during pump start-up and to introduce air to prevent a vacuum following pump shut-down. A critical consideration is sizing the air exhaust to prevent excessive shock when the water columns meet after column separation. Flexibility can be obtained by adding a throttling device for optimizing the release of air.

Air-vac valves may not be able to prevent vapor pockets at high points because air may not

be admitted quickly enough. The location of a vapor pocket in a pipeline is critical and cannot always be predicted accurately, so valves cannot ensure complete protection.

Air Release and Vacuum Relief Valve in Wastewater Force Mains

Use every practical means to avoid the necessity for such valves in wastewater service because of problems with grease and excessive maintenance. Try to design the force main with a positive gradient to its discharge. If that is impossible, design for an air-scouring velocity to occur at least once per day. See Table B-9. If air-vac valves must be used, consult Sections 5-7 and 7-1 on Vent-O-Mat® valves. If the owner or designer insists on conventional air-vac valves, specify stainless-steel trim and provide quick-connects for freshwater flushing. Instead of cleaning the float and linkage mechanism by flushing the valves in the field, however, it is preferable to replace the interior mechanism with a clean one and take the dirty one to the shop for a thorough cleaning and overhaul. Always connect two air-vac valves to a trans-flow three-way isolation valve that can block either air valve but not both at once.

Some engineers object to the use of conventional air-vac valves for wastewater under any circumstances and have always managed to find a different control strategy, but see Section 7-1 for one alternative.

Check Valves

Substantial control of surges is obtainable by selecting the correct valve. Some valves shut off very fast and might be preferred for a short (1-km or ½-mi) transmission line. Some can be adjusted to close very slowly or to close at different rates in three stages. Always choose a valve that closes automatically on stored energy when power fails.

Swing Check Valves

Swing check valves should be supplied with an outside lever with a weight or a spring adjusted to close the valve just before fluid reversal occurs following pump shut-off or power failure. Such check valves are especially useful in low-head (15-m or 50-ft) pumping stations. Note that headloss depends less on flow rate than it does on the adjustment of the weight or spring (see Figures B-2 and B-3).

Cushioned Check Valves

An alternative to quick closure is cushioned closure. Cushioned check valves have either an air-filled or an oil-filled dashpot that can be adjusted for rate of closure. The oil-filled dashpot is much more positive in its action and is more easily adjustable than the air-filled type.

Check valves for wastewater must not obstruct the flow and must not have projections that could accumulate stringy materials. Consequently, such check valves are of the single-disc, top-pivot type. The angle-seated, rubber-flapper type closes the quickest, but its lack of an external indicator of valve position is a disadvantage. A variety of check valves, such as the double leaf and slanting disc types (which are suitable only for water), have varying degrees of resistance to slam.

If an air chamber is installed at a pumping station, the check valves must be compatible in characteristics and location. The stored energy in the air chamber moves a short water column quickly, so a check valve must close very fast. Rubber seats or an oil-filled dashpot to cushion the seating may be helpful in preventing slam. Cushioned check valves are effective in limiting surges in both pump start-up and shut-down, but these valves cannot begin closing until after the pump stops.

Surge Relief Valves

Surge relief valves act to reduce upsurges. They do not control the initial downsurge that occurs on pump shut-down or power failure. Hence, they are most useful in short, steep pipe profiles where reversal of flow quickly follows power failure.

There is a wide variety of valves with guided discs, pistons, flappers, or membranes available in spring-actuated and diaphragm-actuated designs (controlled by springs, air pressure, or hydraulic pressure) that are kept closed until an upsurge arrives. The spring-actuated type is shown in Figure 7-8. Surge relief valves open quickly, remain open until the surge dissipates, and some then close slowly. Because upsurges travel at elastic wave speed, such a valve may not open quickly enough to prevent a very short surge of high pressure. Hence, insert the characteristics of a proposed valve into a computer model of the system to determine what pressure rise to expect. Very short surges are less important than longer ones but are difficult to model.

Surge Anticipation Valves

Surge anticipation valves (Figure 5-14) overcome the disadvantage of the surge relief valve by beginning to

open before the upsurge arrives. The initial reduction of pressure following a power failure is sensed and a timer is then actuated to open the valve and release water before the anticipated high pressure arrives.

The sequence of operation is as follows:

- Pump power failure occurs. Water continues to flow into the pipe away from the pump, but the pressure downstream of the pump drops.
- The flow in the pipe halts, then reverses and flows back toward the pump and closes the check valve. Pressure begins to rise in the pipe next to the pump because the check valve acts as a closed-end pipe.
- The surge anticipation valve has already opened to release water and prevent high pressure. The valve then closes over a period of 3 to 10 times t_c.

This type of valve reduces only the high transient pressures; it cannot control the initial low pressures that occur in the pump discharge piping upon pump power failure or shut-down. The conditions under which such valves can operate are restricted, so consult the valve manufacturer. Pressure relief valves without the "anticipation" feature may be preferable.

Surge anticipation valves are useful for water but, because the ancillary piping can be clogged with debris, they should not be used for raw wastewater.

7-5. Containment of Transients

One method of coping with transients that is especially useful for pipes 200 mm (8 in.) or less in diameter is to specify thick-walled pipe and heavy valves, pump casings, and accessory equipment with higher pressure ratings than necessary for normal operating or test pressures. The disadvantage of such a method (beyond the obvious extra cost incurred) is that many pipe materials and valves may eventually fatigue and fail after several years of being subjected to intense cyclic pressures. Hence, portions of the force main may eventually have to be replaced, which would make this method costly indeed. On the other hand, it is a simple method to use, and the short-term maintenance cost is zero. This solution is common for small pumping systems.

For larger pipes, other devices are more effective and less costly. An overview of the various devices is given in Table 7-1.

Table 7-1. Comparison of Devices for Controlling Hydraulic Transients

Method and application	Advantages	Disadvantages
Pump-control valve (electric, hydraulic, or pneumatic); for water and wastewater	Effective in controlling surges due to pump start-up and shut-down. Allows automatic control system. Can control surges at any operating pressure for the pump and pipeline system.	Ineffective in controlling surges due to power failure. Must have auxiliary power source (e.g., batteries or compressed air) to operate valve when power fails; even then, it may not protect against downsurge and column separation.
Hydropneumatic tanks, primarily for water	Pressurized vessel can be used in almost any pipe and pumping system encountered in the waterworks industry; can be sized to control pressure changes to prescribed ranges; commonly used on pumping station discharge header, high points, or "knees" in pipeline; reliable. Can control both upsurge and downsurge.	Requires a fair amount of complex auxiliary equipment and controls. Requires frequent maintenance like any other mechanical-electrical system; large tanks can be expensive; additional land area required at site, especially for large tanks. Not recommended for wastewater due to buildup of grease and settleable solids and production of gases; if used, specify frequent blow-down to minimize wastewater in tank.
Standpipe; for water only	Nonpressurized tank or tower; simple to construct and operate. Used at high points in pipelines to supply water to pipe to prevent column separation. Does not require electrical power at site.	Cannot be used if normal HGL at high point is more than about 9 m (30 ft) above ground level because tank would be uneconomical. Overflow drain, piping, and water disposal area must be provided if later pressure cycles in pipe system cause water to overflow standpipe. Requires tall tower, which may not be aesthetically acceptable at site; impractical in a pipeline that experiences large variations in the HGL.

Continued

Table 7-1. Continued

Method and application	Advantages	Disadvantages
One-way tank; for water only	Similar to standpipe in function; contains check valve on connecting pipe so tank can be used in system in which the HGL is much above the top of the tank; used at high points in pipelines to supply water to pipe to prevent column separation. Also sometimes used at pumping station discharge header in low-head applications (usually discharge head <15 m or <50 ft); simpler in design and operation than a pressurized air chamber.	Not useful usually on pump discharge if discharge head is greater than about 15 m (50 ft). Requires tall tower, which may not be aesthetically acceptable at site. Requires special attention to selection and specification of the connecting check valve; requires separate fluid supply to maintain water in the tank.
Air release and vacuum relief valve; primarily for water, risky for wastewater, but see Section 5-7	Relatively simple devices to install; used at high points in pipeline to allow air to enter freely to prevent large negative (vacuum) pressures and to exhaust slowly to reduce effects of column separation.	Conventional types require frequent maintenance so that float and linkage mechanisms do not hang up or corrode. They should not be trusted to provide complete protection from column separation. Location in the pipeline is critical and cannot always be predicted accurately. Unconventional types: see Section 5-7.
Swing check valves; for water and wastewater	Can dramatically reduce effect of flow reversal in pipeline after pump shut-off or power failure. Especially useful in low-head pumping. Especially useful in pumping stations in which the discharge head is 15 m (50 ft) or less. No electrical power needed. Simple device to install and maintain.	Cannot prevent downstream column separation in the pipeline if that is the problem. Slam can be a severe problem, but it can be controlled by installing a strong spring-loaded closing device. (see Sections 5-4 and 7-1).
Cushioned swing check valves; for water and wastewater or slanting disc check valves for water only	Same as swing check valves, except slam can be reduced or eliminated. Upsurge can be partially controlled.	Does not control downsurge. May cause pump to run backward. Pneumatic dashpots may be difficult to adjust to control timing of valve operation. Oil-filled dashpots are much easier to control. Consider spring-loaded swing check valves
Surge relief valves; primarily for water	Effective in reducing positive surges caused by reversal of water column in the pipeline after power failure.	Cannot prevent downstream column separation in pipeline if that is the problem. Requires drain piping and disposal area to dispose of the water that drains from the pipeline.
Higher pressure rated piping, valves, and equipment	Simple to use. Used in many small raw wastewater pumping stations and force main systems; short-term maintenance cost is zero.	Pipe and other accessories may eventually fatigue after several years of being subjected to cyclic surge pressures. In the long term, portions of force main may fail and have to be replaced. Cost-effectiveness may be poor.

7-6. Surge Control for Water Pumping Stations

The installations discussed in this section are more or less typical (or at least common), but they are by no means universal.

Deep Well Pump

The start-up of a deep well pump with a high static head poses a special problem. With the pump turned off, the pipe column between pump and check valve is empty. Unless air is admitted to the column, the fast-rising water may slam into the closed check valve (which is holding tons of water at rest), sometimes with a tremendous force that can displace motors and pipes. If such a problem occurs, it can be controlled by

- admitting air into the column (through an orifice) after shutdown so that, upon start-up, the air is exhausted slowly and acts as a cushion; plus
- adding a surge relief valve to bypass water until a pump-control valve (or a check valve) opens; plus
- constructing stout well system piping.

A deep well pumping system is shown in Figure 7-9 together with a typical arrangement of valves. It is assumed that a hydraulic transient analysis has been performed and the need for a surge relief valve established for the particular application. For clarity, the bypass piping containing valves E and D is shown schematically as vertical. Physically, this equipment would probably be installed horizontally. The function of the valves is discussed in the following subsections.

Air release and vacuum relief valve. Valve B removes air during start-up and introduces air after shut-down to avoid a vacuum. The valve must be sized so that optimum air cushioning prevents slam. A good way is to add a throttling device for flexibility in adjustment.

Check valve. Valve C prevents reverse flow if (1) the pump shaft breaks, (2) a failure in the electrical system shuts off the pump with the pump-control valve energizes and opens, or (3) the power fails. Applicable types of check valves include silent, double door, cushioned swing, and tilting disc. By placing the check valve at the location shown, the transmission main is protected from water hammer by the relief valve.

Surge relief valve. Valve E limits excessive pressure during start-up or when the check valve closes with the pump-control valve open. Hence, lighter-weight equipment and fittings can be used. The surge relief valve allows the first part of the well water (which may contain sand and entrapped air) to be exhausted. The valve supplies a free flow of water for cooling submersible motors if the pump-control valve fails to open, and also prevents free discharge and extremely high head, both of which are detrimental to the pump. The pressure setting should be about 10% higher than the normal pumping pressure, and the valve should be sized to handle the full pump flow. Either globe or angle body valves are suitable.

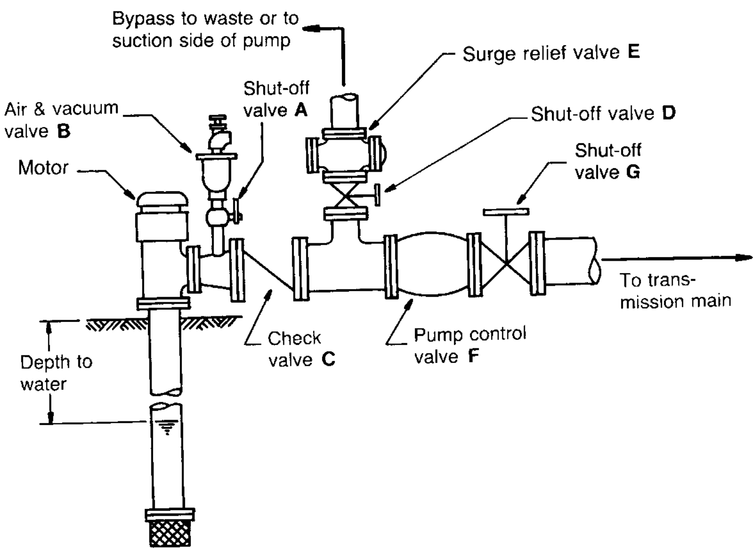


Figure 7-9. Valves for a deep well pumping station with an upstream bypass. After HY/Con Valve Co. [5].

Pump-control valve. Valve F opens and closes slowly to limit surges in the transmission main. An electric timer (or an alternate system) should open it only after all of the air (or undesirable water) has been wasted. On shut-down, the pump is kept running until the valve is about 95% closed.

A globe valve provides good protection against surge and is easy to adjust and to service, but it has the greatest headloss (although it can be sized for "acceptable" headloss).

A butterfly valve has low headloss and is available in large sizes. If repairs are required, however, it must (unlike the globe valve) usually be removed and sent to the factory. Butterfly valves are not often used for pump-control service. Because the curve of flow versus stroke is poor, eccentric plug valves should not be used for controlled closure. Ball valves are excellent because of their good throttling characteristics and low headloss when wide open, but in large sizes they are expensive.

Shut-off valves. Always install enough shut-off valves (A, D, and G in Figure 7-9) so that other equipment can be repaired without draining the entire system. However, valves A and D in Figure 7-9 are unnecessary because, whenever the pump is off, all the piping upstream from the transmission main can be drained by stopping the pump and closing valve G. Valves A and D are actually hazardous (which should be noted in the O&M manual) because if valve A is inadvertently left closed, the air and vacuum relief valve cannot function, and if valve D is closed, there is no protection against water hammer.

Safety features. Provide controls (1) to shut off the pump if the pump-control valve, F, does not open, and (2) to close the pump-control valve and shut off the pump if there is sustained low pressure, which would indicate a pipe break. Always include manual controls to override electrical malfunction of the automatic equipment.

Control-valve bypass. The valve arrangement of Figure 7-10 is superior because it provides better protection against upsurge than the valving of Figure 7-9. The surge protection on pump start-up is equally effective, and because the pump-control valve closes in anticipation of pump shut-down, the bypass arrangement limits the transmission main pressures to the relief valve setting. The configuration of piping and valves in Figure 7-10 is particularly preferred if the pump is to start and stop frequently (several times per day).

The pump-control valve, E, must have a fast-closing feature so that it acts as a check valve if the power fails or a pump shaft breaks. If a downsurge could occur as a result of power failure, another vacuum relief valve (with a large opening) may be needed between valves C and F. The orifice in the air and vacuum valve A must be carefully sized to prevent excessive upsurge during pump start-up.

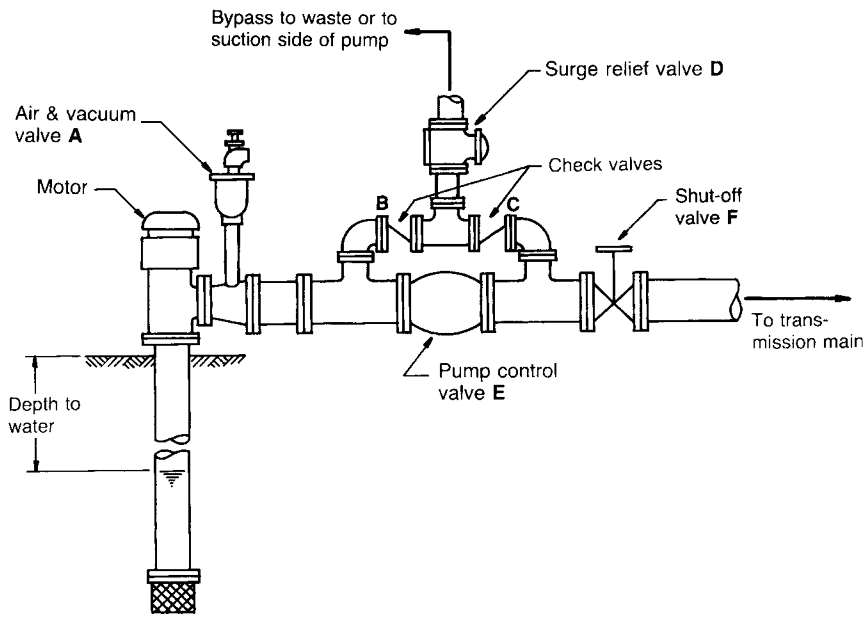


Figure 7-10. Upstream and downstream bypass of a pump-controlled valve. After Hy/Con Valve Co. [5].

The waste pipeline must have a positive air break before discharging into a sewer or drain line to prevent a cross-connection and possible contamination of the well.

The surge relief valve should be protected from excessive wear. So, if the well water contains sand, the system shown in Figure 7-11 is advantageous because the first portion of the discharge (which usually carries the most sand) is vented through valve B. As the surge relief valve, D, is never opened (unless a surge occurs as a result of power failure), the valve must be exercised periodically (say, weekly). The exercising should be part of the pump monitoring program and so specified in the O&M manual. The blow-off valve, B, can be a manual type, or a small pump-control valve can be substituted if pump starts are to be automatic. The pump-control valve, C, must be equipped with a fast-closing feature to prevent reverse flow following power failure.

A still simpler system is shown in Figure 18-13. Whether the protection provided by the valves of Figures 7-9, 7-10, and 7-11 is required depends on the static head, the length and size of the force main, the depth to groundwater, and the inclination of the designer toward conservatism.

Piping flexibility. Although not shown in Figures 7-9, 7-10, and 7-11, a sufficient number of flexible couplings specific to the installation are necessary for properly joining fittings and connections. Considerations include:

- Adjustment for potential misalignment and for preventing strain in rotating equipment.
- Sufficient freedom to permit convenient removal of the pump and other high-maintenance equipment without the necessity for removing other valves or fittings.
- Resolution of hydraulic thrusts generated during various operating modes.

Flexible piping connections that can serve the above purposes include sleeve couplings (such as the Dresser® Style 38), the flanged coupling adapter (such as Dresser® Style 128), and the grooved-end coupling (such as Victaulic® Style 77). Designers should be aware that two couplings are usually required for effectiveness in coping with misalignment and eliminating piping strain.

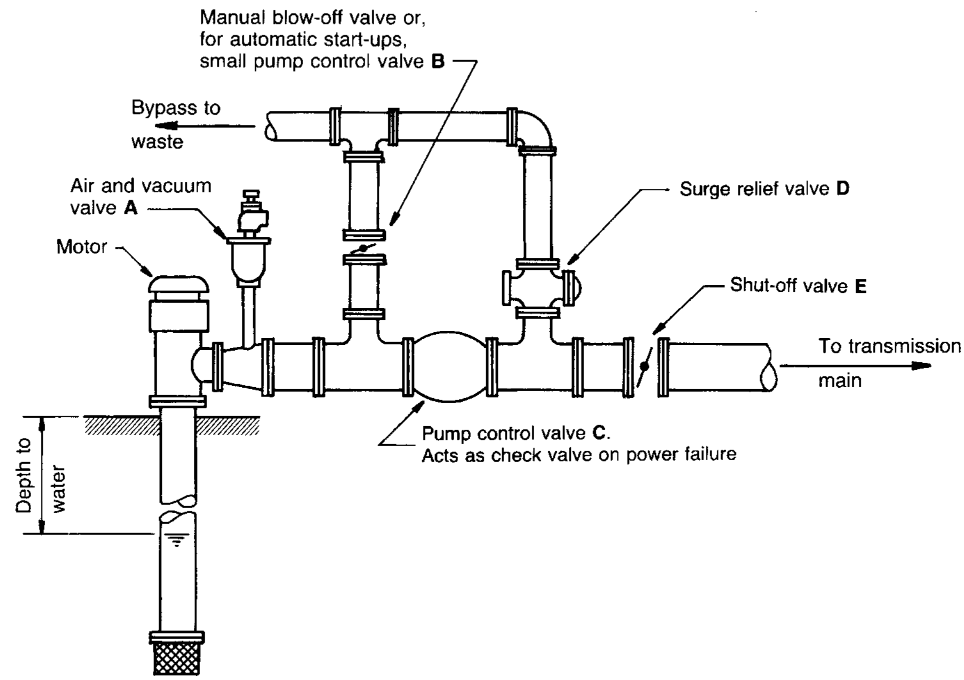


Figure 7-11. Bypass of pump-control valve for sandy water applications. Gate valves are preferred over butterfly valves in the smaller sizes (< 250 mm or 10 in.). Butterfly valves should have gear actuators to prevent excessively rapid closure. Install enough flexible couplings for fitting up. After Hy/Con Valve Co. [5].

Low Suction-Lift Turbine Pump

If the depth to water level is only a few feet (as it might be for a shallow well or for pumping from a clear well or forebay), the same arrangement of valves shown in Figure 7-9 can be used with the following modifications.

Air release and vacuum relief valve. Valve B can be smaller and the throttling device can be omitted because the volume of air to be exhausted is small. In fact, an air release valve might be used in lieu of the air-vac valve.

Pump-control valve. The opening of valve F need not be delayed because the volume of air to be exhausted is small and (if the supply is a clear well or forebay) the initial quality of water is good.

Safety feature. A low-level cutout switch should be located in a clear well or forebay.

Air Chamber with Turbine Pump

For wells without troublesome sand or silt in the discharge, an air chamber may sometimes be the appropriate surge-arresting facility. On pump start-up, air flows through an air vacuum valve and into the air chamber from which excess air is vented through an air release valve. When the pump is shut off, a vacuum breaker allows the water in the pump to drain into the well. If the depth to the water table is large enough, the air in the pump column can be used to maintain the air charge in the hydropneumatic tank, which eliminates the need for an air compressor and level controls.

Turbine Booster Pump

Because the major difference between a well pump and a booster pump is the continuously flooded suction of the booster pump, no air needs to be exhausted upon start-up.

Air release valve. Vacuum relief is not needed, so valve B in Figure 7-9 or 7-10 need only be a small air release valve.

Check valves. The purpose of the check valve is to prevent reverse flow through the pump in the event of a power failure while the pump-control valve remains open. However, pump-control valves that close quickly in the event of a power failure can also serve as check valves.

Relief valve. The relief valve is still needed for several of the reasons given in the subsection entitled "Deep Well Pump" in this section, but some of those reasons obviously do not apply to booster pumping.

Pump-control valve. It is necessary to open (and close) the pump-control valve slowly but, because the system is always full of water, unnecessary to delay its opening.

Safety features. In addition to those listed in the subsection entitled "Deep Well Pump," a pressure switch may be desirable to keep the pump-control valve shut until the pressure developed by the pump exceeds the pressure in the transmission main.

Centrifugal High Service or Booster Pump

The same valve arrangements can be used with centrifugal pumps with flooded suctions for both high service and booster pumping. However, the air release valve should be located at the top of the pump casing and should be equipped with a check valve to prevent air from returning to the casing. A manual petcock on the pump casing is usually adequate if the suction is always flooded.

7-7. Surge Control for Raw Wastewater Pumping Stations

The strategies for controlling surges in raw wastewater pumping are limited to those given in Section 7-1. Those transient control strategies are, however, adequate for nearly all situations. Of course, any proposed solution should be checked by a computer analysis.

Some additional comments on surge control are presented below.

Quick-Closing Check Valve

If the pump spin-down time as determined by computer analysis is sufficiently long (say, more than four or five t_c) and there are no knees, a quick-closing check valve is usually sufficient to prevent surges. Such conditions are common with force mains not much longer than 1 km (0.6 mi) and with uniform gradients of less than about 4%.

Pump-Control Valve

If a pump-control valve is used so that the pump starts and stops against the closed valve, it should be programmed to ease the pump into service and to close on a predetermined rapid initial closure/slow final closure to prevent high pressure spikes on

pump shut-down or power failure. A computer analysis should be used to program the closure time.

The pump-control valve can be an eccentric plug or a lubricated plug, but cone and ball valves have the best characteristics for this service and are preferred even at their higher cost.

If a power failure occurs, a stored-energy system must be used to activate the pump-control valve. But unless the valve closes quickly, the water runs backward through the pump, so the pump, impeller, and wet well systems must be able to handle the backward-flowing wastewater.

Increasing Rotational Inertia

If column separation can occur, first investigate the increase of inertia (WR^2) of the moving parts in the pump and motor. If the required additional WR^2 is not too great, a flywheel is a solution with the utmost reliability (see Section 7-2).

Other Control Strategies

If none of the foregoing can be used to control surge, consider the other strategies outlined in Section 7-1.

7-8. Pipeline Design

Selecting the wall thickness to withstand the pressures expected in the operation of a pipeline is crucial. Wall thickness depends on the pressures due to normal operation, upsurge, and downsurge as well as the pipe material and its safety factors. The safety factors are governed by user groups (such as the American Water Works Association), industries (such as the American National Standards Institute, Inc.), or by manuals of practice such as AWWA Manual M11 [6]. In some standards, a specific overpressure allowance is required, whereas in others a safety factor to be applied to the yield strength of the pipe material is given.

Upsurge

The required wall thickness for positive line pressures for normal upsurge is

$$e = \frac{PD(SF)}{2s_y E_j} \tag{7-2}$$

where e is pipe wall thickness in meters (inches), P is internal pressure in Newtons per square meter (pounds per square inch), D is outside diameter in meters (inches), SF is the safety factor, s_y is the yield strength in Newtons per square meter (pounds per square inch), and E_j is longitudinal joint efficiency associated with the effective strength of the weldment.

The allowable stress, s, may either be specified as a percentage of the yield strength or as s_y/SF. Typical values of SF for the water industry are given in Table 7-2. The longitudinal joint efficiency, E_j, is determined by the type of weld or it may be considered to be accounted for in the allowable stress specification (see ANSI B31.1). The joint efficiency factors in Table 7-3 are used primarily with welded steel pipelines.

Standards for materials of construction often include an allowance for pressure due to hydraulic transients. For example, "ordinary" surge pressures are incorporated in the safety factor of 4.0 for ACP, which conforms to AWWA C400 for internal pressure in combined loading, but the term "ordinary surge

Table 7-2. Typical Safety Factors (SF) for Pipe

	Type of pipe			
Pressure condition	Ductile iron	Steel	PVS	AC
Maximum operating	2.0	2.0	2.0–2.5	2.0–4.0
Upsurge transient	a	1.5	a	a
Downsurge collapsing	4.0	4.0	None	None

[a]Include it in maximum operating pressure.

Table 7-3. Weld Joint Efficiencies for Steel Pipes (per ASME B31.1)

Type of longitudinal joint	Weld joint efficiency factor (E_j)
Arc or gas weld	
Single butt weld	0.80
Double butt weld	0.90
Single or double butt weld with 100% radiography	1.00
Electric resistance weld	0.85
Furnace butt weld	0.60
Most steel water pipelines	0.85
Ductile iron (cast—has no longitudinal welded joints)	1.0

pressure" is not defined. A maximum value of 350 kPa (50 lb/in.^{2}) in addition to the design operating pressure is typical. For higher surge pressures, the pipe wall thickness should be increased. A design procedure for including such exceptional surge pressures in the design of ACP is described in AWWA C401.

According to AWWA C151, ductile iron pipe up to 450 mm (18 in.) is adequate for a rated working pressure of 2400 kPa (350 lb/in.^{2}) plus a surge allowance of 690 kPa (100 lb/in.^{2}). For larger pipes, the operating pressure varies with wall thickness class, although the surge allowance of 690 kPa (100 lb/in.^{2}) remains the same.

Downsurge

The minimum wall thickness required for protection against negative gauge pressures that tend to buckle the pipe often dictates the design in low-pressure systems where vapor cavities may form in high points of pipelines subject to downsurge conditions. The negative gauge pressure required to collapse a circular pipe of uniform wall thickness is

$$\Delta P_c = P_{\text{atm}} - P_{\text{v}} = \frac{2E}{(1-\mu^2)(SF)}\left(\frac{e}{D}\right)^3 \qquad (7\text{-}3)$$

where ΔP_c is the difference between the external and internal pressures on the pipe, P_{atm} is the atmospheric pressure, P_v is the vapor pressure of the liquid inside the pipe, E is the modulus of elasticity, μ is Poisson's ratio, SF is the safety factor, e is the wall thickness, and D is the outside diameter. Equation 7-3 is reasonably accurate for ductile iron and steel. However, because of end effects, wall thickness variation, lack of roundness, and other manufacturing tolerances in steel pipe, use

$$\Delta P_{\text{t}} = P_{\text{atm}} - P_v = \frac{C}{SF}\left(\frac{e}{D}\right)^3 \qquad (7\text{-}4)$$

where C is 3.45×10^8 kPa (5.0×10^7 lb/in.^{2}). Safety factors, SF, are given in Table 7-2.

Several other important pipeline design considerations that are omitted in Example 7-1 include

- Soil loading on buried pipes
- Additional reinforcement or thickness at branches and openings
- Internal or external corrosion allowance
- Temperature effects on the steel stress values
- Effect of column separation and rejoining.

Example 7-1
Determination of Minimum Pipe Wall Thickness

Problem: Find the required minimum wall thickness for a steel water transmission line 500 mm (20 in.) in diameter with the following characteristics:

- Pipe material specification: ASTM A 53, Type E, Grade A
- Yield strength = 210,000 kPa (30,000 lb/in.^{2})
- Maximum operating pressure = 1750 kPa (250 lb/in.^{2})
- Maximum pressure due to surge (static + dynamic + transient rise) = 2500 kPa (360 lb/in.^{2})
- $P_{\text{atm}} = 94.5$ kPa (13.7 lb/in.^{2})
- $P_{\text{v}} = 1.7$ kPa (0.26 lb/in.^{2})
- Longitudinal joint efficiency = 0.85

Solution: Find the required wall thickness based on three criteria: (1) maximum operating pressure, (2) maximum transient pressure, and (3) collapsing pressure.

Maximum operating pressure. From Equation 7-2 and for a safety factor of 2 from Table 7-2.

SI Units

$$e = \frac{1750 \times 500 \times 2}{2 \times 210{,}000 \times 0.85} = 4.90\,\text{mm}$$

U.S. Customary Units

$$e = \frac{250 \times 20 \times 2}{2 \times 30{,}000 \times 0.85} = 0.196\,\text{in.}$$

But the tolerance on wall thickness in ASTM A53 pipe is 12.5%, so increase e by 12.5%:

$e' = 1.125 \times 4.90 = 5.51\,\text{mm}$ $\qquad$ $e' = 1.125 \times 0.196 = 0.221\,\text{in.}$

Maximum transient pressure. The safety factor from Table 7-2 is 1.5 for the upsurge pressure.

$$e = \frac{2500 \times 500 \times 1.5}{2 \times 210{,}000 \times 0.85} = 5.25\,\text{mm} \qquad e = \frac{360 \times 20 \times 1.5}{2 \times 30{,}000 \times 0.85} = 0.212\,\text{in.}$$

Again, increase e by 12.5%:

$e' = 1.125 \times 5.25 = 5.91\,\text{mm}$ $\qquad$ $e' = 1.125 \times 0.212 = 0.238\,\text{in.}$

Collapsing pressure. Find the safety factor from Table 7-2 and rearrange Equation 7-4 to solve for e:

$$e = D\sqrt[3]{\frac{SF}{C}(P_{atm} - P_v)}$$

$$e = 500\sqrt[3]{\frac{4}{3.45 \times 10^8}(94.5 - 1.7)} = 5.12\,\text{mm} \qquad e = 20\sqrt[3]{\frac{4}{5 \times 10^7}(13.7 - 0.26)} = 0.205\,\text{in.}$$

Increase e by 12.5%:

$e' = 1.125 \times 5.12 = 5.76\,\text{mm}$ $\qquad$ $e' = 1.125 \times 0.205 = 0.231\,\text{in.}$

The required wall thickness, 5.91 mm (0.238 in.), is governed by the maximum transient pressure of 2500 kPa (360 lb/in.^{2}). Pipe conforming to ASTM A53 is built according to the dimensions given in ASME B36.10, and the thinnest applicable wall is Schedule 10—6.35 mm (0.250 in.).

7-9. Computer Analysis

The objectives of this section are (1) to show the value of computer analysis, and (2) to provide insight into the effectiveness of various surge control devices. Refer to Watters [3] for extensive treatment of air chambers alone and in conjunction with one-way tanks, and see Watters [3], Chaudhry [7], and Wylie and Streeter [8] for the details of computer analysis, as these are beyond the scope of this book. Several commercial computer programs are available for hydraulic transient analysis.

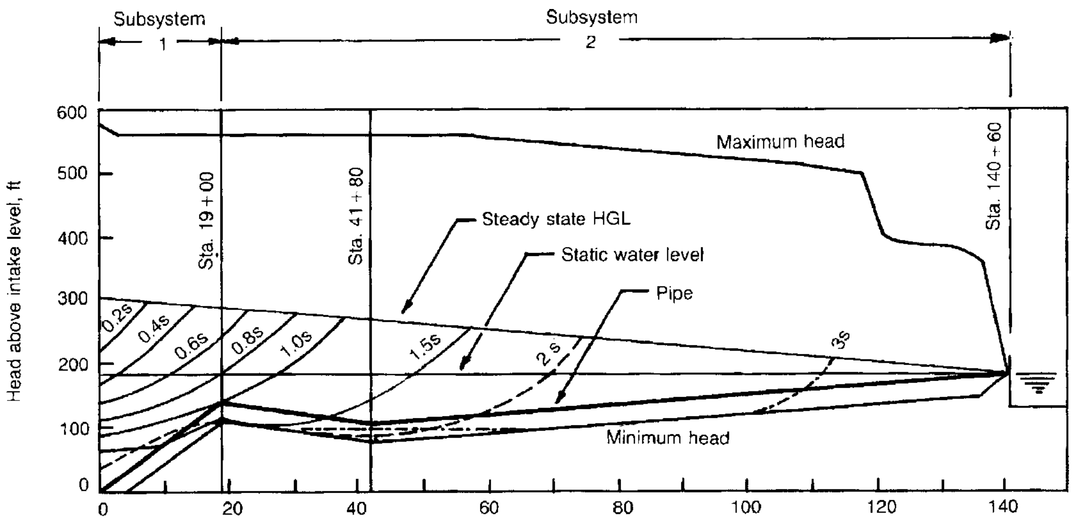
An example of a water pumping system is shown in Figure 7-12a. The heavy line represents the profile of a ductile iron pipe 4285 m (14,060 ft) long with an ID of 420 mm (16.5 in.). Two pumps each discharge 386 m^3/hr (1702 gal/min) through 300-mm (12-in.) quick-closing check valves into the 420-mm (16.5-in.) manifold. The static lift is 57 m (186 ft) and the initial TDH is 93 m (305 ft). The system requires a computer analysis for several reasons (which are given in Section 6-8):

- The flow exceeds 6 L/s (100 gal/min).
- The head is greater than 9 m (30 ft).
- The system contains a knee and the initial gradient is excessively steep.
- The velocity exceeds 1.2 m/s (4 ft/s).

The system was modeled with a program utilizing the method of characteristics on a mainframe computer, which made a complete solution of pressure and velocity at 0.05-s intervals. Column separation would occur in the unprotected pipeline, so surge control measures are necessary.

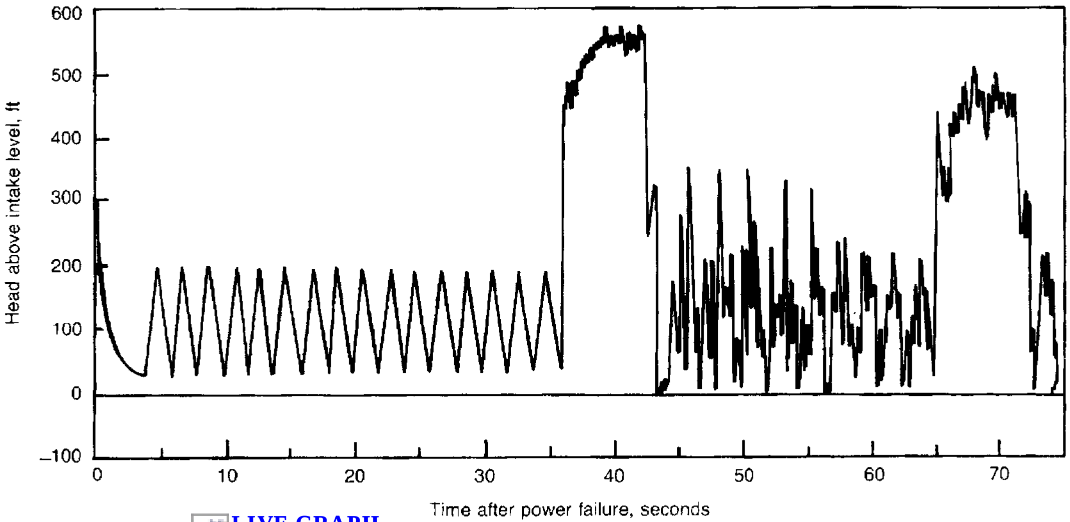
During the filling of the pipe, an air release valve is needed at station 19+00 to expel trapped air and to release air bubbles that inevitably accumulate during operation. After the pipe is filled, it will stay filled (except for vapor cavities if column separation occurs) because the knee is below the static water level.

When power fails at time zero, the pumps stop quickly (within 6 s) because of the initially steep pipeline gradient. Successive hydraulic gradelines are shown at intervals labeled 0.2, 0.4, ..., 3 s in Figure 7-12a. At 1.0 s, the HGL intersects the pipeline at station 19+00, so the pressure drops to zero (note that a "station" is 100 ft). At 1.5 s, the HGL is below the pipeline for a considerable distance, and the gauge pressure falls to the minimum possible—vapor pressure at −291 kPa (−14.5 lb/in.^{2}) as referenced to zero at atmospheric pressure. (Depending on the elevation and water temperature, the negative pressure may be slightly less.) At 3 s, the water is literally boiling and forming vapor pockets over nearly all of the pipeline.



LIVE GRAPH
Click here to view

a



LIVE GRAPH
Click here to view

b

Figure 7-12. Effect of a power failure on an unprotected pipeline. (a) Pressure head along the pipeline; (b) pressure head at pump (station 0+00). Courtesy of Stoner Associates, Inc. (now Advantica, Inc.) [1].

The largest vapor pocket forms at station 19+00 because of the knee, and that splits the pipeline into two hydraulically separate subsystems: (1) stations 0+00 to 19+00, and (2) stations 19+00 to 140+00. For as long as a vapor cavity exists at station 19+00, it acts like a reservoir with a constant pressure of −101 kPa (−14.7 lb/in.^{2})—actually less because of vapor pressure. The steep gradient and high gravity force acting on subsystem 1 rapidly bring the column of water to rest and stop the pumps, and the spring-loaded check valves close before reverse flow can occur. Pressure oscillations travel back and forth between the closed check valves and the constant head source at station 19+00. The frequency of the oscillations should be $4L/a = 4 \times 1900/3800 = 2\,\text{s}$, and, of course, that is evident in Figure 7-12b.

The slope of the pipeline in subsystem 2 is gentle, so there is less force to halt the fast-moving column. Flow continues for a relatively long period but, eventually, it does reverse, and when the large vapor pocket at station 19+00 collapses, the reverse flow suddenly stops with a shock that increases the head to about 550 ft for 7.5 s, the time required for a wave to travel to the reservoir and be reflected back at the elastic speed of 1160 m/s (3800 ft/s) (see Figure 7-12b). Other vapor pockets also collapse and cause a jumble of peaks in the pressure trace. The energy generated by the vapor cavity collapse is large enough to create a rebound effect in subsystem 2 and open another vapor cavity at station 19+00. So, about 65 s after power failure, there is another sudden jump in pressure caused by the collapse of the new vapor cavity. Cavities may open and close several more times before friction dissipates the energy.

To minimize the high pressures, the violent collapse of the vapor cavity at station 19+00 must be prevented. The best defense is to prevent vapor formation in the first place, and the simplest means (assuming it is completely impractical to eliminate the knee) is to install a vacuum breaker valve and an air release valve at the knee. The sizes required are found by trial to be 50 mm (2 in.) for the vacuum breaker and 6 mm (¼ in.) for the air release valve. The vacuum breaker valve admits air into the pipe when the gauge pressure becomes slightly negative and, thus, prevents the high negative pressure that produces vapor. Of course, a cavity still forms at the knee, but now it is composed of air and not vapor. Furthermore, in contrast to the unprotected pipeline—where the head to rejoin the two water columns is a constant differential of 14 m (46 ft) between reservoir and knee plus 10.2 m (33.5 ft) due to the vacuum at the knee—the head for flow reversal now only starts at this value and soon diminishes because as the cavity shrinks, the air is compressed and this air is emitted so slowly by the small air relief valve that contact is the soft, cushioned event shown in Figure 7-13b. Again, the periodic pressure oscillation with the 2-s frequency is caused by the reflection of the pressure wave between the closed check valves and the air pocket at station 19+00. The returning flow in subsystem 2 raises the oscillating pressure slightly with a peak at about 62 s.

The vacuum breaker valve at station 19+00 does not entirely eliminate downstream vapor cavities, although it does keep them small. Elimination of all downstream vapor cavities along flat pipelines requires vacuum breakers spaced at reasonable intervals. Note the further reduction of maximum pressures shown in Figure 7-13a with additional sets of vacuum breaker-air release valves at stations 81+70 and 110+20.

Although an air chamber at the pumping station is often effective, here it does not prevent formation of a vapor cavity and its subsequent collapse at station 19+00, as shown in Figure 7-13c. The air chamber does dampen oscillations between check valves and the knee, and it absorbs some of the shock so that the pressure spike is very narrow. But it does little for the pipe downstream from station 19+00, as can be seen by comparing Figures 7-12a and 7-13a.

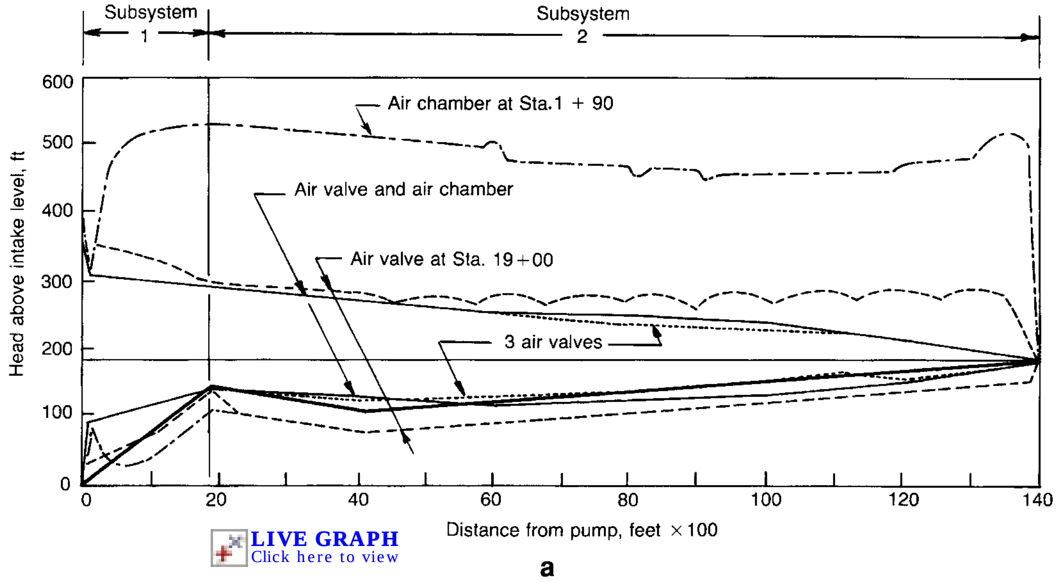
In the mathematical model of this example, the air chamber can be placed no closer to the pumps than station 1+90 owing to the details and limitations of the model. In reality, the air chamber can be placed anywhere inside or outside of the pumping station with results little different from the model.

The use of both an air valve and an air chamber (Figure 7-13d) is beneficial in reducing the oscillations of Figure 7-13b. Together they offer very effective protection from water hammer.

An air chamber placed at station 19+00 might prevent column separation altogether, but because it would have to be very large, it might be uneconomical. Whether the air chamber would work at all, though, is uncertain and depends on the profile of the pipe. A computer analysis would have to be used to determine the size and the effectiveness. The profile in Figure 7-12a is such that the simpler and less expensive air release, vacuum relief valve offers acceptable of protection for a minimum cost.

7-10. Transients in Distribution Systems

As an example of how to identify and characterize possible transient conditions in a distribution system,



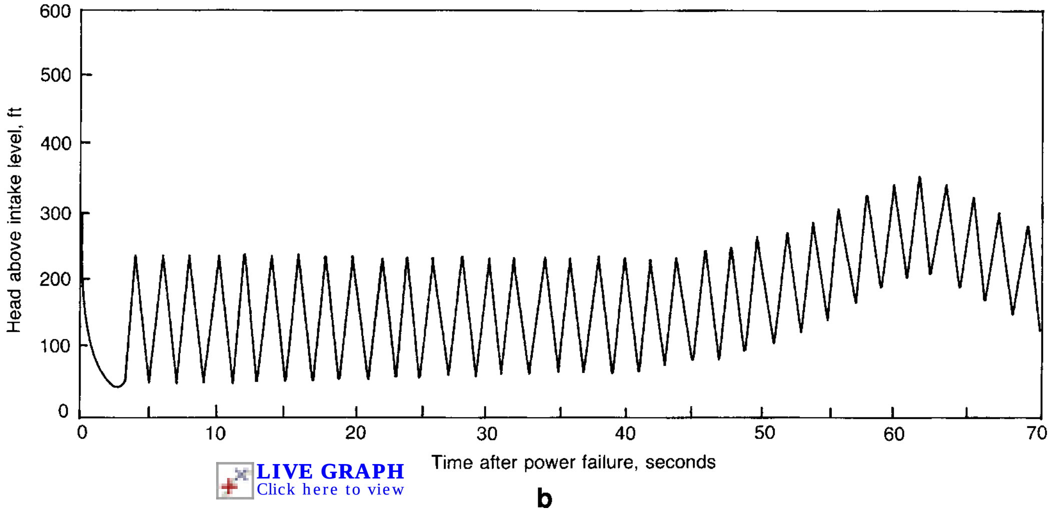
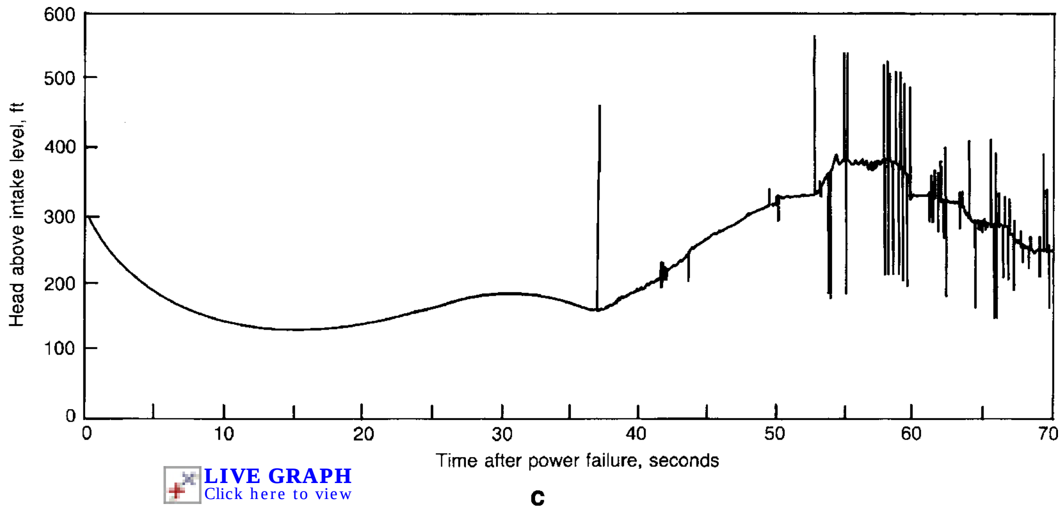
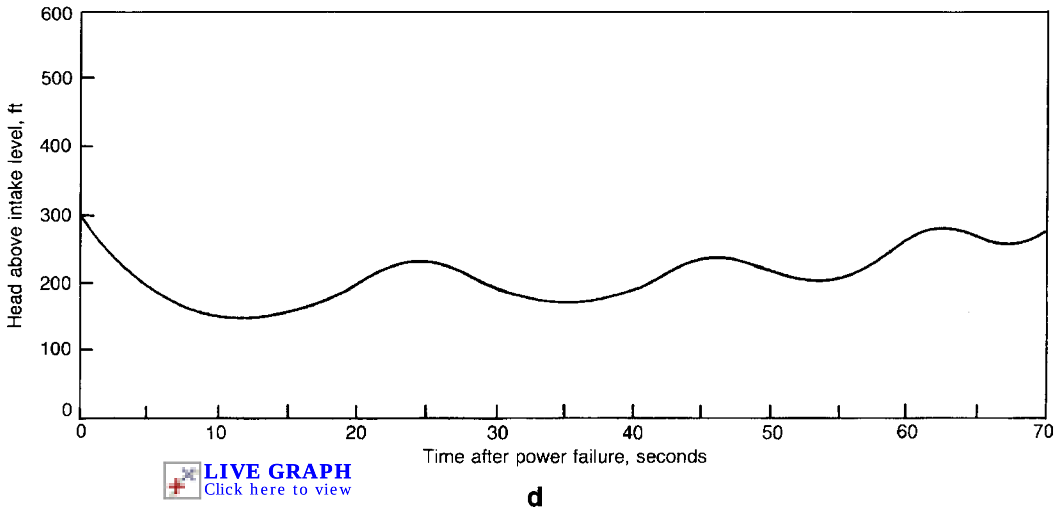


Figure 7-13. Effect of a power failure in a pipeline with surge control devices. (a) Maximum and minimum pressure heads along a pipeline with surge protection; (b) pressure head at pump (station 0+00) for a pipeline with a 2- × ¼-in. air valve at station 19+00.

consider the pumping station and pipeline system shown schematically in Figure 7-14. Each item, such as (1) a pump, (2) an interconnection of two or more pipes, (3) an end of a pipe, or (4) a reservoir is known as a boundary condition. Each boundary condition constitutes a node as shown by circles in the figure. The system shown consists of a source pump with forebay, two in-line booster pumping stations, and several turnouts with solenoid-operated valves through which water flows. The valves, which are of the energize-to-open, de-energize-to-close design, could close completely in 1 min. Each valve also has a manually operated isolation valve immediately upstream.



c



d

Figure 7-13. *(Continued).* (c) Pressure head at pump for a pipeline with an air chamber at station 1+90; (d) pressure head at pump for a pipeline with an air chamber at station 1+90 and a 2- $\times \frac{1}{4}$-in. air valve at station 19+00.

Possible causes of transient conditions are:

- Manual closure of one or more isolation valves.
- Power failure at one pumping station or simultaneously at several pumping stations.
- Shut-off of one solenoid valve at power failure or simultaneous shut-off of several (or all) solenoid valves.
- Combination of pump power failure plus a simultaneous closing of all solenoid valves.

Although any of these conditions—and more—may be possible, sometimes it is unlikely or even impossible for a particular condition to occur. For example, sudden, simultaneous closure of more than one isolation valve is extremely unlikely. Because neither time nor resources permit analyzing all the possibilities, the transient conditions *most likely* to cause the worst problems must be identified, as opposed to all *possible* conditions.

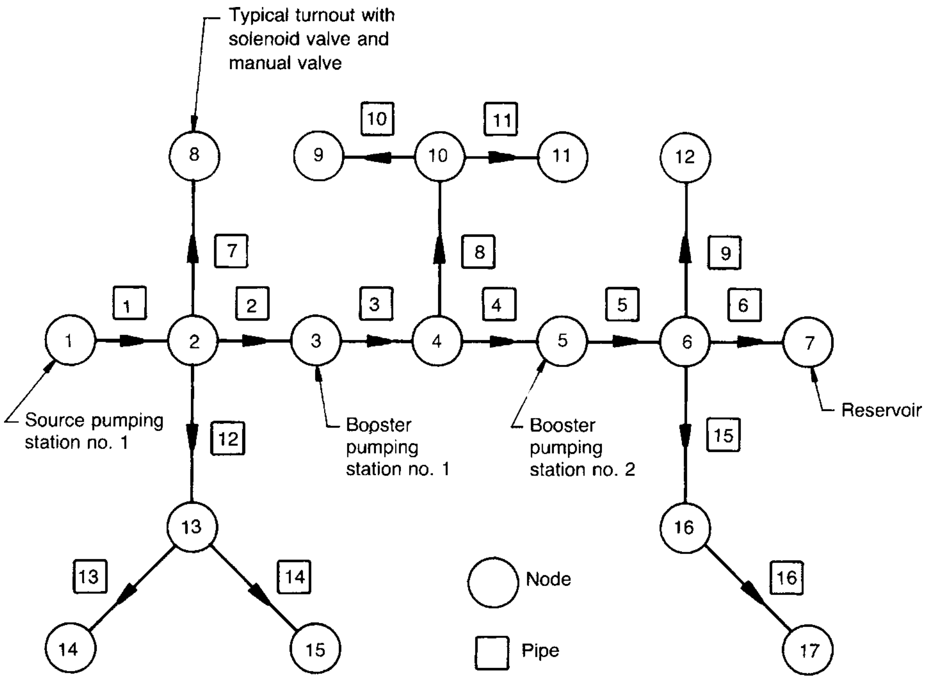


Figure 7-14. Schematic diagram of a pumping system.

The profile of the transmission line can also give clues about possible transient conditions as discussed in Sections 6-7 and 7-9.

7-11. References

1. Stoner Associates (now Advantica, Inc.), 5177 Richmond Ave., Ste 900, Houston, TX 77056, www/advantica.biz.
2. Vent-O-Mat® International Valve Marketing, Inc., P.O. Box 56, Plainfield, IL, 60544, email limeym@aol.com.
3. Watters, G. Z., *Analysis and Control of Unsteady Flow in Pipelines*, 2nd ed., Butterworth-Heinemann, Stoneham, MA (1984).
4. American Society of Mechanical Engineers, *ASME Boiler and Pressure Vessel Code, Section VIII*, United Engineering Center, New York (2001).
5. "Surge protection for water pump stations," Hy/Con Valve Co., Inc., Schenectady, NY (n.d.).
6. AWWA M11, *Steel Pipe: A Guide for Design and Installation*, 3rd ed., American Water Works Association, Denver, CO (1989).
7. Chaudhry, M. H., *Applied Hydraulic Transients*, Van Nostrand Reinhold, New York (1979).
8. Wylie, E. B., and V. L. Streeter, *Fluid Transients,* Feb Press, Ann Arbor, MI (1982, corrected copy 1983).

Chapter 8

Electrical Fundamentals and Power System Principles

STANLEY S. HONG
PHILIP A. HUFF
PAUL C. LEACH

CONTRIBUTORS
Robert S. Benfell
Mayo Gottliebson
Stephen H. Palac
Evans W. Paschal
Richard N. Skeehan
Patricia A. Trager
Alan Vause

This chapter is intended primarily for those project managers who have limited knowledge of electrical fundamentals but must nevertheless develop rapport and communication skills with electrical designers. The fundamentals of elementary electrical theory are reviewed in Section 8-2, power and control systems in Section 8-3, generators in Section 8-4, grounding and ground fault protection in Section 8-5, lighting and power outlets in Section 8-6, circuit diagrams in Section 8-7, and power and control system practice in Section 8-8. Although design principles are given herein, design per se is covered in Chapter 9.

References to books are given in standard form, but references to specifications, standards, and codes are given in abbreviated form, such as NEC (for National Electrical Code) or NFPA 78 (for National Fire Protection Association Lightning Protection Code).

Project engineers should realize that all electrical design must be done or supervised by a professional electrical engineer who is licensed in the state where the project is located. All work should be checked by another qualified electrical engineer. Furthermore, project engineers must be reasonably familiar with:

- Electrical codes
- Definitions (see Section 8-1, Chapter 2, and NEC)
- Electrical symbols (see Tables 2-5 and 2-6)
- Single-line diagrams (see Section 8-7)
- Power and control system elements and practices (see Section 8-8)
- State and federal regulations pertaining to energy conservation—especially lighting
- Requirements of the local building codes
- Environmental Protection Agency (EPA) design guidelines.

8-1. Definitions and Code References

Refer to the latest revision of the NEC and ANSI/IEEE 100 for definitions of electrical terms. See also Section 2-2 and, for voltage terminology, Section 8-8.

The following are some commonly used terms in pumping station design. The sources of quoted definitions are given in brackets.

Ampacity: The current in amperes that a conductor can carry continuously under the conditions of use without exceeding its temperature rating [NEC].

Branch circuit: The circuit conductors between the final overcurrent device protecting the circuit and the outlet(s) and the utilization equipment such as motors, lights, etc. [NEC].

Circuit breaker: A device designed to open and close a circuit by nonautomatic means and to open the circuit automatically on a predetermined overcurrent without damage to itself.

Continuous load: A load in which the maximum current is expected to continue for 3 hours or more [NEC].

Controller: A device or group of devices that governs the electrical power delivered to the utilization equipment in a predetermined manner. A motor controller includes any device, such as a switch or contactor, normally used to start and stop a motor and may include equipment (such as a reactor, resistor, or solid-state system) that limits motor starting current [based on NEC].

Current withstand rating: The maximum allowable current, either instantaneous or for a specified period of time, that a device can withstand without damage [ANSI 100].

Fault: An unintended connection (a short circuit) between phases or between a phase and ground. A fault may result in an excessive current in the faulted equipment, in its supply conductors, and in the supply system itself.

Feeder: All circuit conductors between the service equipment or the source of a separately derived system and the final branch circuit overcurrent device [NEC].

Frequency: The number of periods (or cycles) per unit time.

Hertz: The unit of frequency in cycles per second, abbreviated Hz.

Motor control center (MCC): An assembly of grouped control equipment used primarily for control of motors and associated power distribution applications.

Overcurrent: Any current in excess of the rated current of equipment or the ampacity of a conductor. Overcurrent may result from overload, short circuit, or ground fault [NEC].

Overload: Operation of equipment in excess of the normal full-load rating (or of a conductor in excess of rated ampacity) that would cause damage or dangerous overheating when it persists for a sufficient length of time. A fault, such as a short circuit or a ground fault, is not an overload [NEC].

Power factor: The cosine of the angle by which the current lags (due to inductance) or leads (due to capacitance) the voltage.

Service: The conductors and equipment for delivering electrical energy from the supply system (usually an electric utility) to the wiring system of the premises served [based on NEC].

Switching apparatus: A device for opening and closing or for changing the connections of a circuit. It includes switches, fuses, circuit breakers, and contactors [ANSI/IEEE Std 141-1986 Art 9.2].

8-2. Electrical Fundamentals

Electricity (the movement of electrons) flows easily in conductors such as copper and hardly at all in insulators such as rubber or glass. Electrons flow in the circuit conductors from the negative pole to the positive pole of a battery, but by custom, electrical current is said to flow from the positive to negative pole. In semiconducting media and in electrolytes, both positive and negative charges flow to their respective attracting terminals.

Electricity flows when a voltage source providing electrical pressure (or force) to the electrons is in series with a complete circuit. Ordinarily, a complete circuit consists of the voltage source, conductors, and a load that transforms the electrical energy from the source into usable energy such as heat or mechanical work. A switch provides the means of making and breaking a circuit intentionally and, thus, controls the flow of electricity. If the circuit is broken at any point, electricity does not flow.

Electrical pressure (voltage) can be produced by: (1) chemical reactions (as in batteries); (2) movement of magnets near coils (as in generators); (3) contact between dissimilar metals (thermocouples, for example); (4) friction (lightning, for example); or (5) light energy impinging upon a thin semiconducting film (as in photovoltaic systems or solar cells).

Alternating current (ac) power is the most common, because it is easy to generate, transform (from one voltage level to another), and utilize. It is commonly available as single-phase or three-phase. Direct current (dc) is often used to power emergency lights and to provide control power in large systems. It is used to start engine generator sets, to power dc pump motors in adjustable-speed applications, and as field supply to synchronous motors and generators. But dc is not available from utilities. It must be produced on site.

The potential difference (the electrical equivalent of pressure) between any two points in an electric system is measured in volts (V). The unit of quantity of electrical charge is the coulomb (C). The ampere (A) is the measure of the rate at which charges move past a given point in coulombs per second. The energy or work required to move one coulomb through a potential difference of one volt is the joule (J) and the rate of expenditure of energy is measured in joules per second or watts (W).

In an alternating current system, an ampere of alternating current is defined as the measured value that causes the same heating effect as one ampere of direct current. The ampere is mathematically equal to 0.707 times the peak value of a sinusoidal alternating waveform, and it is sometimes referred to as the root-mean-square (RMS) value.

The measure of the opposition or resistance to the flow of electrical current is the ohm (Ω). In dc circuits and in unity power factor ac circuits, one volt of potential difference is required to cause a current of one ampere to flow through a resistance of one ohm—a relation known as Ohm's Law.

$$I = V/R \tag{8-1}$$

where I is current in amperes, V is volts, and R is resistance in ohms. In dc circuits, the average values for current and voltage are used. In ac circuits with sinusoidal waveforms, RMS (root-mean-square) values are used for current and voltage.

Electrical power in dc and unity power factor ac circuits is the product of current and voltage.

$$P = VI \tag{8-2}$$

where P is power is watts. In ac circuits with a power factor that is not unity, the above relation becomes more complex (Equation 8-3). Nevertheless, power is equal to the product of resistance and the square of the RMS value of current. Power usage at the rate of one watt for a period of one hour is termed one watt-hour (or 3600 J) of electrical energy.

Generation of Electricity

An electrical source as described in the foregoing discussion generates a potential difference called "voltage." The voltage may be steady, pulsing (with constant polarity), or alternating in polarity. A battery generates a steady dc voltage, as do other static sources such as photovoltaic cells and thermocouples. An ac generator produces an alternating voltage with a sinusoidal waveform. A dc generator is an ac generator with either: (1) solid-state rectifiers, or (2) a commutator that switches connections to coils on the rotating armature and thereby produces a pulsating dc voltage.

The movement of the bar magnet across the coil of wire in Figure 8-1 generates voltage because lines of magnetic force or flux move across the coil. Reversing the direction of movement reverses the voltage polarity, as from positive to negative. Alternatively, if the polarity of the magnet is reversed, the voltage polarity reverses. By rotating the magnet as shown in Figure 8-2 the voltage alternately reverses as the north and south poles of the magnet pass the coil. The voltage is produced as a sine wave, which is complete with one revolution of the magnet. If the magnet revolves at 60 revolutions per second, the voltage is produced at 60 Hz. Two magnets arranged

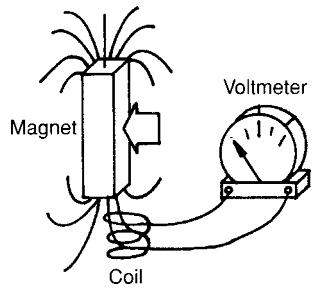


Figure 8-1. Producing potential (voltage) with a moving magnet.

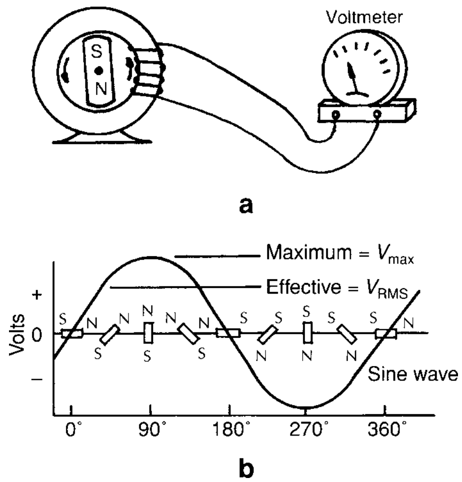


Figure 8-2. A simple ac generator. (a) Voltmeter and rotating magnet; (b) voltage produced.

with alternating polarity likewise produce 60 Hz voltage when rotated at 30 revolutions per second. In a multi-pole generator, the required rotational speed in revolutions per minute to produce 60 Hz voltage is 7200/P, where P is the number of north and south poles (always an even number).

DC generators with commutators are always arranged with the magnets on the stationary member (the field) and the coils on the rotating member to facilitate commutator construction. Most ac generators are arranged with a rotating field, and, to eliminate the slip rings that would be required to conduct the generated current from the rotating member, the coils are on the stationary member. Slip rings and commutators are a significant source of wear and maintenance.

Inductance

When current flows in a conductor, a magnetic field forms that envelopes the conductor for its full length. If the current is steady, the field is static (see Figure 8-3). The presence of the field can be detected by a small magnet such as the needle of a compass. However, if the current starts to change, the magnetic field around the conductor also begins to change, and this changing field generates a voltage that tends to oppose the change in current. This opposition to a change in current flow in a conductor is termed an "inductive effect." The effect can be increased by coiling the conductor into a helix, which increases the magnetic field along the central axis of the coil as shown in Figure 8-4. If an iron bar or some other ferromagnetic core material is inserted inside the coil, a striking increase in the inductive effect occurs, because the iron creates an easy path for the magnetic field and enhances the magnetic flux created by the current.

The unit of inductance is the henry (H). It is the inductance required to generate one volt while a current change of one ampere per second occurs. In direct current circuits the inductive effect is noticed only during changes in the level of current. When the switch is closed in an inductive dc circuit, there is a

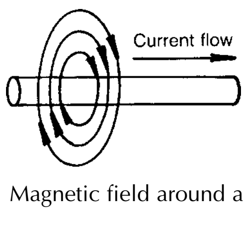


Figure 8-3. Magnetic field around a conductor.

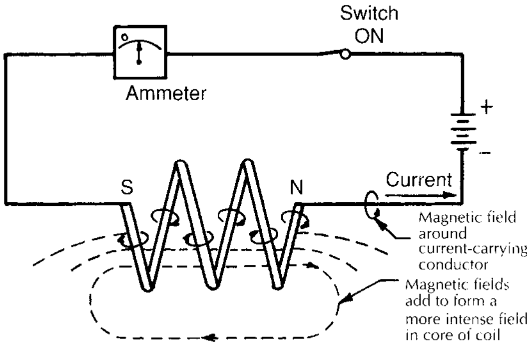


Figure 8-4. Magnetic field due to a coiled conductor.

slight time lag in the buildup of the current. Also, if a steady current is flowing in a circuit with a large coil, opening a switch and attempting to return the current to zero instantaneously causes a voltage to be induced in the coil. This voltage may be sufficient to break down the air gap insulation between the opening switch contacts and cause a momentary arc. Never be in contact with a dc circuit conductor that has a large inductive element in the circuit, because the voltage generated during switching may be several thousand volts and can cause a severe shock or burn.

In an alternating current system where significant inductance is present, the current is limited partly by the inductive effect, because the continuously changing value of the current generates a voltage in opposition to the supply voltage. From an examination sine wave shown in Figure 8-2, it can be seen that the voltage is changing at its most rapid rate as it goes through the zero axis and is not changing at all at the top and bottom of the cycle (90 degrees and 270 degrees). Thus the countervoltage generated by the inductive effect is maximum when the driving voltage is zero, and the current within the inductor is offset 90 degrees from the voltage across the inductor.

Capacitance

Capacitance is a measure of the ability of a capacitor to store an electrical charge. One type of capacitor contains a series of metal plates separated by insulators (e.g., air gaps, mica, or other insulating materials called dielectrics). Alternate plates are connected to one or the other conductor. Another common type of capacitor is a sandwich of metal and insulator sheets rolled into a cylindrical form.

Capacitance is measured in units of farads (F). Common values are measured in picofarads (mmF

or pF or 1.0×10^{-12}F); other applications call for micro-farads (μF or 1.0×10^{-6} F). In power system work, the capacitance is usually designated in terms of the reactive power or reactive kilovolt amperes (kVARs) required from an alternating current source of a specific nominal voltage.

Assume that the switch in Figure 8-5a is closed. Because the two plates labeled "capacitor" are separated by a thin insulating medium, no direct current can flow through the circuit. However, there is a short-duration flow of current in the circuit as electric charges build up on the opposing plates. Thus, capacitors are charged by a short-duration current flow. The larger the plates, the greater is their ability to store charges. No current flows through the capacitor because of the air gap or insulator.

If a capacitor is placed in a simple alternating current circuit and the switch is closed, there is a flow of current in the circuit because, as the source voltage continually reverses, the current is made to reverse and the plates are each alternately charged in alternating polarity. Because current must flow before the charges build up and potential difference exists, the capacitive current is 90 degrees out of phase with the applied voltage and leads it by 90 degrees.

Never touch capacitor terminals after opening the switch. Because of the trapped charges on the plates, a voltage exists across the terminals that may equal line voltage. For safety, discharge the capacitor by shorting its terminals. Commercial units include a small resistor across the terminals to discharge a de-energized capacitor automatically, but the discharge takes several minutes.

Inductance, capacitance, and resistance form the principal electrical elements in power systems and are collectively termed "impedance." Resistance in a circuit results in a component of the total current that is in phase with the applied voltage. Inductive reactance in a circuit draws a current that lags the voltage and capacitive reactance draws a current that leads the voltage. Because the inductive current component lags by 90 degrees and the capacitive leads by 90 degrees they are opposite in effect, and if their respective currents are equal, they are canceled. The result is a purely resistive circuit. Capacitors are used in practical power systems for cancelling inductive effects.

Power

Electrical power, the time rate of doing work, is defined in terms of the heating effect of electricity expressed in watts (W), and one W = 1.0 J/s (9.48×10^{-4}Btu). One horsepower (hp) equals 746 W. The total electrical energy used over a period of time is measured in watt-hours (W·h) or kilowatt-hours (kW·h). A kilowatt-hour equals 3413 Btu.

Direct Current Power

Direct current power is computed using Equation 8-2 from simultaneous readings of voltage and current. The average values of current and voltage are used.

Alternating Current Power

In ac circuits, power is the direct product of amperes and volts only when the load is a resistance (or a load that can be represented as equivalent to a resistance). Power can then be calculated from Equation 8-2 by using RMS values of voltage and current. In this situation, the current is in phase with the voltage and follows the same sinusoidal path with the same

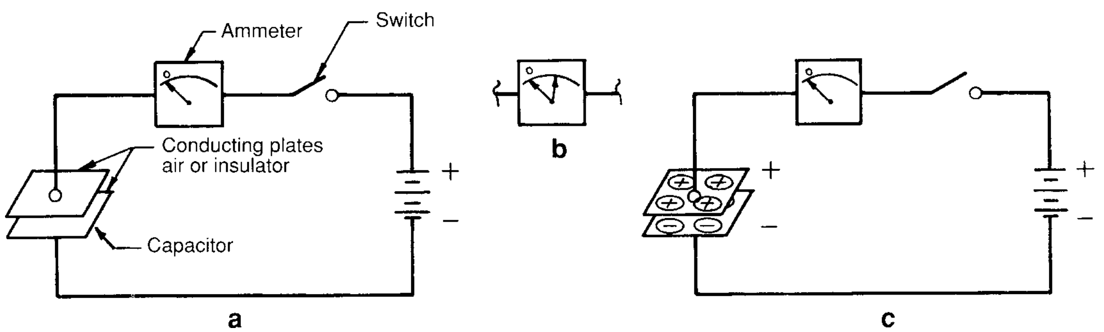


Figure 8-5. Basic capacitor. (a) Capacitor discharged; (b) switch closed and reopened, ammeter deflects during charge; (c) capacitor charged.

relative instantaneous values at all times, as in Figure 8-6.

In a circuit containing pure inductance, the product of the voltage and current is termed reactive volt-amperes. The two values are 90 degrees out of phase, so the real product (or power) averaged over a cycle is zero. In practical inductive circuits, there is always significant resistance, and the current lags the voltage by some angle less than 90 degrees as illustrated in Figure 8-7. Total true (real) power is, then, the product of the voltage, the current and the cosine of the phase angle between the voltage and current (Equation 8-3).

$$P = VI\cos\theta = VI \times Pf \qquad (8\text{-}3)$$

where P is the true power, Pf is the power factor and equals cos θ, θ is the electrical angle by which the current lags or leads the voltage, and RMS values of current and voltage are used.

If the load on the circuit is resistance and capacitance, then the same equation is valid, but the current is leading the voltage, and the cosine of the angle between the capacitance current and the resistive current represents a leading power factor rather than a lagging power factor as in the inductive circuit (Figure 8-8).

In any ac circuit containing resistance, capacitance, and inductance, a phasor (scaled vector) can be laid out on the x-axis (representing true power), capacitive (volt amperes) can be laid out on the upward y-axis (representing the capacitive reactive power), and the inductive volt-amperes can be laid out on the downward y-axis (representing inductive reactive power) as in Figure 8-9a. The difference between the two oppositely directed phasors represents the total circuit reactive power, and this resultant reactive power and the real power are added vectorially to determine the power triangle values. These values are True Power, Reactive Power, and Apparent Power (the latter is the hypotenuse of the power triangle). True Power is illustrated in Figure 8-9c.

Three-Phase Power

Single-phase ac power pulsates. The use of three-phase alternating current smooths the power by adding overlapping power waveforms. Instead of using a single coil (as in Figure 8-2), three-phase power is generated by using three coils spaced 120 degrees apart around the rotating magnet. Actually, the coils would be arranged in three sets of opposite pairs.

The three coils can be arranged in a "wye" connection, as shown in Figure 8-10a. The three sets of coils are labeled A, B, and C in Figure 8-10a. In a wye connection, the phase current (I_f) equals the line

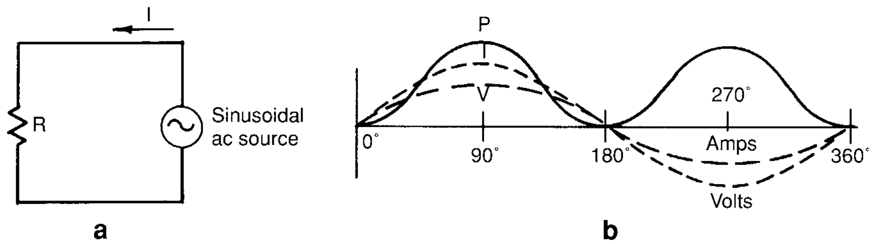


Figure 8-6. Basic resistance current. (a) The equivalent circuit; (b) waveform diagram; instantaneous values of V, I, and P are shown.

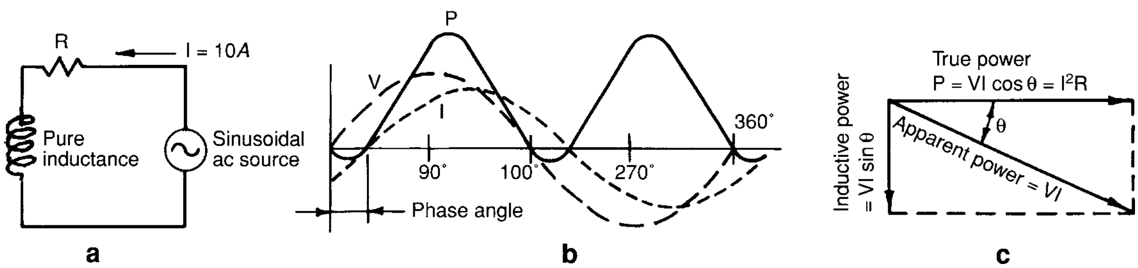


Figure 8-7. Basic inductance circuit. (a) The equivalent circuit resistance is in series with inductance; (b) waveform diagram. Instantaneous values of V, I, and P are shown; (c) phasor diagram.

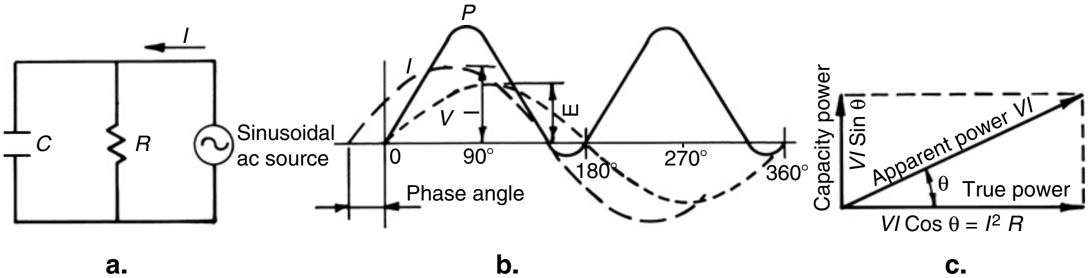


Figure 8-8. Basic capacitance circuit. (a) The equivalent circuit; (b) waveform diagram; instantaneous values of *V*, *I*, and *P* are shown; (c) phasor diagram.

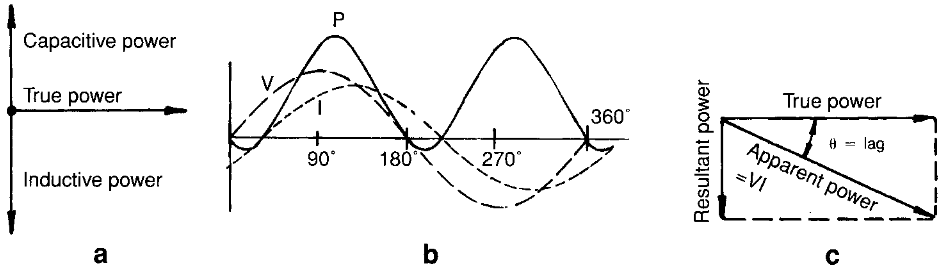


Figure 8-9. Power in circuits with combined inductance and capacitance. (a) Power vectors; (b) waveform diagrams; (c) phasor diagrams.

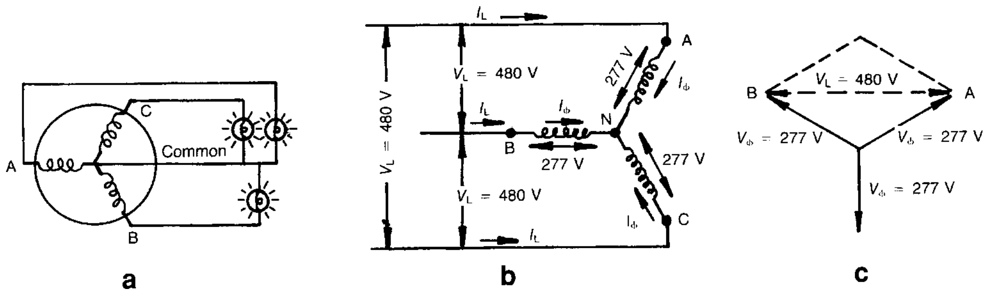


Figure 8-10. Example of generation of three-phase power. (a) The circuit; (b) voltage and current diagram for generator winding connected in wye; (c) phasor diagram.

current (I_L) as shown in Figure 8-10b, but the phase voltages (V_f) must be added vectorially, as in Figure 8-10c, to obtain the line-to-line voltage (V_L). These statements applied to Figure 8-10 can be expressed as

$$I_\phi = I_L \tag{8-4}$$

$$V_L = V_\phi\sqrt{3} = 277\sqrt{3} = 480\ V \tag{8-5}$$

The power, computed from Equation 8-3, is

$$P_{3\phi} = 3V_\phi I_\phi Pf = \sqrt{3}V_L I_L Pf \tag{8-6}$$

Wye connected windings can be three- or four-wire. No common wire is used in the three-wire connection.

Delta connected windings are usually three-wire, as illustrated in Figure 8-11. In delta connections, phase voltage equals line-to-line voltage, but line current must be obtained by the vectorial addition of the phase currents. These statements for the delta configuration can be expressed as

$$V_\phi = V_L \tag{8-7}$$

$$I_L = I_\phi\sqrt{3} \tag{8-8}$$

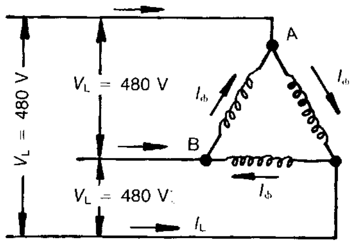


Figure 8-11. Voltage and current diagram for generator windings connected in delta.

As computed from Equation 8-3, power is

$$P_{3\phi} = 3V_\phi I_\phi Pf = \sqrt{3}V_L I_L Pf \qquad (8\text{-}6)$$

Thus, the power is the same whether the windings are wye or delta.

Electrical Measurements

Electrical measurements of interest to the pumping station operator include volts, amperes, watts, power factor, and frequency. Voltmeters and ammeters usually have phase selector switches to minimize the quantity of meters needed. Metering in larger pumping stations, particularly at the highest voltage level, tends to be more complex and comprehensive than in smaller pumping stations on lower voltage levels. Service technicians may need additional instruments to measure and record, for example, circuit and insulation resistance.

Electrical measurements primarily of interest to the electric utility include (kilo)watt-hours, watt-hour demand, and (kilo)var-hours.

Measurement errors may be classified into: (1) accidental or random errors, and (2) such systematic errors as

- Instrument errors
- Errors resulting from external conditions
- Errors caused by the observer.

Accidental errors are mistakes. Instrument errors are the result of shortcomings of the instruments used or of attempting to make measurements for which the chosen instrument is unsuitable. External conditions that produce errors can be caused by magnetic and electrostatic fields, ambient temperature and pressure, humidity, and other factors. Observational errors are primarily the result of subjective interpolation of analog displays, but, as digital instruments have virtually replaced analog instruments, such errors are now of minor interest. Common full-scale errors are ±3 to 5% for analog instruments, ±0.5% for digital instruments, and ±1 to 2% for kilowatt-hour instruments.

The most serious measurement errors are caused by using unsuitable instruments and by stray magnetic fields. An example of the use of an unsuitable instrument is measuring distorted wave forms with an instrument that is incapable of giving useful readings under such conditions, and such errors can easily reach 50%. The most common error caused by stray fields is using a poorly shielded instrument such as a clamp-on ammeter in the vicinity of either an electric motor or a magnetic starter; such errors are typically about 20%.

Permanent-magnet moving-coil (PMMC) instruments are the most common type of analog unit in general use. Shaft torque is generated when current through the coil produces magnetism that reacts with the permanent magnet. Movement ceases when shaft torque is balanced with torque from the return spring(s) or taut band, at which time the instrument may be read. PMMC instruments are dc milliammeters. Rectifiers and thermo-elements are provided for ac operation, series resistors are provided for voltmeters, and parallel resistors are provided for ammeters. Panel-mounted PMMC ac meters are usually used with current and/or potential instrument transformers. These meters read average values, and ac scales are calibrated in RMS values and are, therefore, accurate only for sinusoidal waveforms.

Other analog instruments often used include the moving-iron radial vane, ac-dc units that respond to average values and use series and parallel resistors for range extension, and the dynamometer units, usually watt-meters, ac-dc units that respond to RMS values and use series and parallel resistors for range extension.

Digital instruments are becoming very popular in panel-mounted versions (and in portable versions for maintenance) due to such features as ruggedness, accuracy, automatic range changing, polarity indication, and reasonable cost. Averaging units may be calibrated for RMS ac values and are accurate only for sinusoidal waveforms. RMS units may indicate total RMS values, or RMS values of selected harmonics. Peak capture and hold is a feature useful in measuring motor-starting currents. Circuitry details are beyond the scope of this book.

Panel-mounted instruments used in ac circuits in new pumping stations should be digital units that respond to true RMS values.

Resistance is not ordinarily measured directly. Instead, the instruments have an integral voltage source

(battery, hand-cranked generator, or other dc power supply), and current through the resistor is measured and displayed on the selected scale in terms of ohms.

The portable clamp-on ac ammeter has a movable half-jaw that enables the operator to slip it on over an insulated conductor for fast, easy, and safe current measurement. The jaws, when fully closed, form an iron core that encircles the conductor and intercepts the magnetic field generated by the current in the conductor. By transformer action, a voltage is induced in a secondary coil wound on the iron core. The voltage is measured and read in terms of amperes. Encircling more than one phase (or a phase and neutral) in the jaws cancels the magnetic field and causes a spurious reading or none at all. These instruments are usually multipurpose units that can also measure voltage and resistance.

Ac watt-hour meters are used to measure the flow or usage of electrical energy by means of the induction motor principle. Torque is produced on the aluminum rotor disk by means of magnetic fluxes from current and voltage coils. This torque, and thus disk rotational speed, is proportional to instantaneous power. The disk drives a mechanical or digital register that integrates disk speed with time, producing a watt-hour (or kilowatt-hour) reading. A three-phase meter uses two disks. Practical error limits are within about ± 1 to 2%.

The maximum amount of electrical energy used in a given period is measured with a demand meter, often integral with pumping station kilowatt-hour meters. The maximum demand for a short period (for example, 30 minutes) in any one year typically determines the power company's demand charge for the next year, because their distribution system must be able to provide the customer with that much energy, regardless of whether it is used at other times. Thus, for example, a storm water pumping station demand charge for an entire year may reflect the effect of one 3-day storm last year, even though the pumping station may be unused for 11 months.

Basic Electrical Calculations

The relation between resistance, volts, and current is given by Ohm's Law, Equation 8-1.

$$R = V/I \tag{8-1}$$

where R is measured in ohms (Ω), V is volts, and I is current in amperes (A).

The total resistance of several resistances in series is the sum of the individual resistances.

$$R_T = R_1 + R_2 + \ldots + R_N \tag{8-9}$$

The inverse of the total resistance of several resistances in parallel is the sum of the inverse of the individual resistances.

$$\frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \ldots. + \frac{1}{R_N} \tag{8-10}$$

where R_T is the total resistance and R_1, R_2, and so forth are individual resistances, all in ohms. Lest the reader think that all electrical calculations in the real world are as simple as presented above, one kind of calculation that an electrical engineer would make every day is to find the voltage drop through a simple electrical power feeder $E_D = E_S - E_L$ where the relation between E_S and E_L is given by Equation 8-11.

$$E_S = \left[(E_L \cos\theta + RI)^2 + (E_L \sin\theta + XI)^2\right]^{\frac{1}{2}} \tag{8-11}$$

The voltage drop through the power feeder is E_D, E_S is the voltage at the source, E_L is the voltage across the load, θ is the angle whose cosine is the load power factor, R is the resistance of the circuit in ohms, I is the line current in amperes, and X is the reactance of the circuit (usually inductive) in ohms. The equation is best solved by means of a computer program.

8-3. Power and Control System Elements

Solenoids

A solenoid, illustrated in Figure 8-12, is an electromagnet arranged to produce linear mechanical motion. An electromagnet is formed by a number of turns of wire around an iron core, solid for dc but laminated for ac to reduce eddy currents. An energized electromagnet draws an iron armature toward it by magnetic attraction. The armature may be connected by a suitable mechanical linkage to another object. The range of motion is small, perhaps 2 to 5 mm, except for long, hollow core types where the core is drawn in, perhaps 25 mm (1 in.) or more. A common application in pumping stations is the solenoid valve. Energizing the solenoid causes the valve to operate from open to closed or from closed to open position. Other motions are possible, depending on the design of the linkage mechanism. Solenoids are also used to operate braking systems. Some electric motors are required to stop immediately when the power is turned off. A solenoid-operated brake is

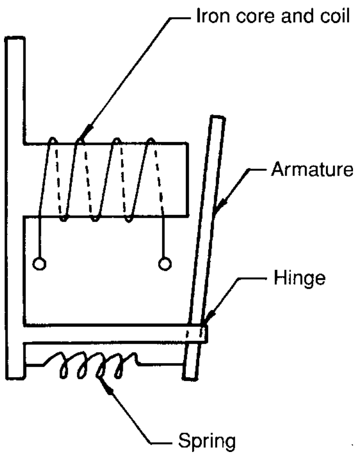


Figure 8-12. Basic solid-core electromagnet.

usually arranged to be held open by a solenoid that receives power when the motor is energized. Upon shutdown or loss of power, the brake is applied by the action of heavy-duty springs.

Solenoids are the basic operating elements of relays and contactors.

Relays

A relay is a solenoid-operated switch. A relay may have one or more switch contacts of various configurations, such as normally open (NO), or normally closed (NC), and various ratings, such as 10 A at 120 V ac. Operating coils may be ac or dc.

A popular relay type used in pumping stations is the industrial control relay. These relays are modular in construction, and switch contacts may be changed or added up to a limit. Other accessories such as timers and mechanical latches with unlatch solenoid coils may be added. NEMA has standardized contact ratings under the designation A600 (10 A continuous, 120 V, 60 A make, 6 A break). Other ratings are also given.

Another popular relay is the two-pole or three-pole double-throw plastic enclosed type. Many of these relays are fully interchangeable between manufacturers. Some are plug-in. Common contact ratings are 10 A or 15 A at 120 V or 240 V ac plus a small horsepower rating. General-purpose power relays are often used to control nonessential pumping station loads such as air conditioning and heating. These relays are available in single-pole or double-pole versions. Single-phase contact ratings are 30 A, 300 V, 1.5 kW (2 hp), not necessarily concurrently. Such relays have very limited current-breaking capability. Plastic-enclosed and general-purpose relays are often provided with equipment that is not in motor control centers, but they degrade the overall reliability of the pump and, therefore, should not be allowed. Only industrial control relays should be allowed. Thus, it is imperative that the proper relays, along with proper control wiring practices, be written into specifications for this equipment; that the shop drawings be checked (and most likely corrected!); and that a jobsite inspection be made to ensure that the specified relays were actually installed.

Timing Relays

Timing relays are delayed-action devices that are primarily relays. Output contact operation is delayed so that they open or close after a predetermined lapse of time after the relay is energized or de-energized.

There are several classifications, including: (1) ON Delay, where the relay contacts do not operate until after a timed period; and (2) OFF Delay, where the relay contacts operate instantaneously upon energization of the relay, but delay going back to their normal (shelf) position after de-energization of the relay.

Electromechanical or pneumatic relay timing is usually limited to a maximum period of about 1 minute, although they may be available to 60 minutes. Timing error is $\pm$ 10%, which is adequate for most pumping station uses. Long operating times can be obtained from solid-state relays and from motor-operated timers. Both of the latter types can provide very sophisticated timing sequences and a wide range of timing. Timing error may be as low as $\pm$ 0.1%.

Programmable Logic Controllers

The programmable logic controller (PLC) is a very useful form of a specialized digital computer that can provide a large number of timing arrangements and logical sequences to operate complex control functions. Isolated output contacts operate the various controlled devices. Because of their economical price and ease of programming, these units have replaced electromechanical relays on many pump sequence control applications. They are cost effective when compared with six to many hundreds of relays and timers. They should always be considered for pumping stations instead of large numbers of relays.

Transformers

Transformers operate on the principle of magnetic induction and are used to change voltage levels in ac

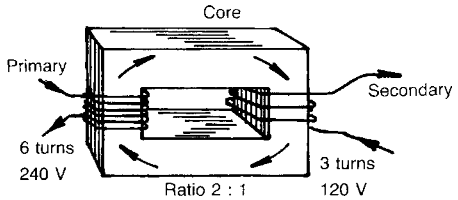


Figure 8-13. A single-phase transformer.

systems and to isolate (electrically) one circuit from another. Two coils, illustrated in Figure 8-13, are wound around the legs of an iron core that forms a closed magnetic circuit. As the current begins to rise in the primary winding it causes a proportional change in the magnetic flux in and surrounding the core. This changing magnetic field, cutting through the secondary coil, induces a voltage that is proportional to the input or primary voltage multiplied by the turns ratio of the two coils.

If a transformer input voltage is 240 V, the output voltage is 120 V for a turns ratio of 2:1. Except for minor losses manifested in heat, the total power input equals the total power output and, therefore, the current in the secondary equals twice the current in the primary.

Although most transformer applications involve the changing of voltage levels in power circuits, some are used solely for isolation and instrumentation purposes. Current transformers are designed for instrumenting power systems and usually have a single-turn primary, which may in reality be a piece of buswork, carrying very large currents. The secondary is wound to feed instruments (including meters) that are rated for 5 A full scale.

Another special transformer that finds some application in pumping stations is the autotransformer, a single-winding unit used primarily for producing small voltage change ratios. The single transformer winding has a tap (or taps) to increase or decrease the output voltage, but it does not isolate the output from the input. Autotransformers are commonly used for reducing the voltage applied to a motor during starting. After the motor has accelerated to a desired speed, the connections are changed with contactors, and either another step of the autotransformer is provided or the motor is connected directly across the line for the balance of the acceleration time. This reduced-voltage starting limits the current required from the utility during the starting of the motor.

Power transformers are available in various types, sizes, and construction. They can be classified per ANSI/IEEE Std 141-1986 as follows:

- Distribution and Power—According to rating in kVA. The distribution type covers the range of 3 to 500 kVA, and the power type of all ratings above 500 kVA.
- Insulation—Classification as liquid or dry types. Liquid is further classified as mineral oil, nonflammable, and low flammable. There are dry-cast coil, totally enclosed, nonventilated, and sealed gas-filled types.
- Substation or Unit Substation—Substation usually denotes a power transformer with direct cable or overhead line terminations. A unit substation is designed for integral connection to primary and secondary switchgear.
- Primary and Secondary—A primary substation has a secondary voltage rating of 1000 V or higher. A secondary substation has a secondary voltage rating of less than 1000 V.

Transformers may be located indoors or outdoors. Dry-type units are best used indoors, where the NEC imposes few special requirements. Liquid-filled units are usually located outdoors, subject mainly to prescribed clearances from building walls and roof overhangs. In very quiet locations hum may be a problem.

Liquid-filled units may be located indoors because of aesthetic, accessibility, security, or noise requirements, but the NEC imposes some rather expensive room (vault) construction requirements with regard to fire resistance, fire suppression, ventilation, and liquid containment in the event of transformer enclosure rupture. These requirements vary with liquid type (oil, nonflammable, etc.) and size in kVA.

Utility-owned transformers are nearly always outdoors. Refer to the utility for their requirements regarding pads or foundations, conduit stub-ups, and guard posts. Coordinate the transformer location and screening with utility and with the appropriate legal authorities.

When selecting transformers, the following data and ratings should be specified:

- Rating in kilovolt-amperes or megavolt-amperes
- Single-phase or three-phase
- Frequency
- Impedance
- Voltage ratings
- Voltage taps
- Winding connections: Single-phase, single-coil secondary or double-coil secondary for dual voltage connection. Three-phase delta or wye
- Temperature rise
- Basic impulse level (BIL)
- K-rating, if applicable. K-rated transformers are specially designed to resist the heating effects of

currents with harmonic frequencies (multiples of rated frequency). In pumping stations, adjustable-speed motor controllers are the major source of harmonic currents.

The desired construction details should include:

- Insulation medium—dry or liquid type
- Indoor or outdoor service
- Accessories—monitoring and safety devices, lightning arrestors
- Type and location of terminating facilities
- Sound-level limitations (if any)
- Manual or automatic load tap changing
- Grounding requirements
- Cooling or provisions for future cooling
- Energy conservation features.

In pumping stations of more than several kVA full-load rating, the transformers connected to the utility are ordinarily three-phase units. Some utilities serve small three-phase pumping stations at 240 V using three pole-mounted single-phase transformers. Two such transformers connected in open delta are sometimes used, but this practice should not be tolerated for fully loaded pumps, because the phase voltages are somewhat unbalanced and cause excessive phase currents and motor overheating. In larger installations, the transformer may be the property of the station owner and may be part of the switchgear lineup. Large transformer ownership can be negotiated with the utility.

Single-phase, dry-type transformers are commonly used to supply 120-V convenience receptacles, lighting, and small power loads, such as control and instrument and computer power. Three-phase transformers with a delta-wye winding connection, shown in Figure 8-10, are also widely used. The wye connection can supply both three-phase and single-phase loads simultaneously. The neutral is grounded and results in a uniform potential relative to ground in all phases.

For long-term variations in supply voltage or for growth in plant load, power transformers should have two 2.5% primary taps above normal and two 2.5% taps below normal. These primary winding taps are usually specified for de-energized reconnection.

The impedance rating of a transformer is usually given in percent of full load ratings, and it is ordinarily specified at 5 to 8%. This impedance limits the current that would flow in the primary and secondary systems of the transformer if a short circuit occurs in the secondary system or at the terminals of the transformer. Therefore, the impedance (in conjunction with the supply system's available short-circuit current) determines: (1) the required interrupting ratings of the load switching devices and fault current interrupting devices (fuses and circuit breakers), and (2) the bracing required in buswork. But the higher the impedance, the higher the voltage regulation, particularly during motor starting. Hence, carefully strike a balance between available short-circuit current and voltage regulation.

Many pole-mounted and pad-mounted transformers furnished by the utilities have low impedances, such as 3%, and thus they allow relatively higher fault current during short-circuit conditions than do higher-impedance units. Low-impedance transformers can result in more expensive switchgear, and thus the cheaper low-impedance transformer may be poor in overall economy. System voltage drop during motor starting is, however, less with low-impedance transformers.

Transformers are rated in kVA at a specific temperature determined by winding resistance measurements in accordance with NEMA standards. The transformer temperature rating is a function of the insulation system used. The effects of excessive temperature rise on insulation in transformers and electric motors are identical. See Section 13-9 and Table 13-2 for more detail. If proper allowance has been made in the specifications for a transformer, plant load increases may be accommodated by the addition of cooling fans and/or an oil circulation heat exchanger system to allow the transformer to withstand a larger load current safely.

Switches

The types of switches normally applied in power circuits include the following:

- Disconnecting switches
- Load interrupter (or load break) switches
- Safety switches
- Control switches.

A disconnecting switch is used for isolating a circuit or equipment from the source of power. It has no interrupting rating and is intended only for operation after the circuit has been de-energized by other means.

Load interrupter switches are intended for operating on circuits of 600 V or above. They are of air or fluid-immersed construction. The load break switch is usually manually operated and has a quick-make, quick-break operating mechanism that operates at a speed independent of the speed of operation of the handle. When combined with a properly rated fuse,

fast fault clearing as well as circuit isolation can be provided. This type of switch (in a metal-clad or metal-enclosed unit substation lineup) serves to disconnect and protect transformers and medium-voltage pump motors. It is sometimes provided as a roll-out unit for safety and ease of maintenance. (See ANSI/IEEE 141.)

Safety switches are commonly used for isolating and disconnecting circuits of 600 V and below. These switches are enclosed and may be fused or unfused. A safety switch designed specifically for motor circuits is rated in horsepower and must be able to interrupt the maximum current drawn by the motor. The maximum current is the stalled or locked rotor current, which is 600% or more of the rated full-load current of the motor. (See ANSI/IEEE 141.)

Control switches are constructed in many types, sizes, and ratings. They are usually rated for a few amperes and for a few volts to about 250 V in either ac or dc designs. They have large, plated contacts in most circuits, and for milliampere control circuits, the contacts are usually gold-plated.

Take precautions to ensure that control switches are operated within their voltage and current rating and that ac switches are not applied to dc circuits. The normal toggle switches for the control of lighting circuits are in a similar category and some are rated not only "ac only" but also for the type of lighting circuit.

Fuses

A fuse is an overcurrent protective device with a circuit-opening fusible link that is heated and melted by the passage of overcurrent through it (per ANSI/IEEE 100-1984). Fuses are calibrated for a specific melting time for a specific current that persists for a defined time. Fuse melting time (time delay) is approximately inversely proportional to the square of the current, but the exact nature of fuse operation must be determined by laboratory tests for each type and rating of fuse. The resulting time-current curves are plotted on graphs with special log-log scales and are used as fuse application guides.

Fuses are available in a wide range of voltage and current ratings and time characteristics. Fuses may be obtained with slow-blow characteristics for such applications as motor starting. These fuses must, without damage, pass up to 6 to 10 times the full-load motor current during the few seconds of acceleration time. Fuses are also available with fast-blow characteristics and with "current-limiting" characteristics. The current-limiting fuse is used in many applications where the downstream equipment is not fully rated for the short-circuit currents available at the supply point. Such systems rated less than 600 V ac can be interrupted in less than $\frac{1}{4}$ cycle and those above 600 V in less than $\frac{1}{2}$ cycle.

Fuses are low-cost, fast-acting, and fail-safe operating devices. They are relatively small, simple and easy to install, and permit a close match to motor thermal characteristics. However, fuses are subject to corrosion and eventually to failure due to repeated heating and cooling, and must be inspected at regular intervals. In a three-phase system, if a ground fault occurs in one phase of a motor and the fuse in that phase opens, the motor will continue to operate as a single-phase motor with currents in the surviving phases increasing as much as 73%. Overload relays will trip off a fully loaded motor, whereas a partially loaded motor may operate to burn-up because the overload relays would not sense overcurrent. Every overload relay-tripping event shortens the life of the winding insulation. In any event, a three-phase motor will not restart on single-phase current.

Circuit Breakers

A circuit breaker is a mechanical switching device capable of making, carrying indefinitely, and breaking current under normal circuit conditions, and also making, carrying for a specified time, and automatically operating to break current (trip) under circuit overcurrent and downstream short-circuit conditions. Most circuit breakers applied in pumping stations are enclosed in plastic cases, and are known as "molded-case circuit breakers." Also applied in large pumping stations are low-voltage power circuit breakers, both open and in plastic cases. Medium- and high-voltage circuit breakers, normally utility-owned, are not discussed herein.

The molded-case circuit breaker is enclosed in a ruggedly constructed molded plastic case. Smaller breakers are sealed. Larger breakers may be opened for maintenance. These breakers have external operating handles but are trip-free after automatic operation; that is, they cannot be reclosed into a fault. They may be obtained in single-pole, two-pole, and three-pole configurations. Handle ties are available for converting single-pole breakers into two- or three-pole configurations. But they are unreliable because if one-pole trips, the other(s) may not. Only multipole breakers with common internal trip bars should be used in pumping stations to avoid single-phasing three-phase motors and to ensure complete circuit de-energization for safety. Molded-case circuit

breaker operating mechanisms wear if operated too often. They should not be used in lieu of motor contactors. Refer to ANSI/IEEE 141-1993.

Molded-case circuit breakers are available in several types. The thermal-magnetic type is the one most commonly used. This type has two automatic trip mechanisms, both of which operate to release the trip bar. The thermal or bimetal trip element operates in response to overload currents in an inverse time manner: the higher the current, the shorter is the response time. This element is designed to follow the heating characteristics of the conductor, and it allows for temporary overloads, such as for motor starting. Refer to Figure 8-14a. The magnetic trip element operates instantaneously (within a fraction of a cycle) upon occurrence of high-current short-circuits, and the trip setting is usually 8 to 10 times the breaker-rated current. Many breakers have a magnetic trip adjustment of 5 to 10 times the breaker rated current, and they should be adjusted to be as low as practicable for maximum short-circuit protection (see Figure 8-14b). Some circuit breakers have only magnetic trips. In pumping stations, most are used in combination motor starters where the overload function is provided in the overload relay. These circuit breakers are often known as motor circuit protectors (MCPs). MCPs have adjustable trips that should be set just above the motor inrush current. Refer to manufacturer's literature for available sizes and adjustment ranges.

Fused circuit breakers are hybrid circuit breakers with special fuses that open on very high fault currents that would destroy the breaker. Fuse operation releases the trip bar and opens the breaker to prevent single-phasing.

Molded-case switches are circuit breakers without trip elements and are used as manual disconnects. Some have nonadjustable magnetic trip units for self-protection. Molded-case switches must be applied with caution because fault currents near their interrupting rating may force the contacts to part. Contacts in all circuit breakers are designed so that high currents aid in forcing the contacts apart. Therefore, they may have a withstand rating below their interrupting rating.

Electronic trip units are now available for molded-case circuit breakers. These are considerably more expensive than their thermal-magnetic counterparts, but they are available with a full range of features that increase pumping station reliability, because they can be adjusted to coordinate with other breakers in the circuit. Hence, a serious short circuit in a branch

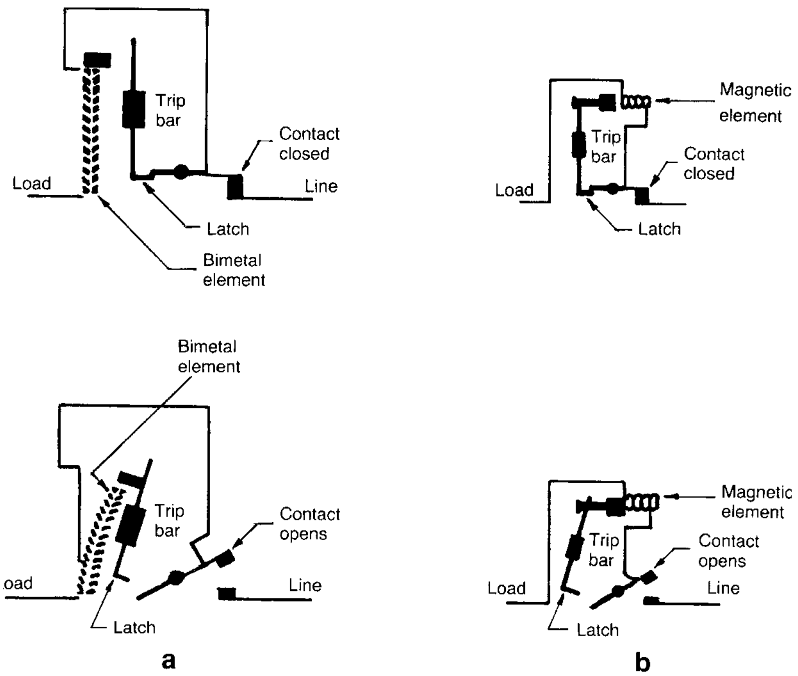


Figure 8-14. Operation of circuit breakers. (a) Thermal action; (b) magnetic action. For both types, the condition of the breaker before being tripped is at the top.

circuit would cause only the branch circuit breaker to trip, thereby sparing the main circuit breaker and permitting the pumping station to continue in operation. Main and feeder thermal-magnetic circuit breakers do not coordinate with each other or with branch circuit breakers.

Molded-case circuit breakers have several ratings. Among them are:

- Rated current: most breakers carry 80% of rated current continuously (at least 8 hours). Refer to manufacturer's literature, and do not specify a breaker rated below 15 A. Use fuses.
- Voltage: 240 or 480: 480-V units may be applied on 240-V systems. Single-pole breakers are rated 120 or 277 V.
- Interrupting rating (RMS Symmetrical): a circuit breaker must be able to interrupt the maximum fault current available to it. Operators should always open and inspect a circuit breaker after such an operation, because it may need replacement.

Standard thermal-magnetic circuit breaker ratings are given in Table 8-1.

Because the utility may increase the available fault current without notice, designers should use discretion in circuit breaker selection. Do not apply those subject to utility fault current near their interrupting rating, and allow for motor contribution (four times full-load current for common induction motors).

Most smaller molded-case circuit breakers are designed for use in circuit breaker panelboards. A circuit breaker panelboard is a factory assembly of circuit-breaker mounting arrangements and connecting buses, and may contain up to 42 single-pole circuit breakers. A main circuit breaker may be included. The enclosure is of sheet metal for safety, and may contain a door (sometimes lockable) over the circuit-breaker operating handles. Circuit breakers may be either plugged into or bolted to the buses. Bolting is recommended for reliability, but bolting is somewhat more expensive.

Circuit breakers may be provided with a number of accessories, including lock-ons, padlock hasps, and alarm contacts. Refer to the manufacturer's literature. Ground fault relaying is available for use with main circuit breakers and with electronic trip units.

Low-voltage power circuit breakers are used in switchboards as the main and tie circuit breakers. They may have trip ratings to 4000 A or higher. Withstand and interrupting ratings are quite high, and they may be obtained with back-up fuses for use up to 200,000 A. Modern trip units are solid state with a full range of adjustments, and ground fault relaying is available. Breaker contacts are opened and closed by a stored energy system (springs) to provide quick-make, quick-break operation. Springs are compressed by electric motor devices. In the event of motor failure, springs may be compressed manually. Remote electrical opening and closing control is possible, such as for switching to alternate electrical sources or standby generators and tie-breaker operation. These breakers are easily inspected, maintained, and serviced.

Table 8-1 Standard Thermal-Magnetic Circuit Breaker Ratings

Continuous current rating	Interrupting rating	Continuous current rating	Interrupting rating
120/240 V		277/480 V	
15–100 A	10,000 A	15–100 A	14,000 A
15–150 A	22,000 A	15–125 A	18,000 A
100–225 A	42,000 A	15–100 A	25,000 A
15–30 A	65,000 A	70–250 A	25,000 A
		125–400 A	30,000 A
		300–1000 A	30,000 A
		15–125 A	35,000 A
		70–250 A	35,000 A
		125–400 A	35,000 A
		15–125 A	65,000 A
		100–250 A	65,000 A
		300–2000 A	65,000 A
		600–1200 A	100,000 A
		600–2500 A	100,000 A

Motor Branch Circuits and Controllers

The requirements for each component of a motor branch circuit are delineated in Article 430 of the NEC. A single-line diagram of a typical motor branch circuit is illustrated in Figure 8-15. The feeder or service disconnect and overcurrent protective device(s) and the feeder or service conductors are not part of the motor branch circuit, but their size calculations are influenced by the rules in Article 430.

The branch circuit disconnect and branch circuit overcurrent protection consist of either a circuit breaker or a fused switch. It must be convenient for a worker to padlock a switch in open position for safe motor or pump maintenance. Circuit breakers in motor control centers are often adjustable magnetic-only devices, often called "motor circuit protectors." They must be adjusted high enough to not trip on starting but to trip on a short circuit. Fuses are sometimes dual-element types selected for both branch circuit and motor overload protection. Fuses are low-cost, easily replaced devices that are fail-safe and easily coordinated with other protective devices. But they can allow single-phasing of the motor. Some fused switches are arranged to open the switch automatically if a fuse opens. Fused switches must be horsepower-rated below 200 hp, and equivalent measures must be taken above 200 hp. Large, important motors, particularly in wastewater lift stations, justify the cost of a single-phase protective relay or of a ground fault protective relay. Single-phase monitors and protection devices cost about $700 per motor at 1997 prices, regardless of motor size. Installation would increase the cost by perhaps 50%. Where power is reliable, the frequency of a single-phasing event might be less than once per 20 years, whereas it might be once per decade in rural communities or where lightning is prevalent. Designers should assess the likelihood of burned-out motors and the loss of pumping against the cost of protection to balance risk with cost.

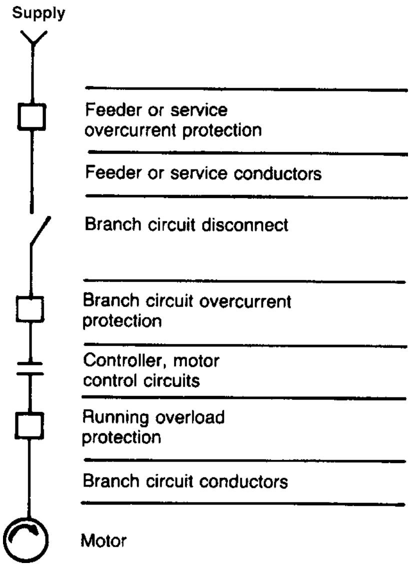


Figure 8-15. A typical motor branch circuit diagram per NEC.

Motor circuit conductors must be selected to carry at least 125% of motor-rated full-load current.

Controllers for AC Squirrel-Cage Motors

The controller must be capable of starting and stopping the motor and of interrupting the locked rotor current. For portable motors $\frac{1}{3}$ hp and smaller, a cord and plug assembly may serve as the controller, For stationary motors $\frac{1}{8}$ hp and smaller that are normally left running and cannot be damaged if stalled or jammed, the branch circuit protective device may serve as the controller. For attended motors, manually operated motor starters and some light switches of suitable ratings are available in single- and three-pole models, For unattended or automatically controlled motors, motor starters using magnetically operated contactors must be installed.

A contactor is a heavy-duty, solenoid-operated switch used primarily as a controller for motor on-off control, but also for heaters and lights. When combined with an overload relay, the assembly is known as a "motor starter." A motor starter with a circuit breaker or fused switch is known as a "combination motor starter."

A motor contactor must be able to interrupt locked-rotor current. Locked-rotor current is drawn during the starting period or indefinitely should the pump or other driven device be jammed or stalled. Locked-rotor current is roughly six times full-load current or more for the high-efficiency line of motors. Contactors are not required to interrupt fault currents. The fuses or circuit breaker perform that function. The locked-rotor current rating determines the horsepower rating of the contactor.

Running overload protection is intended to protect motors, motor controllers, and motor branch circuit conductors from overheating due to motor mechanical overloads and from failure to start. It does not protect against short circuits and ground faults.

Most of the common single-phase motors larger than $\frac{1}{15}$ kW ($\frac{1}{20}$ hp) have an integral thermal switch

(in series with the motor conductors) that will shut off the motor if it becomes overheated. Most three-phase motors are protected by thermal or solid-state overload relays (one in each phase) that are part of the motor starter. The following description applies to relays on Size 5 motor starters and smaller.

Thermal overload relays of the melting alloy and bimetal types have been available for many years. Connected in series with motor conductors and on the load side of the contactor, they carry motor current and are thereby heated. Should excessive current flow for a length of time due to mechanical overload or to a locked-rotor condition, the relay (connected so as to de-energize the contactor coil when hot) is gradually heated. As the motor does not heat instantaneously, the relays are designed to have a time lag matching that of the motor, and this time lag allows the motor to start. Relays are available in quick trip (Class 10), standard trip (Class 20), and slow trip (Class 30) versions, and can be set for manual or automatic reset. Automatic reset should be selected for unattended pumping stations. Because ambient temperatures vary and significantly affect motor heating, noncompensated relays are provided as standard. Ambient-compensated relays are available for motors in a constant ambient condition, such as in a submerged application. Relays have heaters selected to match the motor and other conditions, and may be adjusted within limits as necessary for a closer match to motor-heating characteristics.

Solid-state overload relays are now available as direct replacements for the melting alloy type. They are somewhat more expensive than thermal relays. These solid-state relays have a large adjustment range and so do not need heaters. They may have other features, such as phase loss protection, phase unbalance protection, and electrical remote reset.

A squirrel-cage induction motor, when started at full voltage, draws high starting current inrush and produces high starting torque. A typical NEMA design B motor has an inrush current of 600% of motor full-load amperes and the starting torque is approximately 150% of full-load torque at full voltage. High current inrush and starting torque can cause problems in the electrical and mechanical systems. When a motor is started at reduced voltage, the current at the motor terminals is reduced in direct proportion to the voltage reduction while the torque is reduced by the square of the voltage reduction. If the same NEMA B motor is started at 70% of line voltage, the starting current would be 70% of the full voltage value (that is, $0.70 \times 600\% = 420\%$ FLA) and the torque would be $(0.70)^2$ or 49% of the normal starting torque (that is, $0.49 \times 150\% = 74\%$ Full Load Torque). Reduced voltage starting provides an effective means of reducing both inrush current and starting torque. Reduced voltage starting is used mainly to satisfy the electric utility by controlling inrush current and limiting it to values that can be safely handled without excessive voltage dips and the accompanying light flicker. Other benefits of reduced voltage starting are smooth acceleration up to full speed (thereby reducing hydraulic pumping surges) and a probable increase in the permissible number of starts per hour of the motors.

Motor starters are available as either full- or reduced-voltage or reduced-inrush types. Full-voltage, or "across-the-line" starters, as the name implies, have the supply voltage applied directly across the motor windings. Reduced-voltage starters apply a reduced voltage at starting and gradually, either in steps or a smooth ramp, increases voltage to bring the motor up to full speed. Reduced-inrush types, also considered reduced-voltage types, apply full voltage to the motor windings but due to winding configurations within the motor apply only a percentage of the terminal voltage across the winding.

Across-the-Line Starters

The direct across-the-line (DOL or full-voltage) starter is the most commonly used starter in pumping stations because of its simplicity, ease of maintenance, and relatively low cost. It is nearly always chosen for motors of 55 kW (75 hp) and smaller. Motor starters are available in the following styles:

- Full Voltage Non-Reversing (FVNR)
- Full Voltage Reversing (FVR)

Most of the above styles of motor starters are available in low-voltage starters (below 600 V) and in medium-voltage starters (2300 to 13,200 V). The ratings of motor starters are given in Table 8-2.

No motor starter smaller than NEMA Size 1 should be specified or permitted in pumping stations, and some designers say Size 2. The cost difference is not great, but the larger starters are considerably more robust and ensure greater reliability and longer service life.

Use caution in specifying motor starters. Thorough knowledge of the motor operating conditions and other constraints is required for selecting starter equipment for each application. Prices vary widely, so designers must not specify an expensive soft starter when a simple across-the-line unit would be adequate, nor should an overly expensive starter be specified

Table 8-2 Motor Starter Ratings

NEMA size	System voltage	Maximum hp Single phase	Maximum hp Three phase	NEMA size	System voltage	Maximum hp Three phase
00	120	½		3	208	25
	208		1 ½		240	30
	240	1	1 ½		480	50
	480		2			
0	120	1		4	208	40
	208		3		240	50
	240	2	3		480	100
	480		5			
1	120	2		5	208	75
	208		7 ½		240	100
	240	3	7 ½		480	200
	480		10			
2	120	3		6	208	150
	208		10		240	200
	240	7 ½	15		480	400
	480		25			
				7	240	300
					480	600

because of a lack of design information. The decision should be made after sufficient information has been developed relating to the load conditions, accelerating time of the drive, and accelerating time allowed by the serving utility. Also, in pumping applications, avoid simple on-off controls for pumps when the pumping cycle time is likely to be very short (more than 3 starts/min). Otherwise, derate motor starters by using their jogging horsepower rating (see manufacturer's literature). Starting too frequently causes motors and motor starters to overheat due to the high inrush current. There have been situations where too-frequent starts have caused early failure of a motor. Failure is even more likely to occur if the motor is undersized for the peak load during the short-duration pumping periods, because both the load and the starting heating effects are additive.

Reduced-Voltage Starters

Use of a reduced-voltage starter should be considered: (1) if the power system cannot provide the high starting currents typical of larger motors; (2) if the power company limits starting current or kVA (limit for permitted across-the-line-starts is usually around 75 hp); (3) if a standby generator is used; (4) if the feeder and/or branch circuit is longer than 180 m (600 ft); (5) if the voltage drop due to starting current would be 20% or more (actually, the motor would start and run at a 30% drop, but the contactors might drop out); or (6) number of starts-per-hour needs to be increased.

Various methods of reduced-voltage starting have been developed. The common reduced-voltage starter types are shown in Table 8-3 together with the expected motor voltage, line current, and the output torque of the motor.

Table 8-3. Comparison of Starting Methods

Starting method	% of full line voltage: Voltage at motor	Line current	Motor output torque
Full voltage	100	100	100
Autotransformer			
80% tap	80	64	64
65% tap	65	42	42
50% tap	50	25	25
Primary resistor (typical size)	80	80	64
Part winding	100	70	50
Wye (start) – Delta (run)	100	33	33
Solid-state	0–100	0–100	0–100

Autotransformer Type

The autotransformer starter is simply a transformer configured with contactors to allow a stepped acceleration to full speed. It is accomplished by "tapping" the transformer at 50, 65, or 80% of full voltage. One of these taps is the first step of voltage applied to the motor and is subsequently followed by a second step and finally to full voltage.

Advantages:

1. High torque
2. Low starting line current

Disadvantages:

1. Limited adjustability to load conditions
2. Mechanical shock to system between steps
3. Large size; takes up control room space
4. High contactor maintenance
5. High purchase cost
6. Unable to compensate easily for input voltage variations
7. Uncontrolled deceleration
8. Poor motor protection with the use of bimetallic overload with 20% loss of accuracy

Primary Resistor Type

This type of starting is in very limited use now, but is mentioned as some may still exist in old pumping facilities. Primary resistor starters consist of a line contactor, running contactor, timing relay, overload relay, and a resistor or reactor that acts as the current reducing device. These starters were generally used where the starting current was limited to 50% of DOL current. That means that the starting torque was reduced to approximately 25% of the DOL torque. With the primary resistor starter, the torque increased with speed until, at approximately 90% speed, full-load torque was reached. With the primary reactor starter, due to the more favorable relationship between the voltage drop across the reactor and the motor, the torque reaches a peak of approximately 150% of full-load torque at 90% speed.

Advantages:

1. Multiple acceleration points
2. High starts-per-hour
3. High power factor

Disadvantages:

1. Obsolete.
2. Inefficient

Wye-Delta Type

Wye-delta starters can only be used on wye-delta motors that have six leads that allow for motor windings to be connected in either a wye or delta configuration. During start-up, the windings are connected in the wye, resulting in 58% of line voltage applied across two windings. This reduces both inrush and starting torque to 33% of the delta connected values. After a set time delay, the motor leads are switched to the delta connection.

The wye-delta starter is available in both open and closed transition configurations. Closed transition starters are supplied with an additional contactor and resistor bank used to keep the motor windings energized for a few cycles until the transition from wye to delta is complete.

Advantages:

1. High torque and efficiency
2. Low starting current

Disadvantages:

1. Specific motor required
2. Long acceleration starting times
3. Low starting torque

Part Winding Type

Part winding starters can only be used with part winding motors. During a part winding start, only one winding is energized, which reduces the inrush current to 60–70% (depending on the motor design) and starting torque to 50% of normal starting values with both windings energized. Most (but not all) dual voltage 230/460-V motors are suitable for part winding starts at 230 V.

Advantages:

1. More compact than other electromechanical starters
2. Relatively inexpensive

Disadvantages:

1. Specific motor required
2. Short starting times
3. Low starting torque
4. Non-adjustable

Solid-State Type

In recent years, solid-state "soft start" reduced-voltage starters (SSRVSs) have gained favor in the pumping industry by combining a standard three-lead motor

with a solid-state starter that utilizes semiconductor technology to reduce voltage and hold starting current below a preset maximum value. Such starters often offer a number of programmable options as to how much voltage is applied to the motor during starting.

The SSRV starter uses high-speed switching devices called silicon controlled rectifiers (SCRs) to switch on for only a portion of each half of the sine-wave line power. By doing so, the voltage getting to the motor is reduced proportionately by the amount of time the switch is delayed. SSRV starters cause the start voltage applied to the motor to change with time irrespective of the motor and load conditions, and the motor eventually reaches full speed.

To start a load, the motor must develop sufficient torque to overcome breakaway torque and then develop enough torque over the entire speed range to exceed the work (including losses) torque of the load, plus provide surplus torque to accelerate it to full speed. The flexibility of the SSRV starter is one of its major advantages. The "switch on" or start voltage setting, which is usually adjustable between 0 and 80% of motor FLI, is set just high enough so that there is no time delay before the motor starts turning. The "ramp up" time is set so that the current ramps up to maximum to coincide with the rise in the load torque demand as the speed increases (usually around 10 to 15 s for a centrifugal pump load). The current limit should be set just high enough to provide continuous acceleration to motor full speed (usually around 250–300% FLI for a centrifugal pump load). If it is set too low, the torque is insufficient to accelerate the motor to full speed.

SSRV starters have current passing through them all the time that the starter is running, thereby generating heat and harmonics. A full-speed bypass contactor should be included as an integral part of the SSRV starter. When the top of the acceleration ramp is reached, the bypass contactor is activated and the starter electronics are switched out of the circuit.

Advantages:

1. Flexible programming
2. High number of starts per hour
3. Compact size
4. Low heat loss
5. Low starting current
6. Stepless acceleration

Disadvantages:

1. More expensive than most electrometrical starters (but getting less expensive)
2. Sensitive to electrical power disturbances, line transients, and surges

Motor Control Circuits and Devices

The simplest, most commonly used motor control circuit utilizes momentary start-stop pushbuttons and is illustrated in Figure 8-16. This control circuit is sometimes called "three-wire" control because three control conductors are run from the motor starter to the pushbutton station. Momentarily depressing the start button energizes the contactor M coil, which is sealed in through the normally open M contact (which closes when the contactor main con-

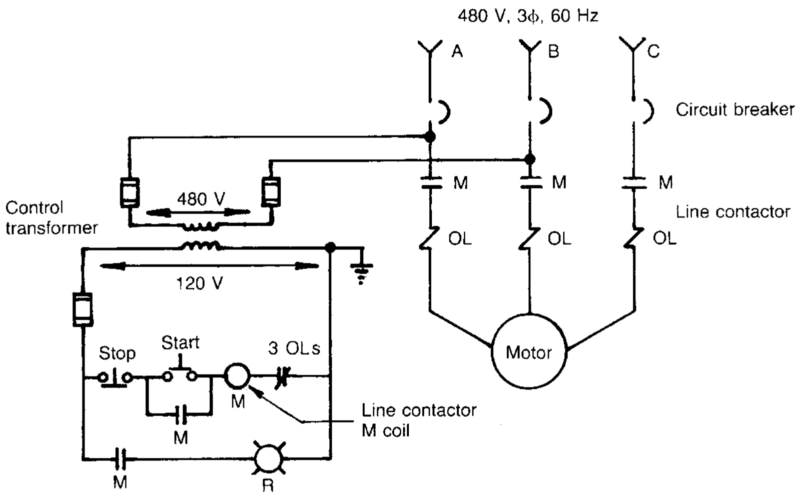


Figure 8-16. Pushbutton three-wire motor control circuit.

tacts close). The start button may then be released without dropping out the contactor, and thus the motor is energized through the closed main contacts of the starter. Depressing the stop button or tripping an overload relay de-energizes the M coil, opens the line contactor, and thus disconnects the motor from the line.

A similar control circuit using a maintained (a contact that remains closed after actuation), two-position (On–Off) or a three-position (On–Off–Auto) selector control switch can also be utilized for motor starting and running control. See Figure 8-17. The selector switch control circuit allows the motor to operate in either the manual or automatic mode. The manual mode consists of the On and Off positions. In the Auto mode (with the selector in the Auto position), the motor is started *or* stopped by the closing or opening of, for example, the wet well level control switch located remotely from the motor controller.

Control circuits for pumping applications may be designed to meet other requirements, such as:

- Pump sequencing, which reduces the frequency of motor starts, exercises rotating equipment and alternates the pumps to start on each successive operation. In this mode of operation, pumps tend to wear out at about the same time, which may or may not be acceptable to the owner.
- A time delay, which is used in a control circuit to delay the restarting of the pump after the restoration of power following power failure.
- A permissive condition that must be satisfied before control action can become effective, such as delay in shutting down an operating pumping unit until the replacement unit has reached a specific speed range and is thus capable of taking over the pumping duty.
- Electrical interlocks, which are usually a series of contacts on relays or other control devices used to enable or prevent the operation of equipment in a prearranged sequence.

Motor Control Centers

A motor control center (MCC) is a unitized assembly of grouped motor controllers, interconnecting buses, wiring spaces, and other electrical controls and equipment associated with providing power and control for several motors. MCCs provide incoming line facilities, main circuit breaker space (if needed), horizontal bus, and a vertical bus in each section. MCCs may be obtained with plug-in cubicles for NEMA Size 1 through Size 4 motor controllers. Several classes of control wiring arrangements are identified by NEMA standards. Layouts of typical MCCs are shown in Figures 8-18 and 8-19. The MCC structure must have an enclosure and a seismic rating compatible with the location in which it is installed. The built-up motor control panel shown in Figure 8-19c is a low-cost alternative to an MCC, but only if labor costs are not considered.

It is important that the MCC (bus bracing, combination starters, and individual circuit breakers) be rated for the short-circuit duty predicted by the fault calculations. If a main circuit breaker is to be provided in the unit, then its setting should be coordinated with the supply overcurrent equipment

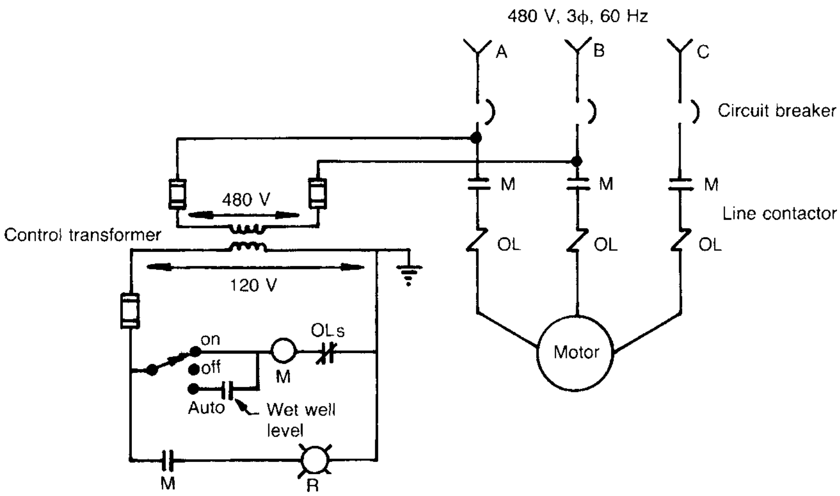


Figure 8-17. Three-position selector switch motor control circuit.

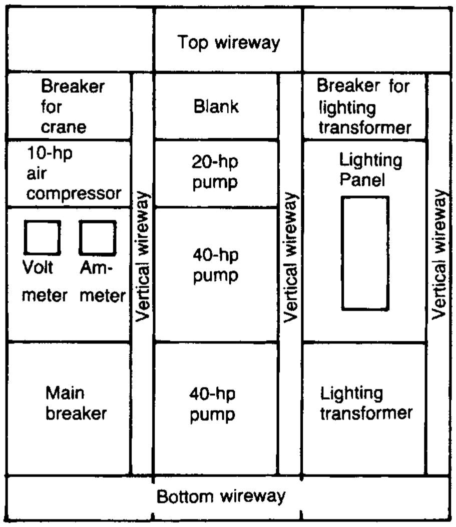


Figure 8-18. Elevation view of a typical low-voltage motor control center. Cubicles are labeled with name of load. Permanent name plates are recommended.

as well as with the branch circuit breakers connected to the buswork. In the coordination with the branch circuit breakers feeding a number of motors, the breaker feeding the smallest motor has the highest interrupting duty, so be certain that protection for the smallest motor has adequate interrupting capacity. The reason for this high interrupting duty on the smallest circuit is that during the initial period of the short circuit, all other running motors contribute current to the fault, thus adding to the short-circuit current from the main power source. Large motors contribute larger short-circuit currents than do small motors.

NEMA Class II, Type B wiring is recommended in the MCC. In this wiring method, factory power and control wiring is terminated on terminal blocks in or adjacent to the cubicle. The terminal blocks form the interface between factory and field wiring. The factory furnishes complete layout, wiring, and control diagrams for each unit as well as for the entire assembly. All terminals are labeled and each wire is numbered and/or color coded. The manufacturer's diagrams should be required to include the entire motor branch circuit and controls.

It is advisable to require: (1) the circuit breaker (circuit disconnect) handle of each unit to be lockable in the open position and (2) provisions for at least two padlocks to be used for locking out the circuit. The door must be interlocked so that the circuit breaker is in the Off position before the door of the unit can be opened.

There must be enough space in each cubicle so that no device is mounted behind another device. All control equipment must be accessible for inspection when the front door of the cubicle is open. Empty spaces or spaces wired for "future" starters must have bus openings completely covered so that when the door is open, no live buses are exposed.

Motor control centers should be installed in easily accessible locations with adequate spacing from other equipment, in accordance with NEC and local and state codes. They should be in a clean, dry location to avoid the need for specifying the more expensive gasketed enclosures. The space should be well-lighted and ventilated (see Chapters 9 and 23).

Where control power transformers with fused primary and/or secondary are provided, it is advisable to install a blown-fuse indicator in the door of each unit for monitoring the control circuit continuity. It may also be desirable in some installations to include door-mounted pushbutton controls for the motor and indicating lamps to show: (1) power available, (2) motor running, (3) automatic or remote control indicators, and (4) running-time indicator. In large stations where computer control and monitoring is used, the running-time records may be more economically kept in the computer files.

Sometimes the plant control panel is installed in the motor control center when there is no other suitable location.

Insulated Cables and Conductors

Cables and conductors are not the same thing. A cable contains two or more conductors.

Cables

Insulated conductors used in electric circuits may be grouped together and surrounded by a jacket to form an insulated cable. Cable jackets must be suitable for the location and application, because they may be exposed to moisture, dirt, etc., without the protection of conduit.

Conductors and Terminations

Most conductors are of copper or aluminum. Both metals are considered to be excellent conductors of electricity. The conductivity of aluminum is only 61% of that of copper, with the result that aluminum conductors are approximately one size larger than the equivalent copper conductor. The specific gravity

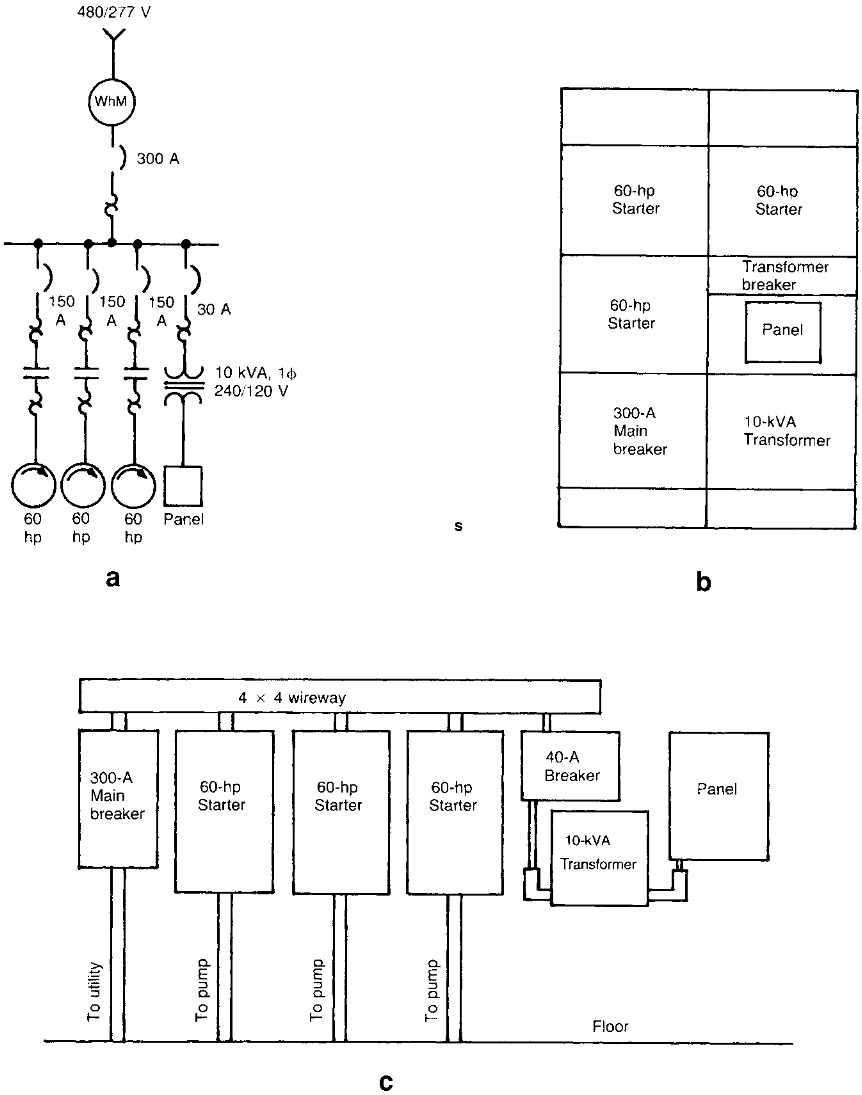


Figure 8-19. Examples of different types of motor control centers. (a) Single-line diagram; (b) motor control center; (c) built-up motor control panel.

of aluminum is 30% of that of copper, so aluminum conductors are lighter and easier to handle.

Conductors may be solid or stranded. The NEC requires that conductors No. 8 AWG and larger be stranded. Smaller conductors may be either solid or stranded. A number of types of stranding are available, and the selection depends upon the application. Conductors No. 2 AWG and smaller are usually 7-strand rope-laid (center strand surrounded by 6 outer strands). In larger sizes, 12 strands surround the basic 7 to form a 19-strand conductor.

Oxide forms on the surface of both copper and aluminum, slowly on copper and rapidly on aluminum. Copper oxide is a relatively good conductor, but aluminum oxide has high resistance and must be removed in the termination process to prevent termination overheating and failure.

Conductor termination workmanship must always be of the highest quality to ensure a reliable, trouble-free electric system. But aluminum terminations are much more sensitive to workmanship than are copper ones. The aluminum must not be nicked when the insulation is cut off. The aluminum oxide must be removed by wire brushing, and the aluminum must be immediately coated with a conductive anti-oxidant compound. The connection

must be torqued to manufacturer's specifications, retightened the next day, and retightened annually, because aluminum has a 31% greater thermal coefficient of expansion, which, together with its tendency to creep, loosens connections (see ANSI/AIEE 141). The only reliable alternative to torquing bolted terminals is the use of compression lugs, and these are recommended for high-quality work for both copper and aluminum conductors in Size No. 2 AWG and larger. Terminals for use with aluminum conductors must be so labeled. *Because of the termination problems, it is industry consensus that only copper conductors should ever be specified and used in pumping stations.*

Buses are electric conductors in the form of bars of various shapes and sizes. Buses are used in motor control centers, switchgear, and elsewhere for high amperage capacity. They may be insulated for safety. Aluminum buses are often tin-plated for corrosion resistance. Bolted connections are made with belleville washers, properly torqued.

Insulation

Most insulating material is composed of organic compounds. Two exceptions are magnesium oxide, used for mineral-insulated cables, and mica, used in some high-voltage cable construction. Insulation and cable jackets for pumping station applications should have physical and electrical characteristics such as:

- Resistance to moisture
- Resistance to heat
- Resistance to ozone
- High dielectric characteristics
- Resistance to abrasion
- Hardness at utilization temperature.

Examples of low-voltage (600 V) conductor insulation are given in Table 8-4.

Medium-voltage (5 to 15 kV) cables may be constructed with semiconducting tapes on the inside and outside of the insulation with a conducting shield between the outer semiconducting tape and the jacket. Medium-voltage cable insulation materials commonly used are:

- Natural rubber
- Butyl rubber
- Cross-linked polyethylene (XLPE)
- Ethylene-propylene (EPR).

Shielded cables should always be used. The use of shielding in medium-voltage cables lowers the electrical stress on the cable insulation by providing a grounded surface equidistant radially in all directions from the conductor. When laid on a metallic surface, an unshielded cable allows a distorted field pattern within the cable structure that causes a high voltage gradient in a portion of the insulation. It shortens the life of the cable.

Current Rating

The ampacity (current rating) depends upon the conductor material, size, type of insulation, and the ambient conditions of its installation. Refer to NEC Tables 310-16 through 310-19 and 310-69 through 310-84 and the notes pertaining to them.

Raceways and Wireways

The NEC requires that all electrical conductors be enclosed and protected by some form of raceway. Jacketed cables, where permitted, are exempt. The most common forms of raceways in pumping stations are (1) conduit, and (2) wireways.

Galvanized rigid steel is the most commonly used material for conduit systems. Grades lighter than

Table 8-4 Insulation Types for Low-Voltage Electrical Conductors

Type	Insulating material	Maximum operating temperature, °C[a]	Application
RHH	Heat-resistant rubber	90	Dry and damp
RHW	Moisture- and heat-resistant rubber	75	Dry and wet
THW	Moisture- and heat-resistant thermoplastic	75	Dry and wet
THHN	Heat-resistant thermoplastic	90	Dry and damp
THWN	Moisture- and heat-resistant thermoplastic, polyamid nylon jacketed	75	Dry and wet
XHHW	Moisture- and heat-resistant cross-linked polymer	75	Dry and wet
		90	Dry and damp

[a]Can be operated at these temperatures only if the circuit is tested and rated by UL at these temperatures.

Schedule 40 steel do not generally exhibit the degree of corrosion resistance required for long life in the damp surroundings that are common in pumping stations. Threaded connections should always be used. The specifications should require full hot-dip galvanizing treatment after fabrication and threading. Field-cut threads should be painted with a zinc-bearing material.

On small projects, if high-grade hot-dip galvanized conduit is not available, consider Schedule 40 or 80 rigid PVC conduit in wet places and where concrete encasement is required. If the pumping station is part of a large project wherein office and shop space is allocated, intermediate-grade galvanized conduit may be specified for these areas. On all projects where PVC conduit is run through or embedded in concrete, special precautions must be taken to avoid breakage during the placement of the concrete.

Wireways are square or rectangular metal channels or troughs with removable covers designed for routing and connecting conductors. In pumping stations, use wireways in dry locations only. The main advantage of wireways is the ease of access for additions or removal of conductors and in adding circuits. A new conduit may be readily connected to a raceway and new power and control circuits added at minor cost compared to the cost of a new conduit run to the nearest power source.

If the mechanical specifications for the project require painting of all mechanical equipment and exposed metal in a particular area of the station, then the exposed conduit and wireways should be painted to conform with the other exposed metal. The zinc coating on galvanized metal must be properly pretreated and primed before painting. Any conduit entering a hazardous location must be provided with an explosion-proof seal fitting filled with a UL-listed sealing compound after all conductors are installed. Conduit runs must be sloped to drain, and breather/drain fittings (or, in nonhazardous areas, weep holes) should be provided at any low point in a run and prior to entering control cabinets, MCCs, and control panels.

All conduits below grade, whether buried, embedded in concrete, or exposed in underground rooms or spaces, are subject to infiltration of water and should be sealed inside the conduit around the conductors and/or cable(s) at each entrance into any enclosure below grade. O-Z/Gedney [1] has excellent sealing bushings and fittings that utilize sealing glands tightened by accessible set-screws that can be retightened during maintenance if necessary.

All conduits or cables entering a normally dry underground room or space in a pumping station and originating outside (buried) or in a room or space subject to possible flooding (such as a wet or clear well) should be installed in through-wall or floor sleeves equipped with sealing glands tightened by accessible set-screws that may be retightened. O-Z/Gedney [1] has suitable sealing sleeves.

Junction and Outlet Boxes

Junction and outlet boxes installed in pumping stations should be hot-dip galvanized cast iron. They should be surface-mounted, spaced a minimum of 6 mm ($\frac{1}{4}$ in.) from walls or mounting plates, and placed as high as possible in dry wells. Many pumping stations are subject to accidental flooding. Wet wells are even more subject to flooding, and, in addition, they are generally classified as hazardous areas because of the high probability of a large concentration of methane or other explosive gases resulting from accidental or illegal dumping of gasoline and solvents into the sewer system.

Avoid placing junction boxes in wet wells (or in any other other moisture-laden locations) by making a concerted effort to feed electrical systems from junction boxes in dry locations through conduit sealed at the entry into the hazardous area. But if there is no alternative, specify nonwicking cable, watertight motor and device terminations, and junction boxes filled with a waterproof sealant. The box itself must be rated as waterproof when properly installed.

One example of the need for a junction box in a wet well occurs in pumping stations with submersible pumps fed by portable cables. A junction box is necessary to transfer from the normal branch circuit conductors to the heavy-duty flexible, multiconductor cable to the submerged motor. One solution is to mount the junction outside the wet well and feed the cable into the wet well through a sufficiently large conduit.

Switchgear and Switchboards

"Switchgear" as defined in ANSI 141 is a general term covering switching and interrupting devices alone or in combination with other associated control, metering, protective, and regulating equipment.

Metal-enclosed switchgear provides sheet metal covering on all sides and top with access via doors and/or removable panels. Other classifications are indoor, outdoor, and walk-in with an enclosed maintenance aisle, as defined in ANSI 141. Further

classification provides for metal-clad switchgear that meets a number of specific safety requirements and is usually applied to systems operating at 1000 V or more.

Switchboards are not defined in ANSI 141 but, although of open construction, they may be provided with equipment similar to that included in switchgear. Modern switchboards are of dead-front construction (no live parts exposed) for operator safety and to meet NEC requirements. Switchboards are usually an assembly of molded-case circuit breakers, whereas switchgear is ordinarily a grouping of power circuit breakers and may include large motor starters. Incoming and outgoing line-termination space must be provided, with particular attention to the required minimum bending radius of large-diameter power cables. Buswork must be braced for the calculated maximum fault current, and the structure must be rated for the prevailing seismic conditions.

Service Entrances

Every building supplied with electricity directly from an electric utility must have a UL-listed service entrance. A service entrance usually consists of a utility-owned kilowatt-hour meter, any current or potential transformers (usually utility-owned), and a disconnect consisting of a fused switch or a circuit breaker. The neutral, if grounded, is grounded on the utility side of the disconnect. Service entrance equipment, including the meter sockets and instrument transformer provisions, must conform to utility requirements.

Meter and disconnect locations vary. For maximum security, both should be indoors, but utility and local regulations may dictate otherwise and must be followed. Sometimes other arrangements can be made.

A second building or structure on the same premises and under the same management must also have a UL-listed service disconnect. Neutral grounding can be at either building.

Equipment Installation

All electrical equipment must be equipped and installed per the NEC. Sufficient access and working space is required for ready and safe access for maintenance. Minimum working space in front of electrical enclosures not requiring rear access and facing metal or concrete walls is shown in Table 8-5. Illumination is required for all working spaces.

Table 8-5 Minimum Working Space in Front of Electrical Enclosures

Nominal voltage to ground	Dimensions, m			Dimensions, ft		
	Depth[a]	Width	Height	Depth[a]	Width	Height
0–150	0.92	0.76	1.91	3.0	2.5	6.25
151–600	1.07	0.76	1.91	3.5	2.5	6.25
601–2500	1.22	0.92	1.98	4.0	3.0	6.5
2501–9000	1.53	0.92	1.98	5.0	3.0	6.5

[a] Or more if required for the access doors to swing 90°.

At least one adequate entrance must be provided for access to the working space in front of electrical enclosures. A continuous passageway is usually sufficient. For equipment over 1.8 m (6 ft) wide, rated 1200 A or more and 600 V or less, and for all equipment over 600 V and 1.8 m (6 ft) wide, two entrances at least 0.61 m (2 ft) wide and 1.91 m (6.25 ft) high are required—one at each end. If, however, the depth shown in Table 8-5 is doubled, only one entrance is required, but the edge of the entrance nearest the enclosure must equal the depth shown in the table.

8-4. Standby Generators and Auxiliaries

Standby generators and their auxiliary equipment are needed in many pumping stations (see Section 9-9 for the need). Permanently installed units may be required by state or local laws or regulations, or they may be included in the station design because of the critical nature of the station, probability of loss of utility power, or the owner's or engineer's standards.

Portable generators provided with cord and plug connections are often used for backup power for small "package" pumping stations, and trailer-mounted engine-alternators of up to several hundred kilowatt ratings have been cord-connected with suitably rated cables and power plugs to serve medium-sized pumping stations during power outages. Bear in mind, however, that power outages occur in stormy weather that may make it difficult to reach a station, and if the power outage covers a wide area, many portable units would be required. Distance and time of response must also be considered. Finally, consider that legislation now allows competition between electric power suppliers—a policy that will inevitably lead to less reliable power because of the need to reduce costs to be competitive. Thus, the need for standby generators is likely to increase.

Codes and Legal Requirements

Generators in pumping stations are installed in accordance with NEC Article 701, Legally Required Standby Systems, or Article 702, Optional Standby Systems. If a local code requirement seems to require Article 700, Emergency Systems, be sure to negotiate a change with the authorities, and never call the generator an "Emergency Generator," because pumping station applications in no way involve life safety. Use Article 702, if possible, due to its minimal requirements and flexibility of operation. Use Article 701 only in case of a legal requirement to do so. See Article 701-2 (FPN), which refers to wastewater disposal and fire fighting applications, but take into account the wet well or clear well storage capacity, including any available storage in upstream piping.

Ratings

Continuous Rating

Two nameplate ratings are offered: prime and standby. The prime rating is that rating at which the engine-generator set is suitable for continuous operation at rated conditions. The standby (or continuous standby) rating is defined as that rating at which the generator set is suitable for continuous operation for only 30 days at the rated conditions. Continuous operation at the standby rating causes relatively high temperatures that age the generator insulation four to eight times faster than at the prime rating. This aging is not considered serious for a standby generator because of its limited use.

Refer to Chapter 14 for engine ratings.

Motor-Starting Rating

Motor-starting ratings are limited by either the engine or the generator. The engine must supply the shaft power required by the actual maximum load applied to the generator. The generator size is normally chosen to match the engine output capability. Selecting an oversized generator where motor-starting kVA demands are high, however, may be advantageous. The running load and the motor-starting requirements should be specified. The largest motor should be started first, if possible. Consult the manufacturer for selection and sizing of the engine-generator set. A 30% voltage dip at motor start is usually tolerable, but 20% is a more common design parameter to prevent contactor opening.

When specifying a generator for use with AFDs, require the generator supplier to work with the drive supplier to ensure that the generator is suitable for the application. Loads with harmonic current content over about 10% of the generator rating cannot be supplied by standard generators.

Fault Rating

With a brushless exciter whose output is dependent on terminal voltage, the worst event is a three-phase bolted fault (three-phase line-to-line short) at the generator terminals, because (with greatly reduced excitation) the terminal voltage collapses to a low value and cannot supply sufficient fault current to trip the breaker on the faulted circuit. Field-forcing equipment should be specified to sustain the rated excitation and current up to three times the generator's rated current output—sufficient fault current to trip the generator breaker. Ensure that downstream and generator circuit breakers are coordinated so that the branch circuit breaker trips first.

An undervoltage relay is recommended for any generator unit, large or small. It is actually a voltage-restrained overcurrent relay that has a current setting low enough to trip breakers and shut down the engine if overcurrent at less than full voltage occurs for a predetermined length of time.

Derating

Refer to Section 14-6 for derating engines for elevation above sea level and for temperature. Derate the generator 1% for each 100 m (3%/1000 ft) above 900 m (3000 ft).

Engine-Generator Controls

The engine instrument panel should never be mounted on the engine or on the engine pad, because engine vibration causes instruments to deteriorate. The instruments for the engine should include:

- Oil pressure
- Oil temperature
- Water temperature
- Intake manifold pressure and vacuum gauges, especially for large units.

A battery-charging alternator and ammeter should be included. Require safety controls for engine shut-down to be manually reset. Such controls should include:

- High water temperature
- Low oil pressure
- Overspeed.

The following instruments and controls should be mounted on the generator control panel:

- An ammeter with current transformers, as needed
- A voltmeter with potential transformers, as needed
- An ammeter/voltmeter phase selector switch
- A frequency meter (45 to 65 Hz)
- A voltage-adjustment rheostat
- An elapsed-time meter
- An annunciator or monitoring panel with fault-indicating lights for low oil pressure, high water temperature, overspeed, overcrank, and generator undervoltage. Standard annunciator panels may be specified to perform all the foregoing functions, and they usually constitute a much more economical system than a hard-wired custom alarm system.
- A fault reset pushbutton
- A generator output circuit breaker
- A three-position (Manual-Off-Auto) selector switch
- An engine-start switch
- An emergency stop pushbutton
- A voltage regulator
- An indicating-light test pushbutton.

Actuating the safety devices must shut down the generator set, indicate the cause of the shut-down by lighting the appropriate indicating light, and provide separate outputs for the remote alarm indication panel and the computer.

Automatic Transfer Controls

Code Requirements

Automatic transfer controls, per NEC 701-7 (1981), are needed for legally required standby systems. The interconnection of normal and standby power sources should not be possible in any mode of operation, except where parallel operation is intended and suitable automatic or manual synchronizing equipment is provided per NEC 230-83. In pumping stations, automatic transfer control equipment is commonly installed, except where portable generator sets are to be used.

Manual Transfer Switches

Manual transfer switches are sometimes used with portable generating equipment. Use kVA- (horsepower)-rated, quick-make/quick-break, heavy-duty switches or mechanically interlocked circuit breakers. If the transfer switch is part of the station service equipment, it must be UL-listed as service equipment or better, and it must have a separate UL-listed service disconnect (circuit breaker) ahead of it in a separate enclosure.

Automatic Transfer Switches

Complete automatic transfer switches (ATSs) are available in suitable enclosures (e.g., NEMA 1) as well as in open style for installation in motor control centers. They must conform to all of the requirements of UL 1008 and be so listed and labeled. Bypass isolation switches (that allow the ATS to be removed for repairs) are desirable for avoiding shutdowns for repairs.

Automatic transfer switches include a switching element, relays, and controls and are available in these forms:

- Molded-case circuit breakers
- Contactors
- Double-throw switches.

Automatic transfer switches should have the following ratings:

- Continuous rating, which is the current rating on a 24-h basis.
- Inrush rating, which is the ability to close a circuit with high inrush currents with minimum contact bounce and welding.
- Interrupting rating, which is the load-interrupting (not necessarily the fault-interrupting) ability under the worst conditions (e.g., locked rotor currents that occur when the motor rotor is locked in place due, say, to a clogged pump).
- Withstand rating, which is the ability to withstand the thermal and magnetic effects of downstream faults.

Automatic transfer switches should include a pause-in-neutral position (with an adjustable time delay that should be set for several seconds) that causes the motor to be disconnected from the power source during transfer and allows the motor voltage to collapse to a safe level prior to re-energization. Failure to allow the field to collapse can result in the motor field being out of phase with the new power source (which may well occur when transferring from a standby generator to the power utility). This out-of-phase condition can instantly destroy the motor by breaking the shaft or tearing out the windings. Because the pause-in-neutral feature may be

unavailable (and adds complexity and reduces reliability), consider other methods such as motor shutdown before transfer and an in-phase monitor relay.

Batteries

Starting batteries for a standby generator are ordinarily lead-acid type with heavy-duty ratings. Cranking batteries for moderate-sized generators are usually size 4D or 8D diesel starting types. These may be series-connected for voltages greater than their 12-V nominal rating. Cranking batteries should be located as close to the engine as possible—many times on a rack that is engine-mounted. Cranking currents up to 1000 A are not uncommon.

Batteries should be specified as part of the engine-generator package, with the specifications written to give the manufacturer some freedom in sizing, while making certain that cranking capacity is adequate for 10 attempts at starting and that the batteries are of the highest quality. Failure of an engine to start is usually a simple problem (e.g., fuel not reaching the cylinders), but one that cannot be resolved by remote control. Three failures to start should shut down the system and generate an alarm. After the operator has fixed the problem, enough cranking power must remain for a start with certainty.

Battery Chargers

Battery chargers vary widely, from simple half-wave rectifiers with a transformer and circuit overcurrent protection to precisely controlled units with voltage and current regulated and with timed alternate high-charging rate followed by a trickle-charging rate after initial recovery from a cranking cycle. Batteries are well-charged by a few hours of generator operation.

A battery charger may be specified with the engine-generator equipment, but if the engineer wishes to include this item as part of the electrical contract, the specifications for the battery charger must be carefully coordinated with those for the engine-generator.

8-5. Grounding

Grounding is a conducting connection, whether intentional or accidental, made between an electrical system and earth or to some conducting body that serves in place of earth.

Two types of grounding related to pumping station design are (1) system neutral grounding, and (2) equipment grounding. The system neutral ground is a connection to ground from the neutral point of a circuit, transformer, rotating machine, service, or system.

System Grounding

System grounding offers improved service reliability, improved fault protection, greater safety, and a reduction of transient overvoltages. The present trend is to use system neutral grounding at all voltage levels. Solid grounding of the system is used for systems operating at 600 V or lower, while at higher voltage levels a neutral resistance grounding unit is used between ground and the neutral of a power transformer or generator to limit the ground fault current to a low value. Grounding principles for ac systems are shown in Figure 8-20. The following methods can be used for system grounding:

- Solidly ground the system by connecting the neutral conductor to the grounding electrode conductor at the service.
- Solidly ground a separately derived system, such as a transformer, by connecting the neutral connector to the grounding electrode conductor. When the source is in the same building as the service, the grounding electrode conductor may be run back to the service.
- When the service conductors originate outside of the building and the neutral conductor is solidly grounded at the service, ground the neutral conductor at the transformer as well.
- When resistance grounding a medium-voltage (600- to 15,000-V) wye system, ground the system at the service by connecting the neutral conductor to the grounding electrode conductor through the resistor. For low-voltage (below 600 V) systems, solidly ground to a grounding electrode.

Equipment Grounding

Equipment grounding is vital to protect people as well as the equipment by: (1) limiting the voltage between equipment and earth to a safe value, and (2) providing a low-impedance return path for fault current to operate the protective devices quickly. The methods of equipment grounding include the following:

- Run the grounding conductor with the circuit conductors.
- Use the metal conduit (if it is electrically continuous) as a ground conductor. If it is not continuous, bond each section to the grounding conductor.

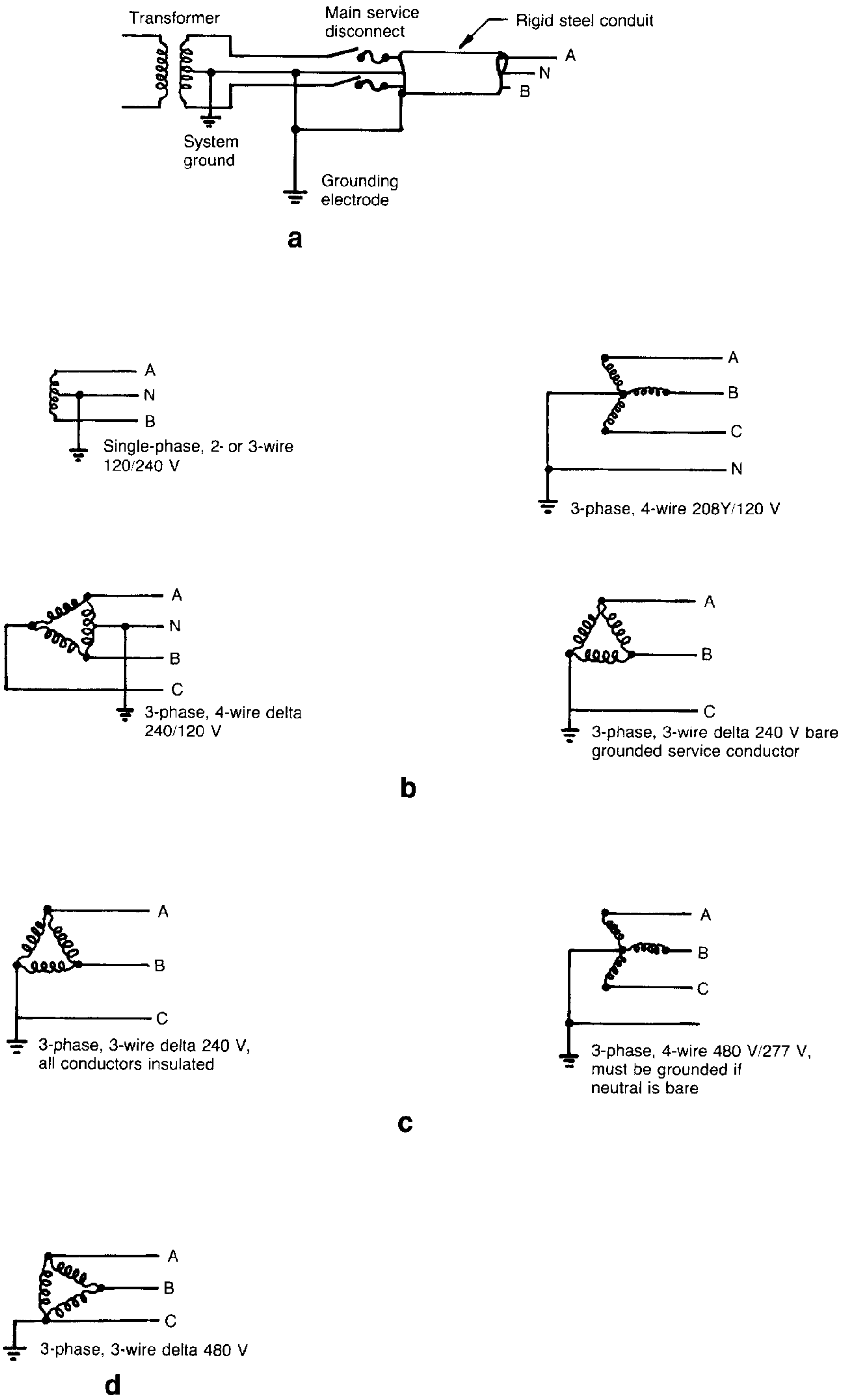


Figure 8-20. Grounding of low-voltage ac systems. (a) Grounding principles; (b) systems that *must* be grounded: 150 V or less to ground or bare service conductor used; (c) systems recommended to be grounded: not over 300 V to ground; (d) a system that may be grounded: over 300 V to ground.

Ground Fault Protection

System Protection

The NEC requires ground fault protection of 277/480 V electrical systems with services 1000 A and higher. It is, however, good engineering practice to examine the possible consequences of ground faults in any polyphase system where an arcing ground fault may burn long enough to create major fire damage.

Ground Fault Circuit Interrupter

A ground fault circuit interrupter (GFCI), or simply a ground fault interrupter (GFI), is used on 120-V outlet circuits to interrupt the circuits when a fault current to ground exceeds a predetermined value but is less than that needed to operate the overcurrent protective device.

Class A GFCI units, designed to trip when line-to-ground current exceeds approximately 5 milliamperes (mA), protect personnel from electrical shocks. The average person can, without harm, withstand currents of 5 mA or more for the time needed to trip the unit. Hand-held power tools and their power cords can be hazardous if their insulation is defective, particularly if the worker is in contact with a conducting surface such as the ground, a concrete floor, or a metal pipe. In pumping stations, GFI units should always be required in outdoor and below-grade receptacles, in hazardous (per NEC 500) locations, in all areas with bare concrete floors and/or metal work benches, and where specifically called for in the NEC.

The ground fault interrupter device may be a part of the circuit breaker, or it may be inherent in a receptacle unit. Receptacles are cheaper and must be used if the circuit run is longer than about 30 m (100 ft). If circuit breakers are used, each circuit must have its own neutral because multiwire branch circuits do not work with a GFCI unit in the circuit breaker. The receptacle unit may be of the type that provides protection only for equipment plugged into it, or it may be a feed-through unit that protects all ordinary receptacles beyond it in the circuit. Two methods of ground fault protection are illustrated in Figure 8-21.

The NEC may require ground fault protection on heat trace circuits for water pipe freeze protection or for de-icing applications. Use Class B GFCI circuit breakers (which trip at 30 mA) for these applications, or ensure that the heat trace control package has this protection.

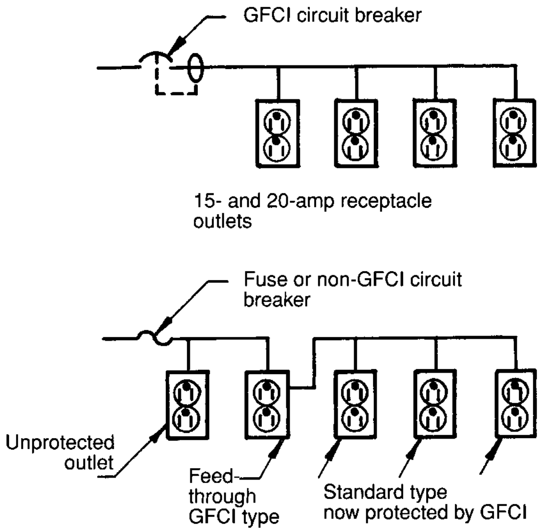


Figure 8-21. Ground fault circuit interrupters.

Surge and Lightning Protection

Lightning arresters and surge capacitors are used to limit the peak voltage of an impulse and to reshape the impulse to absorb the energy. They are a necessity in lightning-prone areas or if the pumping station is equipped with any electronic gear such as AFDs, and they are highly recommended in all pumping stations because of their very low cost (relatively speaking) and the excellent protection they afford motors. They are located at the service disconnect.

The following sources produce sustained overvoltages (surges):

- *Intermittent ground faults:* Intermittent arcing ground faults on 480-V ungrounded systems can alternately strike and clear, leaving dc charges on the line-to-ground capacitance. Surges or overvoltages of five to six times normal may be produced, but a properly sized grounding resistor can eliminate the overvoltages.
- *Series inductive capacitance faults:* The inadvertent connection of an inductance from line to ground on an ungrounded system can cause surges or voltage increases by resonance effect.
- *Circuit switching:* Circuit switching by an air contactor or circuit breaker does not cause overvoltages because there is no trapped charge on the circuit when the arc is extinguished at a current zero. Vacuum contactors in circuit breakers or starters and current-limiting fuses cause overvoltages. Vacuum contactors force the current to

zero before a natural zero occurs. Current-limiting fuses have built-in overvoltage surge suppressors.

- *Lightning:* Lightning is the discharge of a highly charged condenser. The clouds are one plate, the earth is the other.

Lightning Protection

The following discussion is related to basic design considerations for pumping station buildings of ordinary construction. Protection of high-voltage switchyards is usually the responsibility of the power company. The details of lightning protection are quite complex. An art rather than a science, it is best left to a specialist. The role of the pumping station designer is normally limited to specifying the quality of both the system and the materials.

Codes. In NFPA 78 (Lightning Protection Code) a distinction is made between the following building classes:

- *Class I:* An ordinary nonsteel building less than 23 m (75 ft) high.
- *Class II:* An ordinary building more than 23 m (75 ft) high or a building of any height with structural steel framing where the steel framing can be used for the down conductors. All steel is insulated from the ground by concrete.

Design Objectives. The design objectives are: (1) to provide a continuous low-impedance path for the discharge current to follow in preference to higher impedance paths such as wood, brick, or concrete; and (2) to provide air terminals for building parts most likely to be struck, especially ventilators, gables, and roof edges.

Electrical Materials and Components. Electrical materials must be resistant to corrosion under the anticipated environmental conditions or must be suitably protected. The use of aluminum is restricted because of the problems it creates in all electrical work, and furthermore, it must not be used in contact with the earth or embedded in concrete or masonry. Always use copper or bronze. All materials used must be protected from mechanical injury. Typical components of a lightning protection system are air terminals, down conductors, secondary conductors, and ground electrodes (see Figure 8-22).

8-6. Lighting and Power Outlets

Interior Lighting Equipment Selection

Many factors (including room size, ceiling height, tasks performed, frequency and length of room occupancy, and special considerations such as damp, cor-

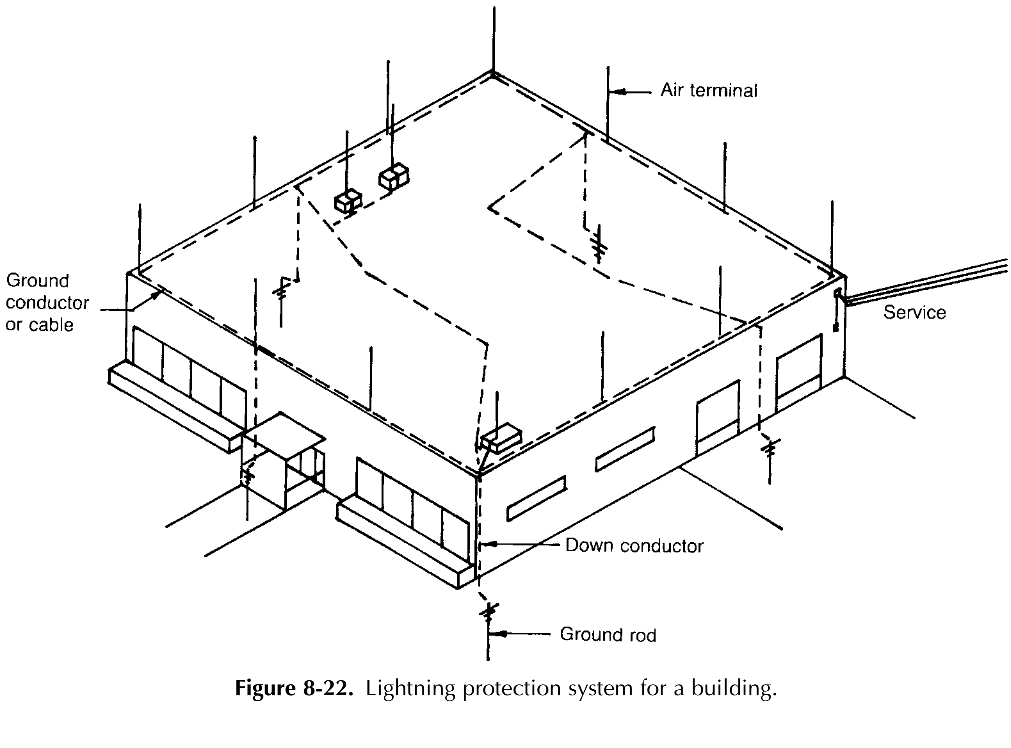


Figure 8-22. Lightning protection system for a building.

rosive, or hazardous areas) determine the type of lighting fixtures and lamps to be used. Fixtures are varied and numerous, but the industrial type (ceiling or wall mounted) is the most suitable for pumping stations. Lamps are described by watts used and type (e.g., 100 W incandescent), but they are rated by lumen output. The number of lumens per watt is called the "efficiency of the light source."

There have been many changes in available lamps since 1985. High-efficiency lamps with novel shapes and bases and unusual wattages have been produced. Recent federal legislation has outlawed the production of a number of low-priced incandescent and fluorescent lamps that once were almost engineering standards. State energy conservation laws have proliferated. The net result is a greater choice of more expensive lamps producing light of higher quality than ever before. It is still possible, however, to make recommendations regarding pumping station lighting.

For new installations, 32 W T-8 fluorescent lamps should be used in industrial fixtures in such nonhazardous areas as control rooms, pump rooms, and dry wells. Hazardous-location fixtures are available but are expensive. Use high-power factor ballasts. Consider electronic ballasts where fixtures have high usage. These ballasts are now reliable, are cost-effective, but still have a high first cost. Low-temperature magnetic ballasts are available for areas in which air temperatures can fall below 10°C (50°F), but verify that lamps can be used with them. Fluorescent lamp life is 20,000 h.

Use the new compact fluorescent lamps in sizes 7 to 26 W single or dual in small rooms with fairly high usage, corridors, and outside for security where the brighter, high-pressure sodium lamps are not wanted. Compact fluorescent lamps have excellent color and 10,000-h lives. Wattages higher than 26 are available.

Use incandescent or metal halide lamps in wastewater pumping station wet wells, in clear wells, and in small areas not often used. Incandescent lamps have low efficiency and short lamp life (750 to 2000 h), and several commonly used wattages have recently been outlawed. Use guards where needed. Metal halide lamps should have back-up quartz lamps where needed, and emergency lights require an off-delay to permit metal halide lamps to restrike. Use fixtures suitable for damp or wet locations where needed. In wastewater pumping station wet wells, use fixtures approved and labeled for use in Class 1, Division 1, Group D hazardous locations per NEC 500.

As alternatives to fluorescent lamps, use high-pressure sodium lamps outside as security lighting, in larger rooms with ceiling heights of 3 m (10 ft) or more where the glare and color is not objectionable, and in hazardous locations such as wastewater wet wells. Never use them where machinery is in motion, because there is a dangerous stroboscopic effect. These lamps are now available as small as 35 W. In larger sizes they are the most efficient commonly used lamps. Lamp life is 24,000 h, and color is constantly being improved. Do not use mercury vapor lamps in new construction. They are relatively inefficient and are now considered obsolete. Metal halide lamps have good color; are available in 32 W, 100 W, and in larger sizes; and have 10,000 h of life, but they are less efficient than high-pressure sodium. Color is the only reason to choose them over high-pressure sodium. Low-pressure sodium lamps emit objectionable color.

Fixtures suitable for use in damp and/or corrosive areas are available with all types of lamps, and the selection depends on location and preference. The fixtures in damp areas must be vapor tight and those in corrosive areas should be made corrosion resistant with a coating of PVC (or the equivalent) on all metal parts.

Energy Conservation

Lighting energy conservation is mandatory in some states and may be imposed throughout the United States by the federal government. Obtain requirements and forms from the appropriate regulatory bodies, and consult the local building inspector. Some states (California, for example) require two-level switching in heated or air-conditioned rooms larger than $9\,m^2$ ($100\,ft^2$), containing two or more fixtures, and drawing $13\,W/m^2$ ($1.2\,W/ft^2$). Such requirements might be waived for unattended pumping stations.

Fixture Location

Space fixtures relatively evenly with due regard for motor control centers and pipe risers. Pumps, motors, and auxiliaries such as chemical feeders and motor control centers need more light than piping areas. Do not obstruct pump and motor lift space, and do allow for bridge cranes and the like. In high-bay rooms such as motor and engine rooms, make certain that lamps in the fixtures can be replaced from the crane working platform.

Switching

Generally, locate a switch at each entry on the door latch side. Locate wet well or clear well switches

inside the pump building (but not in the wet well or clear well) and include a labeled pilot light. Always provide unswitched lights at stairways and along corridors, especially in machine rooms and below-grade rooms where hazards may be present. Sketch a wiring diagram and carefully determine number of wires from the diagram. If the fixture is identified by the switch that controls it, these diagrams need not be included on plans.

Exterior Lighting

Entry or Security Lights

The following suggestions pertain to entry or security lights:

- Locate the lights at each principal entrance.
- Lights should be able to be switched or put on photocell control.
- Use vandal-resistant lenses for lights.
- Use 52- or 90-W incandescent lamps in smaller stations where fixtures are manually switched. Use high-pressure sodium fixtures in stations with photocell control.

Roadway and Parking Lots

There are many types of roadway and parking lot light fixtures utilizing both utilitarian and architectural designs. Use only high-pressure sodium lamps. Photocell or time-clock control is usual. For design information and selections, refer to manufacturers' catalogs and the IES handbook.

Emergency Lighting

Codes and Legal Requirements

Emergency lights are sometimes required by state and/or local law in structures similar to pumping stations where failure of power to the main lighting system would leave building exits in total darkness. Where required, they are subject to NEC Article 700 and NFPA Life Safety Code No. 101, Section 5-10.

Self-Contained Lighting Units

Approved lighting units are available complete with battery, charger, controls, and attached or separate lamp heads. Either unit-mounted or separate lamp heads are available, and the types of lamp heads include: (1) 8- to 25-W sealed beam models, (2) nonsealed beam models with automotive-type bulbs, (3) very high efficiency halogen models, (4) decorator models, and (5) explosion-proof (Class 1, Division 1, Group D from NEC Section 50) models.

Exit Signs

Incandescent and fluorescent exit signs are available with four lamps—two for line voltage and two for battery voltage. They are also available with an integral battery, charger, transfer controls, and down light. The atomic type of exit lights should be considered if the owners are made aware that there is a disposal problem for the luminous tubes at the end of their lives. They require no connections to a power source. LED (light-emitting diode) exit signs are more expensive, and they have no down light, but with lamp life exceeding 20 years, and as they draw 2 W per face, they are very cost-effective over their lifetimes.

Transfer Controls

Transfer controls consist of a relay with a voltage sensor that transfers to battery power and energizes lamp heads when line power fails. They are available with battery low-voltage protection that shuts off the lights to prevent deep battery drawdown.

The following are suggestions and recommendations for emergency lighting:

- If the pumping station is underground, use battery packs wired to the normal lighting circuit (but ahead of the light switch) to illuminate the stairs or ladder.
- Use at least two lamps at each lamp location so that no area is left in darkness if a lamp fails.
- Provide at least 10 lux (1 foot-candle) at floor level for each exit.
- Exit signs are not usually provided in pumping stations. If required, they may not have to be connected to the emergency lighting system.
- Use a sealed, maintenance-free battery rated for 10-yr life.
- Use a solid-state, automatic, two-rate trickle charger with trickle and high-charge indicators and a test switch.
- Use an automatic transfer relay with low-battery protection. Use delayed retransfer when the normal light source is metal halide or high-pressure sodium to allow for restrike time (i.e., the time required for lamp to relight).
- Use NEMA 4× enclosures in cold or damp areas.

- Except in hazardous areas, use sealed beam or sealed halogen lamp heads.
- Do not locate battery units in wet, corrosive, or hazardous areas. In these hazardous areas install only lamp heads connected to a battery that is located in dry, indoor, nonhazardous areas.

Convenience Outlets

Low Voltage

For 120-V outlets, use NEMA 5-20 receptacles, specification grade, on 20-A, single-pole circuits even though the NEC allows NEMA 5-15 if two or more are on a circuit. No more than five should be on a circuit. Locate them near pumps, electrical gear, and outside near pipe or wet well or clear well access. Generally place them so that any location that might require power can be reached with a 15-m (50-ft) extension cord. Do not locate them in a wet well or clear well. Install weatherproof covers if they are subject to weather or splashing, such as where equipment is hosed down.

Special Outlets

The following apply to special outlets:

- *Welding:* The voltage and ampere rating must be compatible with welding machine requirements.
- *Telephone:* Use NEMA 5-15 outlet on 15-A circuits where more than two telephone lines will be brought in.
- *Emergency lighting:* Where permitted by codes, use single NEMA 5-15 outlets on the same circuit as the room lighting. Permanent connection to the supply circuit is required for legally required emergency lighting.
- *Battery charger:* The charger should be sized (and specified) according to battery requirements.
- *Light fixtures:* Use twist-lock outlets near chain-suspended fixtures, and consider hard-wiring two fixtures together.

8-7. Electrical Circuit Diagrams

Single-Line Diagrams

A single-line diagram is used to show the electric power system from source to load in symbolic form. Conductors and their connections are omitted, and only circuit elements, their relationships, and the flow of energy through the system are shown. The diagram should be simple, utilize standard symbols, and present a complete picture that enables the observer to: (1) assess the blocks of power required, (2) identify the major feeders, and (3) ascertain the main protective devices responsible for plant outages.

An example of a single-line diagram of a radial circuit arrangement of a small pumping station is shown in Figure 8-23. The radial system is the simplest and most economical type of circuit arrangement. It offers good reliability and probably is the type most often used in small pumping stations. The drawback of the radial system is that a failure in the utility feeder, main breaker, or motor control center bus shuts down the whole station. The flexibility of being able to transfer manually or automatically to a standby power source is illustrated in Figure 8-24.

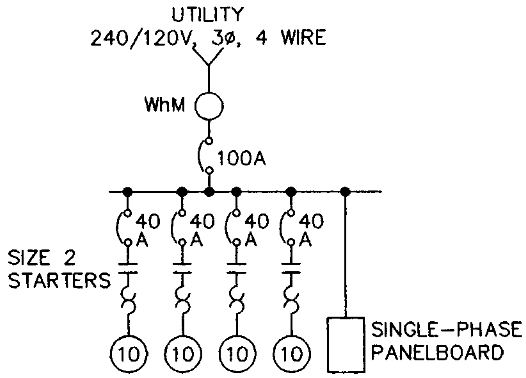


Figure 8-23. A typical single-line diagram for a small pumping station. Courtesy of Brown and Caldwell Consultants.

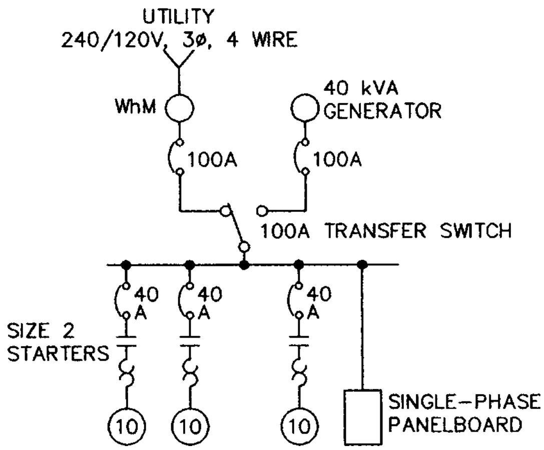


Figure 8-24. A typical single-line diagram for a small pumping station with standby power. Courtesy of Brown and Caldwell Consultants.

Similar flexibility is offered in Figure 8-25 with added reliability from using two utility power services. Each feeder is provided with a full-sized transformer and main incoming circuit breakers. The motor control center includes the two normally closed main breakers and a normally open tie breaker interlocked to prevent parallel operation of the two incoming lines. In the event of a power outage on a utility feeder, its associated main breaker can be opened (usually automatically) and the tie breaker closed, which allows the entire pumping station to be fed from the second source feeder. This pumping station includes two variable-speed pumps (one dc drive and one wound-rotor, slip-power recovery) and two constant-speed pumps. As all of the motors are of equal power (a blunder as explained in Chapter 15), the standby pump must be one of the variable-speed units, and it must be assumed that the other variable-speed pump is out of service. Hence, under a critical condition, there are one variable- and two constant-speed pumps operational. A typical sequence of starting pumps as flow increases might be as follows:

- One operational variable-speed pump accommodates low flow.
- A constant-speed pump is added and the variable-speed pump operates at its lowest permissible speed (usually at a flow rate of about 35% BEC) in an on–off mode until the flow increases to about 135% of a single pump's capacity. Now both pumps can operate continuously.
- The second constant-speed pump is added, and the variable-speed pump again operates at minimum capacity in an on–off mode until the flow requires it to operate continuously.
- When both variable-speed pumps are available, operational flexibility is improved so that on–off operation can be avoided.
- If the capacity of the variable-speed pumps was about 50% greater than that of the constant-speed

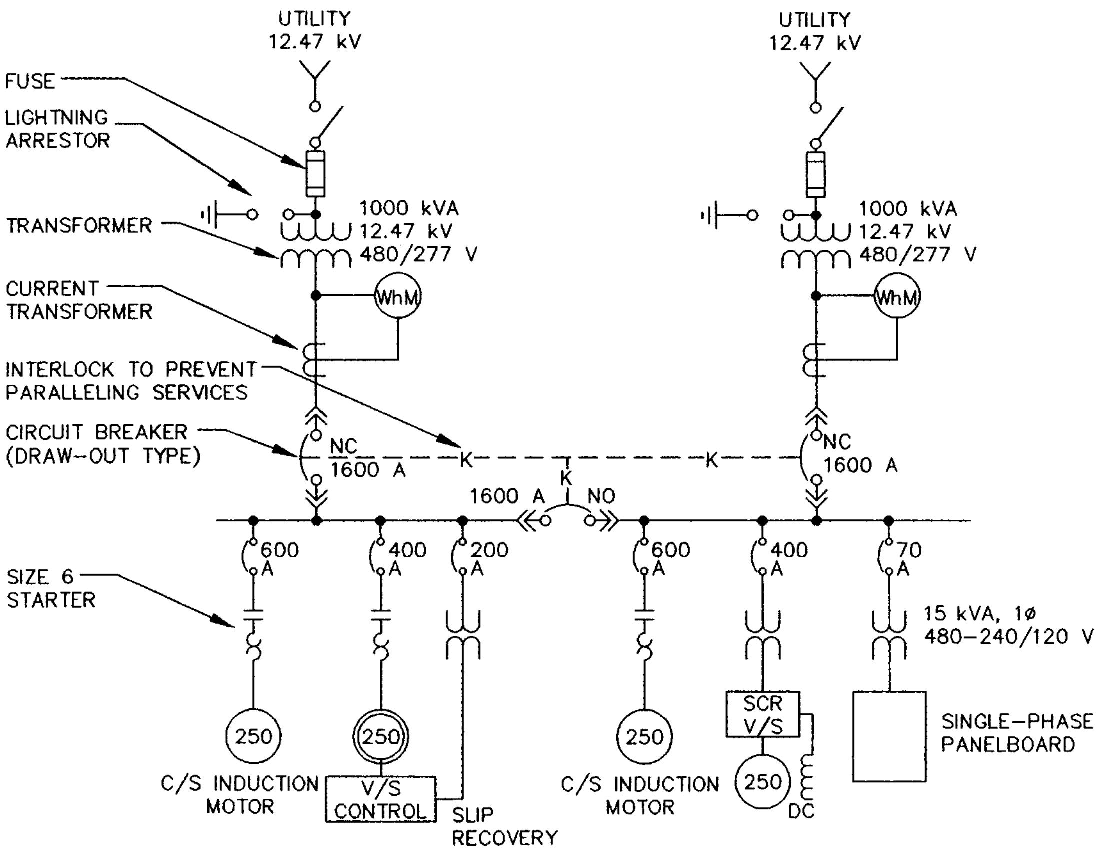


Figure 8-25. A typical single-line diagram for a large, low-voltage pumping station with dual incoming service. NC, normally closed; NO, normally open; SCR, silicon-controlled rectifier. Courtesy of Brown and Caldwell Consultants.

pumps, there would be no need for on–off operation (see Section 15-5).

The pumping station in Figure 8-25 is, however, an old-fashioned design. Wound-rotor and dc motors would be avoided nowadays in favor of induction motors driven by AF converters. Note, too, that the hydraulic efficiency of the system is less than it would be if all four drives were variable speed (see Figure 15-8 for proof). Furthermore, three variable-speed pumping units is a viable alternative with savings in: (1) size of wet well, (2) size of dry well, (3) number of pumps and amount of piping, (4) complexity, (5) upsets at the treatment plant, and, of course, (6) cost of power.

Single-line diagrams are a very necessary and useful tool in short-circuit calculations and protective device coordination studies, because they are a skeleton representation of the system, and thus it is simple to follow through a number of levels of protection. Protective relay settings are derived from the diagram and the analysis of the time-current graphs of the various fuses and circuit breaker trip elements or relays.

Control Diagrams

Control diagrams represent the exact connection of all the elements of a control system. Interconnections of the elements with other systems that are diagrammed elsewhere are also shown. Simple motor control diagrams are shown in Figures 8-26 and 8-27. All the elements of the control system are usually shown horizontally on the diagram and the interconnections are drawn horizontally and vertically. The resulting diagram resembles a ladder and is thus termed a "ladder diagram" by many users.

All elements are identified and the function of each is either noted on the diagram or included in a supplementary explanation of the control element and its function. All lines of a ladder diagram should be numbered consecutively. Auxiliary relay contacts are usually shown to the right of the main diagram on or near the same line as the circuit element. The contacts are then individually identified with interconnection information. Relays are identified as CR1, CR2, and so forth, for control type units, and TR3, TR4, and so forth, for timing units. Timing relays are further identified at the coil symbol with the time setting and at the contact with information, such as TCE (time closing on energization), TOE (time opening on energization), and TCD and TOD for the respective operations on de-energization of a relay. The ladder diagram allows the engineer to follow the sequence of operation of a complex control system logically. Ladder diagrams are often used in training maintenance personnel, but their most important use is for analyzing systems that have failed to operate.

Many PLCs are programmed on a ladder diagram basis for the convenience of technicians who have no specialized training in computer programming.

Coordination

Coordination of overcurrent protective devices and of protective relays greatly improves the reliability of a pumping station, and it should always be required by the project engineer. In a coordinated system, the branch circuit protective device (usually a circuit breaker) trips off a faulted circuit. Should it fail to trip in a reasonable time, the upstream overcurrent device will trip, but the outage area will then be larger. Thus, all pertinent overcurrent devices have proper time delays and are said to be coordinated on a time basis. Fuses easily coordinate with each other and with a single downstream circuit breaker. Non-adjustable circuit breakers do not coordinate with each other. Short-circuit current to a faulted branch circuit flows through the branch circuit breaker, the feeder circuit breaker, and the service circuit breaker, and they may all trip instantaneously, thereby shutting down the entire station. Fortunately, most faults are ground faults, and the ground-loop impedance keeps the short-circuit current to a sufficiently low value to prevent that scenario. Many commercial power systems are designed with 10,000-A branch circuit breakers and fully rated main circuit breakers for low first cost, and the main must trip to protect the branch circuit breaker for a branch circuit fault drawing more than 10,000 A. This cascade system is obviously unacceptable where a reliable power system is needed.

Coordination studies are performed to verify coordination using manufacturer-supplied time–current curves on log–log graph paper. The studies verify circuit trip and fuse ratings and predict settings for adjustable circuit breakers and protective relays.

8-8. Power and Control System Practices

Voltage Terminology

System voltage: The root-mean-square phase-to-phase voltage on an ac electric system. All subsequent defined voltages are in terms of root-mean-square phase-to-phase values unless noted otherwise.

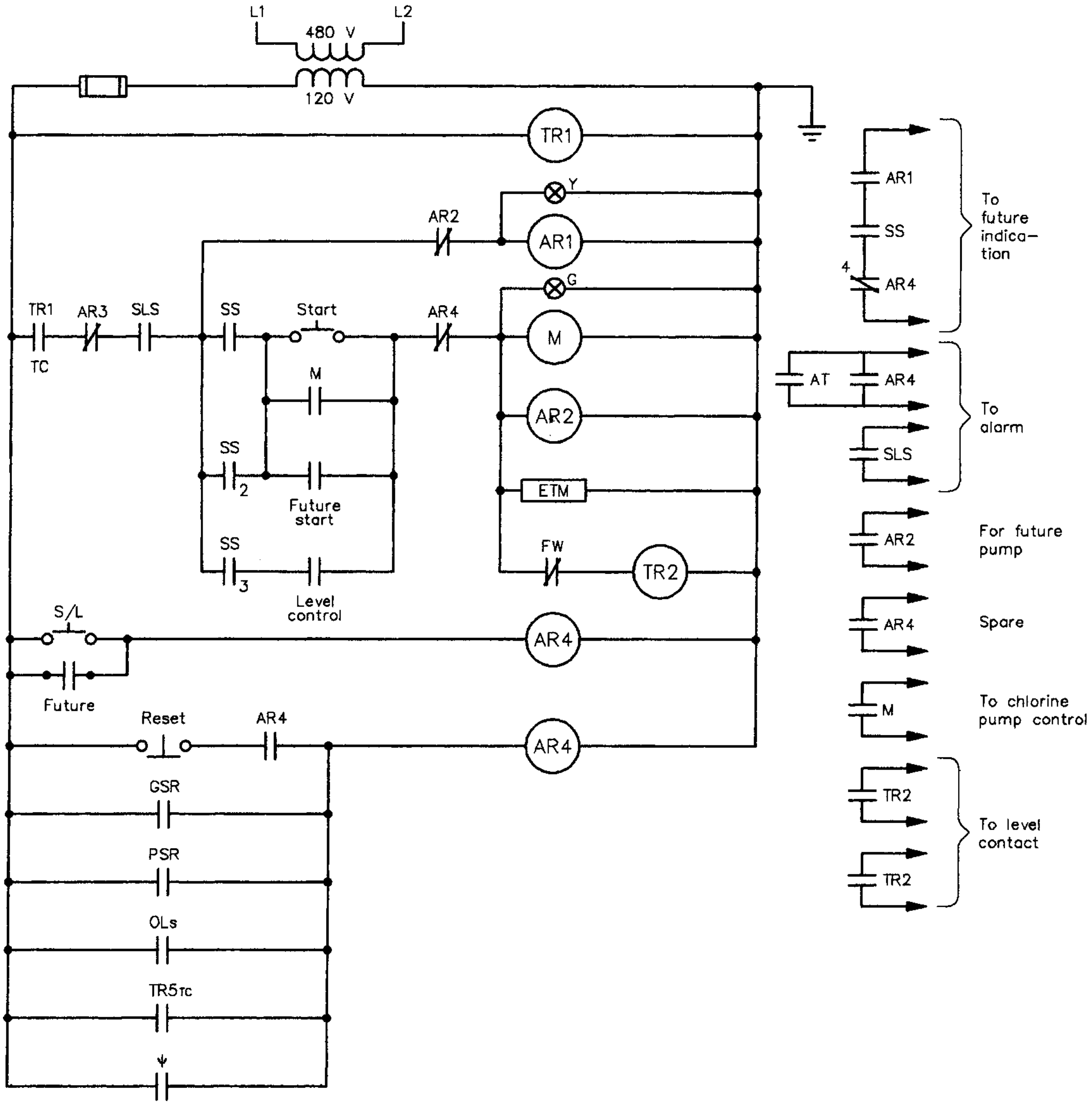


Figure 8-26. A typical schematic diagram for booster pump number 1. Booster pumps 2 and 3 are similar. L, line; OL, overload; PSR, power-sensing relay; SLS, suction valve limit switch; TC, time closing; (ψ), motor temperature detector (see Figure 8-25 for other abbreviations). Courtesy of Brown and Caldwell Consultants.

Nominal system voltage: The voltage by which the system is designated and to which certain operating characteristics are related.

Maximum system voltage: The highest voltage that occurs on the system under normal operating conditions, and the highest voltage for which equipment and other system components are designed for satisfactory continuous operation without derating of any kind.

Service voltage: The voltage at the utility's point of service to the customer—ordinarily the metering point.

Utilization voltage: The voltage delivered to the line terminals of the utilization equipment.

Standard Nominal System Voltages

Standard nominal system voltages supplied by most electric utilities are given in Table 8-6. Utilities usually maintain them to within ±5%, but where energy conservation is of concern, the voltages are maintained to within +0, −5%.

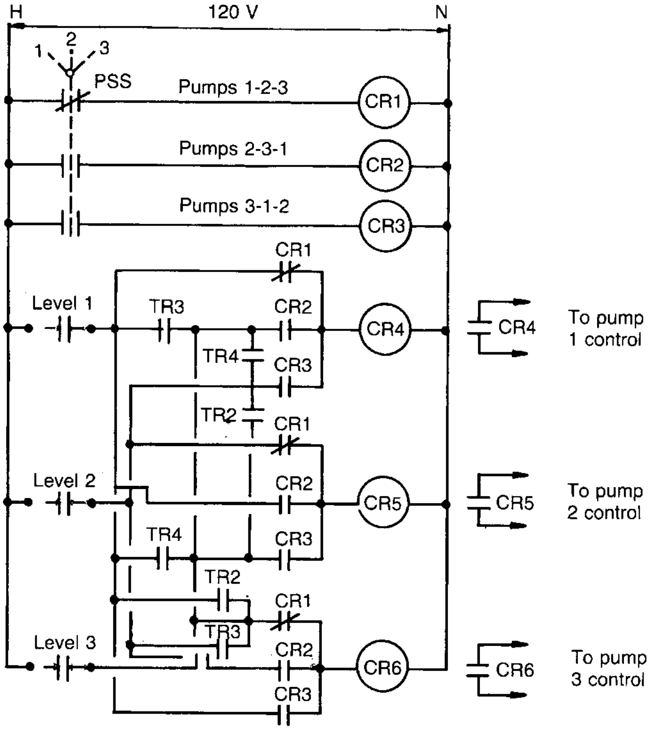


Figure 8-27. A typical schematic diagram for pump selection and level control. CR, control relay; H, hot; N, neutral; PSS, pump sequential selector switch (see Figures 8-25 and 8-26 for other abbreviations). Courtesy of Greeley and Hansen Engineers.

For other system voltages and for exceptions to the values in Table 8-6, refer to ANSI/IEEE Std 141-1986. In some localities, the voltages offered may differ from the ANSI standard voltages.

Station Voltage Selection

The preferred nominal system voltage for an electric service depends on the electrical load. For a pumping station where all electrical devices may operate simultaneously, the connected load is used. It is the sum of the ratings of all electrical equipment in the pumping station computed to the nearest hp, kW, or kVA. Motors are counted at their horsepower rating regardless of their actual loading, and all other devices are counted at their nameplate rating in kW or kVA. Conversions are usually made on a one-to-one basis (1 hp = 1 kVA). Welding machines are rated in hp at 1 kVA per hp.

Recommendations for pumping station service voltages are given in Table 8-7. Utility limitations may be different and must be checked.

Voltage Ratings for Utilization Equipment.

Nameplate ratings applicable to motors are listed in ANSI/IEEE Std 141-1986, Table 7. A few of these ratings are given in Table 8-8.

Motors used on 208-V systems should be rated 200 V. Motors rated 220 V or 230 V do not perform satisfactorily and should not be used.

Load Estimating

Load estimating evolves in several major stages. Preliminary load estimating is done during the predesign stage of development of proposals for the pumping station design, usually with very preliminary estimates of the total plant pumping horsepower. It can be accomplished largely on the basis of discussions with the project managers and reference material from company design files on similar previous jobs.

The complexity of load estimating depends on the size of the station and the experience level of the

Table 8-6 Standard Nominal System Voltages

Single-phase	Three-phase
120 V, 2-wire	208/120 V, 4-wire
240/120 V, 3-wire	240/120 V, 4-wire
208/120 V, 3-wire[a]	240 V, 3-wire delta[b]
	480/277 V, 4-wire
	480 V, 3-wire delta[c]
	2400 V
	4160 V
	12,000 V

[a]Available in some inner-city areas with three-phase networks.
[b]Common in rural areas with pole-mounted transformers. Insist on transformers to ensure balanced voltages.
[c]By special request. Be sure the utility does not ground the neutral at the transformer.

Table 8-7 Pumping Station Service Recommendations

Load	Voltage
5 kVA (7 hp)	240/120 V, single-phase, 3-wire or 208/120 V, single-phase, 3-wire
40 kVA (50 hp)	240/120 V, three-phase, 4-wire or 208/120 V, three-phase, 4-wire or 240 V, three-phase, 4-wire
400 kVA (500 hp)	480/277 V, three-phase, 4-wire[a]
Above 400 kVA (500 hp)	4160 V, three-phase[b] or 2400 V, three-phase[b]

[a]Should be used for motors as small as $\frac{1}{2}$ hp. Provide 240/120V, single-phase, 3-wire or 208/120 V, three-phase, 4-wire for pumping station auxiliaries, did from the 480/227 V system.
[b]Use for large motors. Provide 480/277 V, three-phase, 4-wire for larger pumping station auxiliaries, derived from the high-voltage system. Provide also 240/120 V, single-phase, 3-wire or 208/120 V, three-phase, 4-wire for smaller pumping station auxiliaries derived from the 480/277 V system.

Table 8-8 Nameplate Ratings for Motors

Nominal system voltage	Nameplate voltage	Power range kW	Power range hp
Single-phase motors:			
120	115	≤ 0.37	≤ ½
240	230	≤ 1.1	≤ 1.5
Three-phase motors:			
208	200	110	150
240	230	150	250
480	460	300	400
2400	2300	1500	2000
4160	4000	3000	4000

project designers. For many years, successful professional design firms have developed estimating information from preliminary data sheets. These sheets allow for a variety of useful information relating to the various plant drives. The principal load is usually associated with the pumping units. Other important loads depend on whether the station is in an area of extreme summer heat or extreme winter cold, where extensive cooling or heating equipment may be required. The number of individual pumping units affects the size of the standby generator. The number of pumps also affects the complexity of the control and monitoring system of the station. All of these factors affect the electrical load estimate as well as the electrical system construction cost.

Data sheets should be generated in a computer system so that the data are accessible to all employees who are responsible for a portion of the total estimate.

Load estimation for small or "package" pumping stations is very simple, as there is frequently no special auxiliary equipment to complicate the station design.

A few checklist items that apply to load estimation for many pumping stations follow:

- Number of pumps and required horsepower of each
- Number of pumps operating at any time and sequence of starting
- Special requirements, such as adjustable speed drives
- Auxiliary drives required
- Preliminary layout of structure and number of operating levels
- Special use areas (such as shops) and their electrical loads
- Heating, cooling, and ventilating requirements
- Standby power requirements (if any)
- Applicable electric utility rate schedules
- Available electric utility voltage levels and distance to their point of service
- Inside lighting requirements/outdoor lighting requirements
- Review local electrical codes and try to get information from the inspection authority on special interpretations that may affect the electrical construction costs. An example is the treatment of possible hazardous areas such as Class I, Division 1 versus Class I, Division 2.

8-9. Reference

1. O-Z/Gedney, Main Street, Terryville, CT 06786.

8-10. Supplementary Reading

1. Beeman, D., *Industrial Power Systems Handbook,* McGraw-Hill, New York (1955).
2. *IES Lighting Handbook,* Illuminating Engineering Society, New York (latest edition).
3. *Lightning Protection, Institute Standard of Practice,* 3rd ed., Lightning Protection Institute, P.O. Box 458, Harvard, IL 60033-0458 (1987).
4. McPartland, J. F., *National Electric Code Handbook,* 22nd ed., McGraw-Hill, New York (1996).
5. Smeaton, R. W., *Switchgear and Control Handbook,* 3rd ed., McGraw-Hill, New York (1996).
6. Catalogs of various manufacturers, such as General Electric, Square D, Westinghouse.

Chapter 9

Electrical Design

STANLEY S. HONG
PHILIP A. HUFF
PAUL C. LEACH

CONTRIBUTORS
Robert S. Benfell
Mayo Gottliebson
Stephen H. Palac
Patricia A. Trager
Alan Vause

The primary purpose of this chapter, as with Chapter 8, is to provide project leaders with some understanding of electrical design, to aid communication with electrical specialists, and to help in coordinating electrical design with other disciplines. The chapter is appropriate for civil engineers, and it may also be appropriate for electrical engineers with limited experience in power engineering. A working knowledge of the principles of electricity, discussed in Chapter 8, is a prerequisite.

The topics discussed here include the coordination of electrical design both with other disciplines and with utility companies (Sections 9-1 to 9-4), the suitability of materials (Section 9-1), harmonics (Section 9-11), and construction services such as field testing (Section 9-12). Most of the essential electrical calculations for a pumping station of moderate size are given as worked examples. The examples include electrical load estimation, overcurrent protection, lighting, engine-generator sizing, and conductor, fuse, and breaker sizing for main and branch circuits.

Terminology is given in Sections 2-2 and 8-1, and abbreviations are defined in Section 2-1. The text and computations in the examples are in U.S. customary units for correlation with present codes and standards in the United States. Conversions to SI units are given in Appendix A, and some are also given in this chapter.

References to a code, a standard, or a specification are given here in abbreviated form, such as NEC (for National Electrical Code) or NFPA Standard 493-1978 (for a National Fire Protection Association standard). The coded numbers are entirely sufficient for identifying the reference.

Much of the material in this chapter is referenced to the NEC, so read the referenced passages in sequence with this text. Keep in mind that the NEC is subject to local interpretation. Check with local authorities for applicable codes and regulations. The design must follow safety and protection standards established by OSHA and UL. Standards and recommendations published by organizations such as IES, IEEE, NEMA, ANSI, and NFPA should also be followed.

9-1. Final Construction Drawings

Long before final construction drawings are prepared—in fact, at the beginning of the project—the project leader should consult the electrical engineer to ensure that an adequate space is reserved for electrical

equipment and to forestall interference with mechanical equipment and the building envelope. During the appropriate design stages, the project leader should also cooperate with the electrical engineers to ensure that the electrical drawings and specifications are complete and compatible with all of the other drawings and specifications.

List of Electrical Drawings

The following is a suggested list of electrical drawings required for a typical pumping station. The list would be modified according to the size and complexity of the job.

- Electrical legend
- Single-line diagram
- Control diagrams (schematics)
- Site plan (power and lighting)
- Building power plan (including control wiring)
- Building lighting plan
- Fire alarm, communications, and security plans as required
- Equipment elevations and schedules
- Conduit and wire schedules
- Details.

Symbols

In the electrical legend, always include a list of symbols used. See Chapter 2 for common electrical symbols.

Coordination with Other Pumping Station Drawings

The project engineer is responsible for coordinating the electrical design with the other disciplines (see Table 1-1)—a task vital for the continuity and completeness of the drawings. The project engineer should:

- Verify that electrical equipment, including light fixtures, does not conflict with structural elements or mechanical equipment, and that necessary space and mounting provisions exist.
- Check the plans and specifications of the other disciplines (particularly process and mechanical disciplines) for compatibility with electrical systems such as voltage, phase, alarms, and controls.
- Be sure that the electrical engineer checks every piece of equipment that requires electrical energy or control. Commonly missed items include instruments, limit switches, solenoid valves, and alarms.
- Check to see that conductor sizes, dimensions, etc., are shown only once. On other drawings, these items should be shown (if at all) as "reference-only" data to avoid conflicts.
- Perform a final complete check of electrical drawings against all other pumping station drawings and specifications just before the drawings are sent out.

Raceway System

For raceway systems,

- Show conduits only in schematic form except where noted specifically or detailed otherwise.
- Electricians often double-up home runs for lighting and receptacle circuits, control wire runs, and so on. If doubling-up is undesirable, so note or detail the home runs.
- A minimum conduit size of 19 mm ($^3/_4$ in.) is suggested.
- Detail and/or specify the mode of conduit entry into structures, including underground penetrations for the prevention of groundwater entry.
- Detail conduit duct banks carefully: include conduit layout, section, acceptable types of conduit, reinforcing bar sizes and extent, concrete encasement, and slope direction for drainage.
- Remember that PVC conduit deteriorates if exposed to sunlight or possible impact damage. Embedded PVC conduit may suffer severe damage during pouring of concrete in walls and particularly due to vibrators.
- The installation of spare (for example, not less than 10% of the total) conduits (capped at both ends) from panelboard and motor control center (MCC) or motor starter panel to exterior or to attic is suggested.
- In corrosive areas such as wet wells and chlorine storage rooms, use only PVC-coated, hot-dipped galvanized (inside and outside) rigid steel conduits for corrosion resistance. Require that all bare spots in the conduit system be recoated with PVC (or an equivalent).
- The trend is toward the use of aluminum conduit, but aluminum reacts quickly with the lime in damp concrete, so *it must not be encased in concrete*. For first-class construction, use only PVC-coated steel conduit in pumping stations.
- Allow contractors to size junction boxes and wireways in accordance with codes unless the designer requires oversize boxes for special use.
- Provide drainage for large handholes and manholes.

- Install pulling eyes on walls opposite every electrical conduit entry inside manholes and handholes to aid in pulling wires.
- Use cast, hot-dipped galvanized ferrous boxes for embedded systems—receptacle, switch, and junction boxes in all pumping station levels where embedment might be allowed or specified.
- NEC 370-3 restricts the use of nonmetallic boxes in metallic raceway systems. Such boxes cannot be used unless an internal ground jumper is provided between each metallic conduit. Ground jumpers are impractical with conduit bodies (condulets) and small boxes such as trade types FS and FD, so PVC-coated condulets and FS/FD boxes are recommended. For larger boxes, it is practical to field install an internal ground jumper, and nonmetallic boxes can be used. Nonmetallic boxes should have nonmetallic or stainless-steel hinges, mountings, and clasps.

Wiring

For wiring,

- Show feeders, the pump motor branch circuit, and the principal ancillary equipment branch circuit conductor sizes either on plans, on the single-line diagrams, or on conductor schedules, but show them only once.
- Indicate number and minimum size of control, lighting, and outlet wiring on drawings or specifications. By note, direct installer to adjust wire size for the type of wire, conduit fill (defined in Section 2-2), voltage drop, and need for 60°C (140°F) ampacities at 15- to 100-A circuit breakers and fused switches. An installer can size conduit per the NEC, but it is better for engineers either to size conduit or to limit the percentage of fill if any conduit is to be sized by others.
- Coordinate process and instrumentation wiring carefully with respect to: (1) number of control wires serving each instrument and device, control requirements, and 120-V power; (2) shielded cable for 4- to 20-mA dc signals and other instrumentation signals (require the shield to be grounded at one end only) (if the cable is double shielded, follow manufacturer's directions); (3) for intrinsically safe wiring, refer to NFPA Standard 493-1978 for complete details; (4) combine circuits carefully to avoid exceeding the maximum voltage buildup; (5) use separate steel conduits for these circuits; and (6) in panels, separate wiring a minimum of 50 mm (2 in.) from all other panel wiring.
- Equipment ground-wire considerations: (1) A ground wire from the system ground to each 480-V motor and panel is recommended; (2) bond all ground wires to both ends of rigid metal conduit; (3) ground all metal enclosures, water pipes, and machines as required by NEC 250-42, 43, 44(d), and 80; and (4) provide a ground wire in every PVC conduit.
- Use duct seal in conduits where they emerge from ground and enter enclosures to prevent passage of water into the enclosures and building. Slope underground ducts to drain water to manholes.
- Number all equipment and devices. Require numbered wire tags at each end of all conductors.

Equipment Space Heaters

All space heaters should be powered from a 120-V panelboard so that equipment that is idle or down for repairs has condensation protection. At motors, control panels, and so on, provide labeled heater disconnects for safety during maintenance. Provide thermostats in equipment for operating space heaters and connect motor space heaters through a normally closed auxiliary motor starter contact so that the heater is on when the motor is off and vice versa. For large, important motors, consider a neon light across the contacts to indicate heater continuity when the heater is off. Consider low-voltage motor winding heating from a separate transformer in the motor starter cubicle (see Chapter 13), but note that there will be no heating when motor branch circuit is off.

Power and Telephone Utility Requirements

Instruct the contractor to contact the appropriate power and telephone companies for all service requirements and to conform to them. The engineer, however, must contact the utilities in the early stages of design to obtain their agreements to serve the facility adequately (see Section 9-3).

9-2. Specifications

Some specific hints for consideration in writing the electrical specifications are given in this section (see also "Specification Language" in Section 28-1).

In the "General Requirements" portion of the electrical specifications, provide an article titled "Electrical Devices Furnished with Mechanical Equipment" and have each specifier of equipment include a special

callout of electrical contacts, alarm functions, and any auxiliary electrical power requirements. Reference these callouts to the "Electrical Devices" specification.

Information and sample specifications may be obtained from the Construction Specifications Institute [1]; their format is becoming popular and is increasingly being required by clients.

Coordination with Other Specification Sections

Coordination is very important to ensure the continuity and completeness of specifications. Wherever any equipment requires electric power, control, or instrumentation, check the following with the designer and be sure that this information is included in the equipment specifications:

- Voltage and phase
- Disconnects
- Enclosures
- Control philosophy, such as a local–remote selector switch with an On–Off pushbutton
- Alarm contacts
- Special features or circuits to ensure compatibility with other electrical equipment
- Any desired special component or wiring specifications.

In addition:

- Require that all conductors used for interwiring between panels and for input or output purposes be brought to numbered terminal strips and that all wiring (internal and external) be tagged at both ends with preprinted wire markers. Require that the wire marking code be submitted for acceptance prior to the manufacture of the equipment.
- Require that a copy of the final issue "As Delivered" control diagram be attached to the inside of the door of the equipment cabinet.
- Require the control diagrams to be drawn in the ladder-diagram format with all contacts and lines identified (numbered or coded) and all relays and control devices designated by appropriate names and numbers.
- If a PLC is to be provided, require that a ladder-diagram format control diagram be submitted for acceptance and require that a tape or disk copy of the final program accompany the equipment.

Seismic Requirements

All equipment must be attached or supported so that no hazard to personnel will result from any seismic force or motion. Contact the project structural engineer for help in fulfilling and enforcing this requirement. The seismic zone for every part of the country is designated in UBC Section 2332, Figure No. 1 (in 1996), and the contractor and suppliers must be instructed to follow all of the requirements. Zones vary from 1 through 4. The worst, Zone 4, corresponds to an earthquake with a magnitude of 8 on the Richter scale. Although equipment is usually not required to operate properly during a seismic event, specify that large equipment (such as a large circuit breaker) must not change position and that large engines and motors must not start or "bump" from momentary energization.

Equipment Labeling

Labeling, an indication of the testing and listing of the equipment by a recognized testing laboratory, is required by most local and state inspection authorities and must be required for all standard equipment. Custom equipment and control panels can be expensive if the authorities insist on a label. Whenever possible, design around equipment that is UL-listed as a complete unit (panel and all).

Shop Drawings

With the possible exception of standard, widely used materials, it is suggested that shop drawings be required for each item for which detailed specifications were written. Check the shop drawings thoroughly. Specifications should include (1) the equipment and information that must be submitted and (2) a submittal schedule.

9-3. Contacting Utilities

It is necessary to contact each utility early to obtain their requirements and the name of the person assigned as their representative. Back up all telephone conversations with confirmation letters, and visit the site with the utility representative.

Electric Power Utility Contact

An underground (instead of an overhead) service from the electric power utility is preferred because of reliability and appearance. Provide the service from the nearest pole or vault to the pumping station. In addition,

- Obtain all current rate schedules that might be applicable to the station. If an abbreviated rate schedule is given, ask for a complete rate book.
- Discuss the advantages and disadvantages of each schedule with the utility representative.
- Analyze demand charges, seasonal demand or energy charges, basic energy charges, power factor clauses, and the basis of the demand charge (15-min demand, 30-min demand, connected horsepower, etc.) and discuss them with the utility representative.
- Use an aerial photographic survey to mark the location of the pumping station and the preferred point of service. Review the proposed location in the field with the utility representative and send a marked copy to the utility.
- Obtain the utility's agreement on acceptable starting methods and the frequency of starts for the proposed pumping station drives. If they do not allow across-the-line starting for pump drives of the proposed size, discuss the alternatives, select the best one for reliability and economic factors, and inform the utility of the selection.
- Discuss transformer ownership and the utility's maintenance policy if the transformer is provided by the owner.
- Obtain the utility's transformer specifications and find the size they would likely provide, its impedance, and its primary fuse size and insulation (dry or oil).
- Check the utility's policy on overhead and underground service installation and determine whether they share costs or furnish the entire service. If they furnish it, determine whether underground services are encased in concrete. Cable for pumping stations should be in a concrete-encased conduit and not directly buried.

Telephone Company Contact

Obtain agreement with the telephone company on the following:

- Approximate required date of service
- Service point (e.g., pole)
- Routing of telephone company cables and the location of their equipment in the owner's facilities
- Clear definition of items (and their interface) furnished by telephone company and by owner (contractor)
- Telephone terminal facilities: type (plywood panel, enclosure, etc.) and size, 120-V power requirements, and grounding provisions
- Telephone extension conduit size and type
- Telephone extension outlet box size, type, location, and mounting height
- Pumping station equipment interface requirements, for example, for telemetry (space, proximity to 120-V sources, need for terminal block or surge suppressors).

9-4. Construction Information to Utilities

The following information should be sent to the electric power utility following the completion of the design.

- *Specifications:* special switchboard, substation, and special grounding specifications;
- *Drawings:* site plan, building power plan, single-line diagram, service gear elevation (switchboard, MCC, etc.), and substation plans and elevations; and
- *Other data:* the bid opening date and the approximate date service is needed.

The telephone company should be sent

- *Specifications:* terminal cabinet specification, including size;
- *Drawings:* site plan and building plan containing telephone provisions; and
- *Other data:* the bid opening date and the approximate date service is needed.

9-5. Load Estimation

To facilitate the design of the electrical power service entrance equipment, substation, motor control center, and so on, a load or motor list such as that shown in Example 9-1 should be utilized. The motor and equipment lists should be started at the first stage of electrical design. If no data sheets are available from the designers at the first meeting between the various design team members, a tentative list should be prepared by the electrical engineer and copies should be distributed for general discussion. The motor and equipment list must be kept up to date during the course of design as motors and loads change in rating or are added or deleted. One of the most subtly important points of the motor and equipment list is an early decision on a name for each equipment item exactly as the designer wants it inscribed on the nameplates. Even though the exact name of a particular item may change during the course of design, all parties involved will use the same name to describe a piece of equipment.

Following the motor and equipment list, the electrical calculations for the pumping station include the following:

- Sizing the branch circuit breaker or fused switch, starter, and conductors for each of the main pumps
- Sizing the circuit breaker, starter, and conductors for the sump pumps and other incidental motor loads
- Sizing the lighting and small power transformers, circuit breakers, and conductors
- Sizing the heating and ventilating equipment panel circuit breakers and conductors
- Sizing the main (service) overcurrent protection device, transformer, primary and secondary protection, and conductors
- Lighting calculations, layouts, and circuiting
- Sizing the power factor correction capacitor bank
- Sizing the standby engine-generator.

These calculations are illustrated in Examples 9-2 through 9-10.

Example 9-1
Electrical Load Estimation

Problem: Estimate the service electrical load for a pumping station with four constant-speed, 25 hp, 720 rev/min motors driving three duty pumps and a standby pump. The maximum planned capacity is 4500 gal/min at a maximum TDH of 40.7 ft for a Hazen–Williams C of 120 and 38.3 ft for a C of 145. The present required capacity of 3600 gal/min can be achieved with two pumps.

The floor elevations are: ground, 112.5 ft; basement, 102.5 ft; pump room, 90.0 ft; and the three floors are 21 by 42 ft in plan. The ground floor contains a trolley hoist, the motor control center, and an engine-generator. Air handling units are in the basement. The wet well floor is at elevation 89.25 ft, and the wet well plan size is 8 by 34 ft.

Solution:

Pump motors. According to NEC Table 430-150, the full-load current (I_{FL}) for a 25-hp, Code F, 720 rev/min, 460-V, three-phase, 60-Hz motor is 34 A. [Note: check the manufacturer's data for 720 rev/min motor to determine the actual full load current. Slow-speed motors (<1200 rev/min) typically have lower power factors and therefore draw more current than is shown in NEC tables, so compute with the manufacturer's data if they are available. If power factor correction at the motor terminals is included in the design, the full-load current may be reduced to the NEC value (or even less).]

Heating and ventilating. The power requirements for heating and ventilating (H&V) must come from the H&V engineer, who determines them on the basis of the size of rooms, insulation value of the building envelope, heat sources, and climate (as explained and calculated in Chapter 23). The H&V loads used in this example are higher than normal and would be appropriate for a pumping station in a severe climate. The H&V load is typicaly less than 10% of the total load.

Other electrical loads. Estimates for other loads can be made either by the project leader or by an experienced electrical engineer. As the design progresses and the loads become known more exactly, the estimates below may be modified.

Item		Power					Service			
				Full load						
Name	No.	hp	kW	A	V	Phase	Continuous	Intermittent	Design A	Essential load on generator, A
Pump	1	25		34	460	3	X		34	34
	2	25		34	460	3	X		34	34
	3	25		34	460	3	X	X	34	34
	4	25		34	460	3		X	Stdby	Stdby
Sump pump	5	0.5		1	460	3		X	1	1
	6	0.5		1	460	3		X	Stdby	
Instrument air compressor	7	0.5		1	460	3		X	1	1

Item		Power					Service			
				Full load						
Name	No.	hp	kW	A	V	Phase	Continuous	Intermittent	Design A	Essential load on generator, A
Pump room exhaust fan	8	0.5		1	460	3			1	1
Control room exhaust fan	9	0.5		1	460	3			1	1
Control room exhaust fan	10	0.5		1	460	3			1	1
Engine room exhaust fan	11	0.5		1	460	3			1	1
Air handling unit	12	0.5		1	460	3			1	1
Duct heater	13		16.5	21	460	3			21	
Pump room H&V control	14		0.5	1	460	1			1	1
Wet well										
intake fan	15	0.5		1	460	3			1	1
exhaust fan	16	0.75		1.5	460	3			1.5	1.5
duct heater	17		34.8	43.5	460	3			43.5	
Wet well H&V control	18		0.5	1	460	1			1	1
Lighting transformer	19	(9 kVA)								
Totals		8.5 62.75		10.5 222.5	460	3			10.5 187.5	9 121.5

Maximum power required (for a corrected power factor of 95%) is

$$P_{3\phi} = \sqrt{3} \times 460 \times 187.5 \times 0.95/1000 = 14.9\,\mathrm{kW}\ (149.4\,\mathrm{kVA})$$

Demand factor: actual = 0.85. Demand factor: used = 1.0. (The demand factor is an arbitrary percentage of connected, rated design loads that is used to determine actual operating loads. It allows for nonconcurrent loads and motors that are not running at full load.)

Minimum feeder to MCC = 187.5 + 25% of largest motor = 187.5 + (0.25 × 34) = 196.0 A.

Minimum main circuit breaker = 187.5 A + 50% of largest motor = 187.5 + (0.5 × 34) = 204.5 A.

Minimum standard circuit breaker = 225 A.

Design load plus 20% "cushion" = 187.5 × 1.2 = 225 A.

Anticipated power needed (for a corrected power factor of 95%) is

$$P_{3\phi} = \sqrt{3} \times 460 \times 225 \times 0.95/1000 = 170.2\,\mathrm{kW}(179\,\mathrm{kVA})$$

Minimum feeder to MCC = 225 + (0.25 × 34) = 233.5 A. Minimum feeder conductor (NEC Table 310–16, 75°C column) = 250 kcmil (thousands of circular mils) rated for 255 A.

Minimum main circuit breaker = 225 + (0.5 × 34) = 242.0 A. Next smallest standard circuit breaker rated for 300 A. Minimum feeder conductor = 350 kcmil rated for 310 A.

Minimum standard transformer = 225 kVA.

Use: 225 kVA, 12.47 to 480/277-V transformer;
300-A three-phase main circuit breaker in MCC;
350-kcmil feeder conductors (rated for the main circuit breaker); and
#2 AWG (American Wire Gauge) ground conductor.

9-6. Overcurrent Protection and Conductor Sizing

The NEC has definite standards for selecting overcurrent protective device ratings and for sizing conductors. Conductor sizing is based on the load current of the equipment connected to the circuit and is influenced by circuit length (voltage drop) and ambient temperatures. It is usually wise to allow spare capacity. The NEC and local and state codes and rules must be followed as a *minimum* safety standard for design. Furthermore, consider designing for long-time service and load growth. Overcurrent protection is accomplished by applying current-sensing techniques at many points in the utility's distribution system as well as within the user's system. Sensing equipment may be self-destructive (e.g., fuses), resettable (e.g., circuit breakers), or remotely located (e.g., protective relay applications).

The electrical design engineer's role is to develop a system so coordinated that an overloaded motor will be removed from service before the motor insulation or bearings are damaged, and only the affected equipment (in this example, the overloaded motor) will be removed from service. A properly coordinated overcurrent system will enable fault conditions to be isolated from the system without affecting the operation of other equipment. Coordination studies range from simple to very complex. A thorough discussion of the method applicable to system studies and coordination is contained in IEEE Standard 242.

A single-line diagram of the pumping station of Example 9-1 is shown in Figure 9-1. The utility fuses are pole mounted, and they protect (1) the utility from overloads (overcurrent, faults) within the pumping station electrical systems, (2) the transformer, and (3) the service cable connecting the transformer to the utility.

Fuses

Fuses are commonly used by the utility to protect their system from faults in the customer's electrical system.

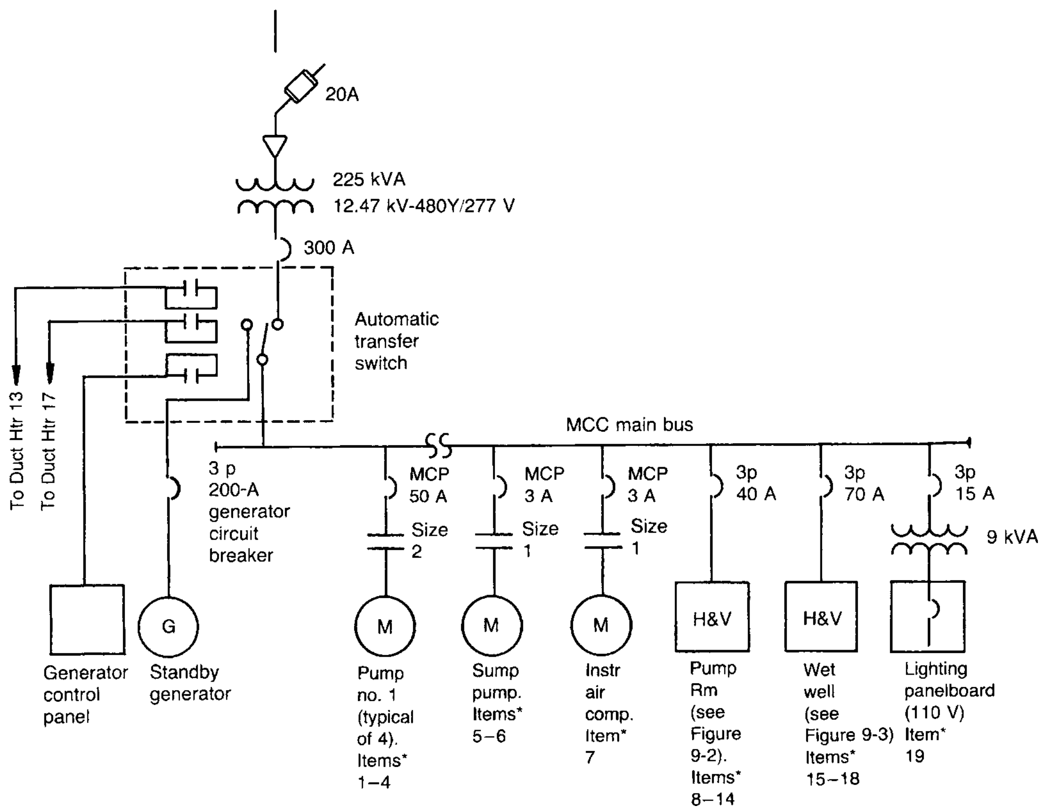


Figure 9-1. Pumping station electrical single-line diagram. Htr, heater; Inst, instruments; Comp, compressor; H&V, heating and ventilating; AHU, air handling unit; Cont Rm EF, control room exhaust fan; Int, intake. *See load estimates in Example 9-1.

Several classes of fuses are encountered in industrial design such as that for pumping stations. An example is pole-mounted fuses used by utilities. These may be of the expulsion type in which the gases generated by the melting fuse and the heated casing during an overcurrent condition cause a mechanical indicating device to function. The typical dropout fuseholder unlatches and the blown fuse remains with the hinged portion of the holder.

The fuses are generally selected by the utility, and so it is necessary to obtain the exact fuse data and available fault currents from the utility before starting a coordination check.

Other fuses commonly used in pumping stations include medium-voltage transformer primary fuses on large installations, medium-voltage current-limiting fuses on medium-voltage (2400 and 4160 V) motor starters, low-voltage current-limiting fuses in low-voltage switchgear (to reduce the fault current available to the loads), and low-voltage transformer primary fuses.

In the pumping station design examples in this chapter, fuses are used only to protect control transformers, and they are used in both the transformer primary and secondary sides. The electric power utility is to furnish the primary fuse at their service pole.

Circuit Breakers

Circuit breakers are electromechanical switching units that are designed to open a circuit when specified overload or fault conditions occur. Either thermal-magnetic or magnetic-only molded-case circuit breakers are normally used in pumping stations. The power circuit breaker type may be preferred for large or high-voltage motors. The type of circuit breaker chosen is based on current withstand, interrupting, and coordination ratings and on economic factors.

The thermal-magnetic circuit breaker is tripped by a magnetically operated element for severe faults and by a thermal element for long-duration overloads. Thermal-magnetic circuit breakers may be adjustable and nonadjustable and are used as main and branch circuit breakers and are used to protect almost all types of loads.

An adjustable-trip form of the magnetic-only circuit breaker is most often used in conjunction with a motor starter to protect motor branch circuits. This unit, called a "motor circuit protector (MCP)," is sized and adjusted to pass the starting current of a particular motor and will not trip when the sustained current is less than the setting (which can be as much as 13 times the full-load current of the motor). It will trip, however, under severe fault conditions. The overload relays required on the motor starter provide the required motor overload protection.

Example 9-2
Branch Circuit Calculations for Pump Motors

Problem: Size the branch circuit breaker, starter, and the branch circuit conductors for each of the main pumps of Example 9-1.

Solution: From Example 9-1, the motor full-load current, I_{FL}, is 34 A.

Size the branch circuit breaker. NEC 430-152 allows the circuit breaker to be sized to a maximum of 250% of full-load current, which is $34 \times 2.5 = 85$ A. Use a 70-A circuit breaker for a normal inertia system. If the inrush current (approximately $6 \times I_{FL} = 204$) does not persist more than about 10 s, the circuit breaker will not trip. Be careful to check the time–current curves of the circuit breaker.

If the drive inertia of the centrifugal pumps were higher than normal, the circuit breaker should be sized above the 250% value in accordance with NEC 430-52 (which limits the maximum setting to 400%) to accommodate the necessary longer acceleration time of the drive.

Size the motor circuit protector. For adjustable-trip, magnetic-only circuit breakers, NEC 430-152 allows the MCP to be set at 700% of full-load current, so $34 \times 7 = 238$ A. NEC 430-52 limits the maximum setting to 1300% of the full-load current, or $34 \times 13 = 442$ A. Select an MCP with a 50-A rating and an adjustable-trip range of 150 to 580 A. The initial setting should be approximately 200 A (see, for example, literature from the Square D Company; the ratings of other manufacturers are similar).

Size the disconnect switch at motor. If a disconnect switch is required at the motor, NEC 430-110 applies, and this requires the switch to be rated at least at 115% of the motor's full load current, or $34 \times 1.15 = 39$ A.

The next larger standard-size disconnect switch is rated at 60 A, so use a 60-A disconnect switch (in a housing suitable for the environment) with an auxiliary control contact that opens when the switch is in the open position. The contact should open before the main contacts open in order to drop out the motor starter.

Size the motor starter. The motor starter manufacturer's data show that an NEMA size 2 starter is rated for 25-hp, 460-V, three-phase motor-starting duty, so use a NEMA size 2 starter. A size 3 starter would be specified if it is likely that a 30-hp motor will be substituted in the near future or the motor is of a type that draws more current than is shown in NEC tables. Low-speed motors often require larger starters; check with the starter manufacturer.

Size the branch circuit conductors. Branch circuit conductors must be rated at least at 125% of the motor's full-load current, or $34 \times 1.25 = 42.5$ A. Using a maximum conductor temperature of 60°C, NEC Table 310-16 lists No. 8 AWG copper at 40 A and No. 6 AWG copper at 55 A.

Use three No. 6 AWG TW or THW insulated copper conductors plus No. 6 or No. 8 AWG ground conductor (NEC Table 250-95 lists minimum-size grounding conductors). THW insulation is a thermal plastic material that softens when heated. Many designers prefer XHHW insulation, a cross-linked synthetic polymer, which is thermal-setting and hardens when heated.

Checking Branch Circuit Voltage Drop

NEC 210-19, footnote a, recommends that the branch circuit voltage drop be less than 3%, and the total drop in feeders and branch circuits must not exceed 5%. Good design limits the voltage drop to less than 2% each for feeders and branch circuits.

Another facet of voltage drop (which is not governed specifically by the NEC but is essential to good operation) is the motor-starting voltage drop in the branch circuits and feeders. An induction or synchronous motor draws about six times the full-load current at the moment of starting, and the draw decreases to the current necessary to drive the load during the remainder of the acceleration time and the normal running time. Ordinarily, a 13 to 15% voltage drop is acceptable and provides sufficient voltage for accelerating the drive. The starting voltage drop with No. 6 AWG copper conductors is approximately 1.4%, and the running drop is about 0.3%. The starting voltage drop is a problem only on long runs. The method for calculating the voltage drop is given in Example 13-1.

Miscellaneous Drives and Loads

Two important drives, the pump room sump pump and the instrument air compressor, are rated at $\frac{1}{2}$ hp each. These drives should be served by branch circuits of the motor control center at 460 V (three phase).

The exhaust fans and the duct heaters could also be served by individual motor circuits. Because of the interlocking requirements and to facilitate safe maintenance practices, two heating and ventilating panels should be provided. The arrangement of the panels is shown in single-line diagram form in Figures 9-2 and 9-3.

Each heating and ventilating control panel includes all of the power and control functions for the duct heater and its interlocked fans. The manufacturer of the panel should be given the option of providing a transformer large enough to serve the fans at 120 V or 208 V (single phase) from a 208Y/120-V panelboard in the H&V control panel.

Interlocks from the station's automatic transfer switch cut off the duct heater when power is being taken from

Example 9-3
Branch Circuit Calculations for Sump Pump

Problem: Size the circuit breaker, the starter, and the branch circuit conductors for the sump pump (and all other loads of identical horsepower rating) in Example 9-1.

Solution: One manufacturer's 1150-rev/min, 460-V, $\frac{1}{2}$-hp motor is listed with a full-load current of 1 A and a locked-rotor current of 12.5 A.

Size the motor circuit protector. Size the MCP at 700% of full-load current per NEC 430-152, or $1.0 \times 7.0 = 7$ A.

Select an MCP rated at 3 A with an adjustable-trip range of 8 to 22 A—slightly above the 7 A required. Depending on the exact characteristics of the motor, the setting should be very close to (or slightly exceed) NEC limits.

Size the starter. Although the manufacturer's data show a size 0 starter, the starter should be a NEMA size 1, which is the minimum recommended by the authors.

Size the branch circuit conductors. These must be rated at 125% of the full-load current, or $1.0 \times 1.25 = 1.25$ A. The use of minimum-size wire, No. 12 AWG stranded copper conductor, is recommended; No. 14 AWG conductor is adequate.

---

the station standby generator. Fans and louvers are still required to operate under these conditions.

The pumping station lighting and small ancillary loads operate at 120 V (single phase) or at 208 V (single and/or three phase). A step-down transformer and circuit breaker panelboard are used to feed this load. The calculations for feeding such equipment are illustrated in Example 9-4.

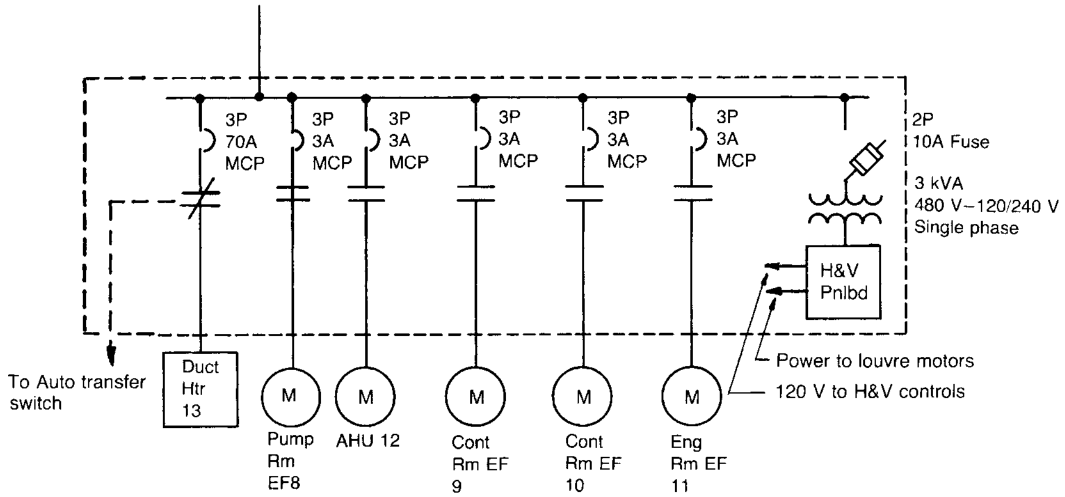


Figure 9-2. Pump room heating and ventilating panel.

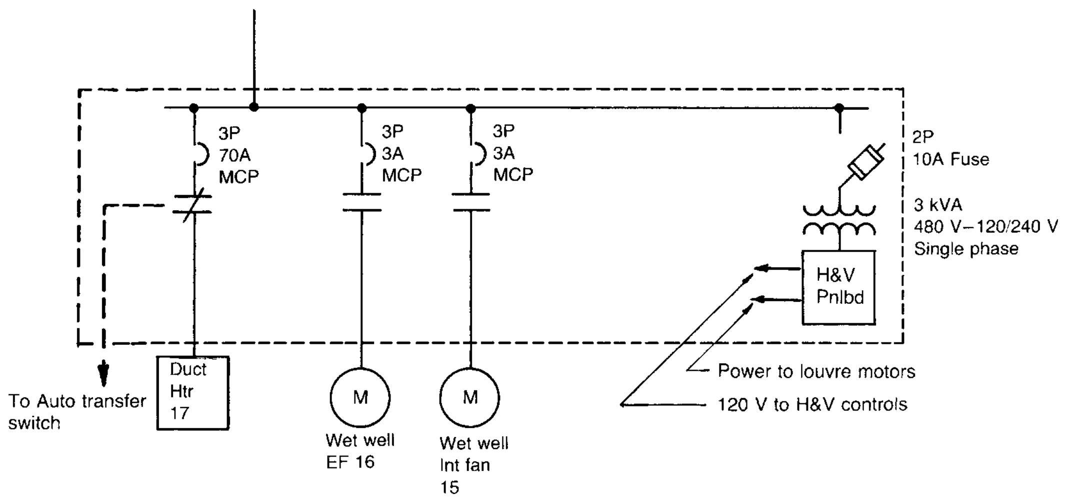


Figure 9-3. Wet well heating and ventilating panel.

Example 9-4
Lighting and Small Power Transformers

Problem: Calculate the size of the lighting and small power transformers in Example 9-1.
Solution: The principal loads on the lighting panelboard are the pumping station's lighting and receptacle circuits and the battery charger. Allow 1000 W for the battery charger. Allow 3000 VA for lighting in the pump building and 500 VA in the wet well. Allow 1500 W for receptacles. The calculated load is 600 VA or 6 kVA. The next larger size transformer is 9 kVA.

A 9-kVA, three-phase, 480-V delta primary, 208Y/120-V secondary transformer should be used.

Size the transformer primary circuit breaker. NEC 450-3b governs the sizing of transformer overcurrent protection. Where circuit breaker protection is provided in both the primary and secondary, the maximum primary overcurrent protection is 250% of full-load current. The maximum secondary overcurrent protection is 125% of the full-load current. From Equation 8-6 ($P_{3\phi} = \sqrt{3}V_L \times I_L \times Pf$),

$$I_{FL} = P_{3\phi}/(V_L \times \sqrt{3}Pf) = 9000/(480\sqrt{3} \times 0.95) = 11.4 \text{ A}$$

if a power factor of 0.95 for the lighting and small loads on this panelboard is assumed (usual for fluorescent lighting).

$$250\% \text{ of } I_{FL} = 11.4 \times 2.5 = 28\text{A}$$

This maximum must not be exceeded, so use the next lower size, which is a 20-A primary circuit breaker.

Size the secondary circuit breaker (panelboard main breaker). The full-load current is

$$I_{FL} = 9000/(208 \times \sqrt{3}) = 25\text{A}$$
$$125\% \text{ of } I_{FL} = 25 \times 1.25 = 31\text{A}$$

Use a 30-A secondary circuit breaker. Secondary protection should always be provided.

Size the primary conductors. The primary is protected by a 20-A circuit breaker (refer to NEC Table 310-16). Because the 60°C rating of a copper conductor at 20 A requires No. 12 AWG size, use three No. 12 AWG copper conductors plus one No. 12 AWG ground conductor.

Size the secondary conductors. The secondary is protected by a 30-A circuit breaker. Because the 60°C copper temperature rating at 30 A requires a No. 10 AWG conductor, use four No. 10 AWG copper conductors (three phase and one neutral conductor), plus one No. 10 AWG ground conductor.

Size the panelboard circuit breakers. The panelboard circuits should all be rated at 20 A, and all of the conductors should be No. 12 AWG copper.

The H&V panels require 460-V, three-phase power. The sizing of the feeder circuit breaker and feeder conductors is illustrated in Example 9-5.

Example 9-5
Heating and Ventilating Electrical Equipment

Problem: Determine the panel loads, the size of the circuit breakers, and the size of the conductors for (1) the pump building heating and ventilating and (2) the wet well heating and ventilating in Example 9-1.

Solution: (1) *Pump building H&V panel*. Per NEC code, for the largest motor add 25% of 1 A (= 0.25A) for minimum conductor size. The duct heater is rated at 16.5 kW, so the rated current is

$16{,}500/(460 \times \sqrt{3}) =$	21 A
Four $\frac{1}{2}$-hp motors at 1 A each =	4 A
Transformer, 1 kVA, single phase =	2 A
Largest motor × 25% =	0.25 A
Total H&V panel load =	27.25 A

Choose a circuit breaker for at least 125% of total calculated load:

$$27.25 \times 1.25 = 34\,\text{A}$$

Use a 40-A circuit breaker in the MCC.

Conductors must have an ampacity (current-carrying capacity) of no less than 40 A. In the 75°C column in NEC Table 310–16, No. 8 AWG conductors are rated at 40 A and No. 6 AWG conductors are rated at 55 A. Use three No. 8 AWG copper conductors and one No. 8 AWG ground conductor.

(2) *Wet well H&V panel*. The wet well H&V panel is calculated in a similar manner. A circuit rated at 60 A minimum is required. Use a 70-A circuit breaker in the MCC, and use three No. 4 AWG copper conductors and a No. 8 AWG ground conductor.

Service Load

Now that the branch circuits for motors and for heating and ventilating are sized, the total station (service) load can be determined, and the main transformer, its protection, and the conductors can be sized.

Example 9-6
Service Circuit Transformer, Protection, and Conductors

Problem: Find the sizes of the service circuit breakers, the main transformer, the primary and secondary protection for the transformer, and the service conductors for the pumping station in Examples 9-1 to 9-5.
Solution: Summarize all the pump station loads:

Pumps: $3 \times 34\,\text{A} =$	102 A
Air compressor and sump pump:	2 A
Lighting load:	11 A
Pump building H&V:	27 A
Wet well H&V:	48 A
Total station load:	190 A

Treat all loads as “continuous duty.” Per NEC 430-62(b), the rating of the main circuit breaker may be sized for possible future larger loads.

Size of main transformer. The transformer’s kilovolt-ampere rating should exceed the calculated load of the station by 15 to 20% or more depending on several factors. The lack of a quickly obtainable replacement transformer may dictate operation at a somewhat lower total temperature than the full-load rating would produce. Predictions of substantial load growth within 10 yr make oversizing worth studying. A slight oversizing, such as 15%, allows for some

adjustment in main pump sizes or in H&V or other auxiliary equipment without the necessity of rebuilding the entire station electrical system. For a station load of 190 A, $190 \times 480 \times \sqrt{3} = 158$ kVA and 15% oversizing adds 23.7 kVA for a total of 181.7 kVA, so use the next larger standard size unit, which has a rating of 225 kVA.

Size of transformer primary and secondary protection. Assume that the fuse will be provided by the utility. Although the utility is not required to design to NEC standards, the fuse size is likely be the same because it is based on load size, and utilities use conservative conductor current ratings for their equipment.

For transformers with less than 6% impedance and a primary voltage greater than 600, NEC allows a maximum fuse rating of 300% of the full-load current. The 480-V secondary circuit protection cannot be more than 250% of the full-load current. For a utility voltage of 12,500, the primary is $225{,}000/(12{,}500\sqrt{3}) = 10.39$ A, so the maximum fuse per the NEC is $10.4 \times 3 = 31$ A.

A 30-A fuse could be used, but a 20-A fuse designated 20T type provides transformer protection closer to the overloading point of the motor.

The secondary is $225{,}000/(480\sqrt{3}) = 270.6$ A. The maximum circuit breaker (main MCC circuit breaker) is $271 \times 2.5 = 677$ A; the minimum circuit breaker (the total connected load plus 50% of largest motor) is 207 A.

Use a 400-A frame circuit breaker with a 300-A trip.

Sizing the service conductors. The service conductors should be sized for any foreseeable load growth that can be accommodated without the necessity of replacement. The minimum size is that which will be protected by the main circuit breaker, namely, 300 A at 75°C (refer to NEC Table 310-16).

Use three 350-kcmil copper phase conductors plus one No. 2 neutral (grounded) conductor in a concrete-encased duct 4 in. in diameter. The neutral conductor must be sized according to NEC Table 250-94 as a minimum.

9-7. Lighting

Use the IES [2] lumen method for lighting calculations. Manufacturers' selection charts are too inaccurate for small buildings because they are based on somewhat different criteria. A lighting level of 20 foot-candles (ft · cd) is usually sufficient for pumping stations if there are adequate GFCI receptacles for work lights needed for repairs. The following lighting levels are recommended:

- Offices: 50–100 ft · cd
- Control areas: 30–75 ft · cd
- Equipment areas 15–50 ft · cd
- Outdoor areas: 1–5 ft · cd.

Many states now have a lighting limitation based on watts per square foot, and this limitation should be investigated prior to design.

Obtain fixture data from manufacturers' catalogs. These data are classified by several parameters based on Equations 9-1 and 9-2. The room cavity ratio, *RCR*, is defined as

$$RCR = 5H(L + W)/A \tag{9-1}$$

where H is the distance from the work plane (usually 2 ft 6 in. above the floor in offices and control areas, 3 ft above the floor in equipment areas, and the floor itself elsewhere) to the bottom of the light fixtures, in feet, L is room length in feet, W is room width in feet, and A is floor area in square feet.

From the IES handbook [2], the relationship between illumination and the number of fixtures is

$$\text{Number of fixtures} = \frac{\text{ft} \cdot \text{cd}}{(\text{number of lamps/fixture})\ (\text{lumens/lamp})} \times \frac{\text{area}}{(CU)\ (LLD)\ (LDD)\ (RSD)} \tag{9-2}$$

where area is in square feet, CU is a coefficient of utilization depreciation and varies from 0.01 to 0.99 (Table 9-1), LLD is lamp lumen depreciation, LDD is luminaire dirt depreciation (dimensionless), and RSD is room surface depreciation (dimensionless).

Table 9-1. Coefficient of Utilization (*CU*),[a] Manufacturer's Data for a Selected Two-Lamp, 40-W Industrial Fluorescent Fixture[b,c] (spacing/mounting height = 1.5)

Room cavity ratio	R_w (%) with $R_{cc} = 80\%$			R_w (%) with $R_{cc} = 50\%$			R_w (%) with $R_{cc} = 10\%$		
	50%	30%	10%	50%	30%	10%	50%	30%	10%
1	0.68	0.65	0.62	0.62	0.60	0.58	0.55	0.54	0.52
2	0.59	0.54	0.50	0.54	0.50	0.47	0.48	0.46	0.43
3	0.52	0.46	0.41	0.48	0.44	0.39	0.43	0.39	0.37
4	0.46	0.40	0.35	0.43	0.37	0.33	0.38	0.34	0.31
5	0.40	0.34	0.30	0.37	0.32	0.28	0.34	0.30	0.27
6	0.36	0.30	0.25	0.33	0.28	0.25	0.30	0.26	0.23
7	0.33	0.27	0.22	0.30	0.25	0.22	0.27	0.23	0.20
8	0.30	0.23	0.19	0.27	0.22	0.19	0.25	0.21	0.18
9	0.27	0.20	0.17	0.25	0.20	0.17	0.22	0.18	0.15
10	0.24	0.18	0.15	0.23	0.18	0.15	0.20	0.16	0.13

[a] *CU* is for a 20% floor cavity reflectance (R_{fc}).

[b] R_{cc} = effective ceiling cavity reflectance; R_w = wall reflectance.

[c] Coefficient of utilization varies depending on type of fixture, type of lamp, and manufacturer. Refer to manufacturer's data.

Example 9-7
Lighting a Small Pumping Station

Problem: Assume a pumping station control room is 21 × 42 ft with a 15-ft ceiling. Mount lamps 12 ft above the floor. Assume the work plane is 3 ft above the floor. Design lighting to provide 40 ft · cd of illumination on the work area.
Solution: Select 4-ft-long, two-lamp industrial fluorescent fixtures with 32 WT-8 lamps and an electronic ballast. The precalculation procedure outlined in the IES handbook [2] provides a detailed explanation of these parameters:

- *LLD* (lamp lumen depreciation): Use 0.90 for fluorescent lights.
- *LDD* (luminaire dirt depreciation): Use IES Category III and assume fixtures are cleaned annually: therefore use "medium" (0.87).
- *RCR* (room cavity ratio) from Equation 9-1:

$$RCR = H(L + W)/A = \frac{5 \times (12 - 3)\ (21 + 42)}{21 \times 42} = 3.2$$

- *RSD* (room surface depreciation): use 0.95.

Find the coefficient of utilization (*CU*) from the IES handbook [2], or use the manufacturer's data as shown in Table 9-1. For two-lamp industrial fluorescent fixtures, 2850 lumens per lamp, and typical room reflectance values of 50/50/20, $CU = 0.46$ by interpolation. From Equation 9-2:

$$\text{Number of fixtures} = \frac{(40)(21 \times 42)}{2 \times 2850 \times 0.46 \times 0.90 \times 0.87 \times 0.95} = 18.0$$

Use 18 fixtures (2 rows of 9 fixtures). Fixture spacing must not exceed the fixture's designated spacing/mounting height ratio multiplied by the mounting height above the work plane in either direction. The illumination on the work area is found from Equation 9-2:

$$\text{Actual foot-candles} = \frac{18 \times 2 \times 2850 \times 0.46 \times 0.90 \times 0.87 \times 0.95}{21 \times 42} = 39.8$$

Locating fixtures is best done by drawing the room to scale, making paper cutouts of the fixtures to scale, and moving the fixtures around until the best effect is obtained. Assume the lamps line up with the long dimension of the room. Then:

$$\text{Transverse spacing} = (2\tfrac{1}{2})/(12 - 3) = 1.2 < 1.5 \text{ OK}$$

$$\text{Longitudinal spacing} = [42 - (9 - 1)4]/(12 - 3) = 1.1 < 1.5 \text{ OK}$$

Similar calculations can be made for the other rooms in the pumping station. More than one light fixture and lamp combination may often appear appropriate for an area or space. The final selection may require: (1) calculations for each light fixture/lamp combination proposed, (2) trial layouts with consideration for room geometry and contents, and (3) consideration, perhaps, of state lighting limitations. For even lighting at the work plane, conform to the fixture manufacturer's designated spacing/mounting height limitations in locating fixtures. Ensure that fixtures are accessible for cleaning and lamp changing.

9-8. Power Factor

Capacitors are used to correct the power factor, as computed by Equation 9-3.

$$Pf = \frac{\text{actual power (kW)}}{\text{apparent power (kVA)}} = \frac{\text{active current}}{\text{total current}} = \cos\theta \tag{9-3}$$

The relationship between power, current, and power factor in Example 9-8 is shown in Figure 9-4.

Example 9-8
Power Factor Relationship

Problem: Calculate the power, current, and power factor relationship for Pump No. 1 (see Example 9-1). The motor characteristics given by the manufacturer are 25 hp, 460 V, three phase, 34 A, and 0.75 *Pf*.
Solution: Use Equations 8-4 and 8–6 to calculate current and power values.

$$\text{Active (true) power} = P_{3\phi} = \sqrt{3}V_L I_L Pf = \sqrt{3} \times 460 \times 34 \times 0.75 = 20{,}300\,\text{W} = 20.3\,\text{kW}$$

$$\text{Apparent power} = \sqrt{3}V_L I_L = \sqrt{3} \times 460 \times 34 = 27{,}100\,\text{VA} = 27.1\,\text{kVA}$$

$$\text{Reactive power (see Figure 9-4a)} = \sqrt{27.1^2 - 20.3^2} = 17.9\,\text{kVAR}$$

Correction of Power Factor

Power company penalties for low power factor can be severe. It is often economical to raise the power factor to meet the power company's minimum limit. It is neither economical nor necessary to correct the power factor to 1.0, but raising the power factor to about 0.95 is likely to be economical.

Example 9-9
Capacitor Sizing

Problem: Calculate the capacitor sizes needed to raise the power factor of the pumping station from 0.75 to 0.95.
Solution: From Example 9-1, the design amperage is 187.5, so the normal operating station load is $\sqrt{3}(460)(187.5\,\text{A}/1000 = 149.4\,\text{kVA}$. At the present power factor of 0.75, the kilowatts (kW) used are $149.4 \times 0.75 = 112.0$ and the reactive kilovolt-amperes (kVAR) are $\sqrt{(149.4)^2 - (112.0)^2} = 98.9$. The vector diagram of power is shown in Figure 9-4b.

To increase the system power factor to 0.95, draw the vector diagram of Figure 9-4c. The kVA must remain constant. Therefore, $\text{kW} = (\cos\theta)(\text{kVA}) = (0.95)(149.4) = 141.9$ and

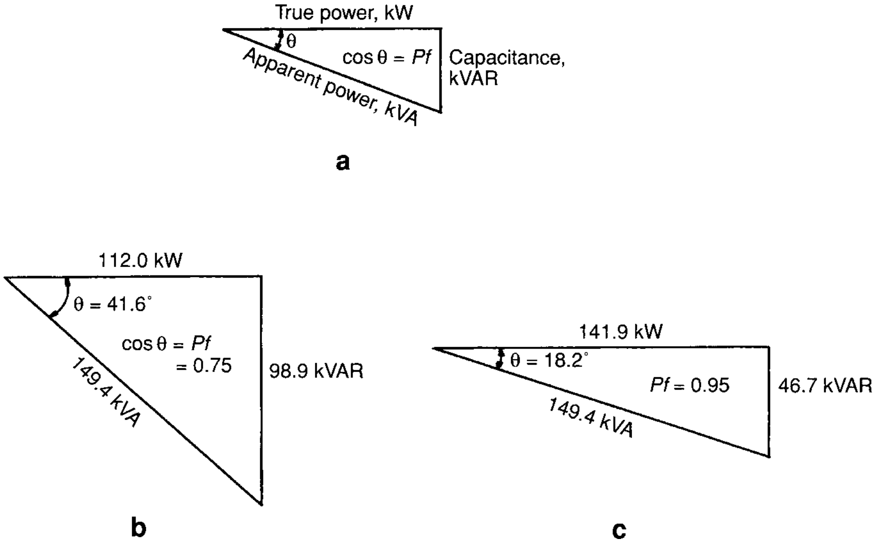


Figure 9-4. Vector diagrams for power factor correction. (a) vector diagram symbols; (b) power vector diagram without power factor correction; (c) power factor diagram with power factor correction.

$\text{kVAR} = \sqrt{(149.4)^2 - (141.9)^2} = 46.7$. The rating of the capacitor must change the capacitance from 98.9 to 46.7 kVA, so the capacitor rating must be $98.9\,\text{kVAR} - 46.7\,\text{kVAR} = 52.2\,\text{kVAR}$.

The 52.2 kVA is based on a station load of 187.5 A or 149.4 kVA. Because it is not economical to increase the power factor to a value greater than unity, less kVAR capacitor correction should be connected to the system at reduced station load. Therefore, it is desirable to distribute the capacitors to the system loads so that, as system load is turned on, additional capacitance is also added. The pump motors are typically the largest loads in a pumping station, so it is often advantageous to connect capacitors directly to each motor so as to switch them on with the motor. Calculations should be done so that the amount of capacitance added to each motor does not correct the motor *no-load* power factor to more than unity.

Install a total of 55 kVAR capacitance distributed as follows:

Each motor: 15 kVAR,

Bus capacitor: 10 kVAR (but sized with care to make up the deficiency after motor capacitors are finally chosen).

Capacitors

Obtaining satisfactory power factors by using capacitors is preferable to using more expensive synchronous motors unless synchronous motors are otherwise required. Switch capacitors with motors; that is, connect the capacitors to the motor terminals so that both units are switched on at the same time. Select capacitors to supply approximately (without exceeding) motor no-load magnetizing current or nearly no-load current. If the motor characteristics are not accurately known during design, check the shop drawings or confer with the manufacturer and resize the capacitor if necessary.

Conductors

The conductor ampacity should not be less than 135% of the rated capacitor current, nor should it be less than one-third of the ampacity of the motor branch circuit conductors (see NEC 460-8.a).

9-9. Engine Generators

A small, skid-mounted engine generator (gen-set) is shown in Figure 9-5.

Need for Engine Generators

The need for standby power from enginegenerators is: (1) site-specific, (2) dependent on the requirement of absolute reliability, and (3) dependent on the willingness of the owner to pay for on-site generation facilities. Diesel engines are usually preferred because they are smaller than gas engines and hence less expensive. However, storing diesel fuel on-site requires double-wall storage tanks, and deterioration of the fuel is a problem. On the other hand, if natural gas can be piped to the site, no storage is required. Gas may not be available at some sites, and in earthquake-prone areas it is not completely reliable. Pollutant emissions from gas engines is minimal, whereas it is more difficult to control pollutant emissions from diesel engines. Furthermore, regulations tend to become more stringent with time (see Section 14-14). Altogether, the need for and choice of standby power generation requires the most careful consideration.

In the field of electric power supply, substations supplied from main transmission power lines are the most reliable. Substations rarely fail even when the power lines or even the substations themselves are struck by lightning. Generally, only one or two outages occur per year. The next most reliable component is the distribution substation, which operates at lower voltage. One to six outages per year might be considered average. The least reliable are long power lines to isolated communities where outages are more frequent and last for longer periods. These lower-voltage power lines suffer more frequent power outages because of accidents, lightning, construction, and so on.

Power outages usually involve only a small area (for example, a quadrant of a small town), but occasionally an entire city or even several states are affected. Natural disasters such as hurricanes, tornadoes, earthquakes, and fires tend to create many hours or days of outages. In 1965, the blackout of November 9 and 10 left most of the northeastern United States and two Canadian provinces without electric power for most of two days. In 1977, a blackout left New York City and Long Island without electric power for several hours. Beginning October 17, 1989, the San Francisco (actually Loma Prieta) earthquake interrupted power that was not restored to all parts of the area for several days. In 1991, a grass and brush fire burned the distribution power lines to the only booster pumping station that supplied water to northeast Oakland, so there was no effective way to fight a fire that destroyed a large, upscale residential district. On July 2, 1996, a flashover between a tree and a 345,000-V transmission line triggered a chain of events that, due to record-setting demands for power, led to outages that lasted up to 6 hours in some areas throughout 15 states and affected approximately 2 million customers. More recently, hurricane damage to electrical infrastructure in Florida in 2004 left large areas of the state without power for several days and even weeks. Both water supply and wastewater transmission systems were interrupted for most of the period because of a lack of on-site power generation at most pumping and treatment facilities. A designer would indeed be wise to study thoroughly the history over the last four or five decades of power outages in the district of a pumping station and to realize that, with deregulation, things may get worse—not better.

Some pumping plants can be without power for as much as an hour (or even more) before flooding occurs, and some utilities have concluded that, for their situation, this grace period makes standby power unnecessary (see Section 24-11). On the other hand, the public has become more intolerant of utility failures. This attitude, coupled with the deregulation of power generation (a move that can only result in trimming costs by every producer to remain competitive) is likely to result in less reliable power in the future. Even though power in any given area has a history of good reliability, there is no guarantee that such reliability will continue in the future.

No blanket rules can be given. Investigate each project separately. To reach the most rational decision, examine the history of power failures, the consequences of down time, and the costs—both monetary and subjective—such as political concerns, customer exasperation, possible litigation, and so on.

Engine-Generator Sizing

Preliminary calculations for determining the engine-generator size are shown in Example 9-10. The final size, however, should be determined by the manufacturer based on an even more careful consideration of station loading and starting requirements and the characteristics of the generator and engine.

The calculations for Example 9-10 are typical, but maximum flow might be a rare occurrence of short duration, and the diversity of loads (the probability that all demands may not occur at the same time) can often reduce a power requirement. On the other hand, some heating might be required. A more extensive analysis might result in a different engine-generator unit. A typical engine-generator set is shown in Figure 9-5.

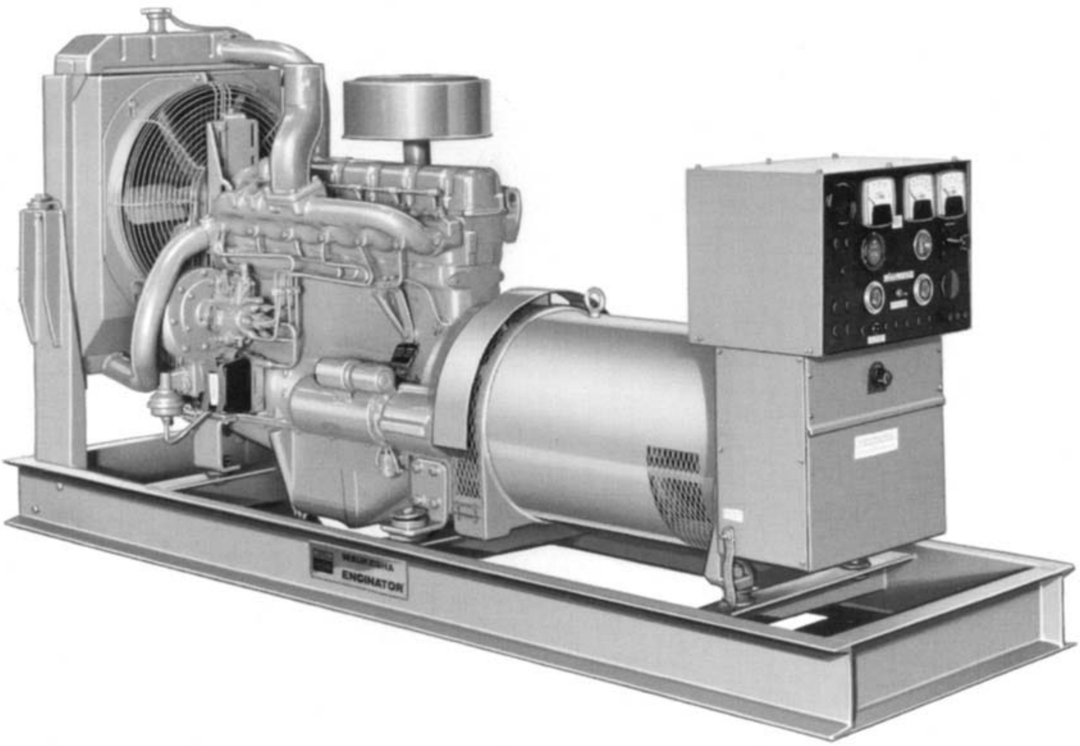


Figure 9-5. A typical engine-generator set. Courtesy of Waukeshaw Engine Div., Dresser Industries, Inc.

Example 9-10
Size of the Engine-Generator Set

Problem: Determine the generator and the engine power rating requirements for the entire pumping station described in Example 9-1. Note that as more pumps come on line, the head on the pumps increases, but the incremental flow rate decreases, with the net result that (for this pumping station) one pump operates at 21.2 motor hp, two pumps operate at 20.0 hp each, and three pumps operate at 18.4 hp each.

Solution: At full load, the power factor of a 720-rev/min motor is about 0.75, but adding capacitors to the circuit, as shown in Example 9-9, increases the power factor to 0.95. The locked-rotor power factor of a standard 1800-rev/min, 25-hp motor is 0.44, and for a 720-rev/min motor, the power factor is estimated to be 0.40. The addition of a 15-kVAR capacitor (Example 9-9) would have little effect on the locked-rotor power factor. Starting a Code F motor requires 5.3 kVA for each horsepower.

In Example 9-1, the essential loads on the generator for heating and ventilating are 5 A in the pump room and 3.5 A in the wet well. During a power outage, lighting can be restricted to 11 A. The sump pump and instrument air compressor consume 1 A each, but these loads can be ignored during motor start-up because they are small, intermittent, and not likely to exist during the worst-case inrush, which persists only momentarily. Hence, the total load with no pumps operating is $5 + 3.5 + 11 = 19.5$ A. Because personnel discomfort during power outages can be acceptable, but freezing (even of small pipelines such as for seal water supply) cannot, it may be necessary in very cold climates to include some of the heater loads in the sizing calculations.

Item	Starting kVA	Starting kW	Running kVA	Running kW
Ventilation and lighting (V&L) loads				
$19.5 \times 460\sqrt{3}/1000$				15.5
15.5/0.95 *Pf*			16.3	
Start first pump				
25 hp × 5.3 kVA/hp	132.5			
132.5 kVA × 0.40 *Pf*		53.0		
Add V&L loads	16.3	15.5	16.3	15.5
Total	148.8	68.5	16.3	15.5
Run first pump at 21.2 motor hp				
21.2 hp × 0.746 kW/hp (1/0.88 efficiency[a])				18.0
18 kW at 0.95 *Pf*			18.9	
Add V&L loads			16.3	15.5
Total load			35.2	33.5
Start second pump				
25 hp × 5.3 kVA/hp	132.5	53.0		
Run first pump and V&L	35.2	33.5	35.2	33.5
Total	167.7	86.5	35.2	33.5
Run 2 pumps at 20 motor hp each				
20 hp × 2 × 0.746 kW/hp (1/0.88 efficiency[a])				33.9
33.9 kW at 0.95 *Pf*			35.7	
Add V&L loads			16.3	15.5
Total load			52.0	49.4
Start third pump				
25 hp × 5.3 kVA/hp	132.5	53.0		
Run 2 pumps and V&L	52.0	49.4	52.0	49.4
Total	184.5	102.4	52.0	49.4
Run 3 pumps at 18.4 motor hp				
18.4 hp × 3 × 0.746 kW/hp (1/0.88 efficiency[a])			49.3	46.8
Add V&L loads			16.3	15.5
Total load			65.6	62.3

[a]Efficiency for high-efficiency motors is about 92 to 94%

The critical generator loads are the kilovolt-amperes and kilowatts for starting the third pump. During the start, the first two running motors can act momentarily (for half a cycle) as generators and, hence, reduce the demand on the generator, but the effect lasts much less than a second and is difficult to assess, so the effect is ignored, especially as it may take up to 6 s to start the third pump.

Because the calculated size of the generator is 184.5 kVA, specify a 200 kVA standby duty rating. The future flow of 4500 gal/min requires about 15% more horsepower, but only about

5% more generator output because the inrush current for the 25-hp motor does not change. Therefore, a 200-kVA generator would be adequate.

The engine can be sized on the basis of running power, partly because the inertia is so great that the engine will maintain its speed during the few seconds of high inrush current and partly because the engine can be overloaded for a few seconds.

$$\text{Size of engine} = \frac{62.3\,\text{kW}}{0.85\ \text{generator efficiency}} = 73.3\,\text{kW} = 98.2\,\text{hp}$$

Use an engine of not less than 100 hp (continuous duty rating).

This approximate analysis may be refined by methods explained in manufacturer's literature [3], but always confirm engine-generator size calculations with the manufacturer. Manufacturers may calculate required engine and generator size somewhat differently, so supply all of the worst-case figures. Also supply generator site, altitude, and ambient temperature because these affect the generator rating.

9-10. Short-Circuit Current Calculations

The purpose of short circuit current calculations is to find the maximum current available under short-circuit conditions at each required location and to be certain that the circuit interrupting devices (circuit breakers and fuses) have the ability to interrupt this level of current. It is also important to be certain that other devices required to carry this high level of current have appropriate withstand ratings so that the device will not fail at the current level. Except in some medium-voltage circuits with local on-site generation, the maximum available short-circuit current would occur if all phases or lines were connected together.

A preliminary calculation is needed during design for properly sizing, rating, and specifying equipment. The designer must use the best available data in making this calculation, although the equipment manufacturer and the exact data may be unknown.

If the expected fault currents approach equipment interrupting and withstand ratings (typically 22,000 A at 480 V), a final calculation is made during construction to verify the equipment ratings and set adjustable overcurrent protective devices. This work is often done by the equipment supplier using a computer for speed and accuracy. Make sure that the specifications detail this requirement properly. Spot check the calculations to ensure that the proper field data are used.

Fault (Short-Circuit) Current Magnitude

The magnitude of a fault current depends primarily on the capacity of the supply system, but on-site motors and generators can contribute to the fault current. In Figure 9-1, the characteristics of each contributor are:

- Utility. The fault current magnitude is assumed constant during the fault period.
- Local generators are not usually on line in pumping stations. If they are, the magnitude of the fault current decreases during the fault period. Practically, the fault current may be assumed to be constant in low-voltage systems.
- Synchronous motors act as synchronous generators driven by load and rotor inertias. The fault current magnitude decreases substantially during the fault period.
- Induction motors. Current is generated while the magnetic field is present. The induction motor's field decreases rapidly when the driving voltage drops to a small value (as in a fault or when the contactor or circuit breaker opens). The fault current magnitude is initially about that of the locked rotor current, but it decreases rapidly, usually over a few cycles. If the running power factor is corrected to near unity, the magnitude may not decrease as expected due to self-excitation.

The total or asymmetrical fault current is composed of an ac component plus a decaying dc component. The initial magnitude of the dc component depends on the circuit reactance/resistance (X/R) and on the instantaneous voltage of the faulted phase at the instant the fault occurs. To allow for the dc component, multiply the symmetrical value by 1.6 (for high-voltage fused starters, current-limiting fuses, and circuit breakers) or by 1.25 (for low-voltage fuses).

It is the asymmetrical current that must be interrupted by low-voltage breakers and fuses due to short (e.g., 1.5 cycles) interrupting times. Low-voltage breakers, however, are now rated in symmetrical current, so the asymmetrical current need be computed

only for fuses. Medium-voltage breakers and fused switches have both withstand and interrupting ratings (which are lower due to their longer interrupting times), so the dc decay must be taken into account. For sample calculations, consult the literature [4, 5, 6].

Coordination

A coordinated system is one in which the overcurrent devices are rated and adjusted so that the device just ahead of a fault will safely clear the fault, dropping only the downstream part of the circuit. If that device should fail to clear a fault, the next upstream device will clear after a suitable time delay to give the proper device a chance to operate. In a coordinated system, the withstand ratings of equipment and cable are suitable for withstanding fault currents for the time needed to clear the fault without damage to the insulation (see the UBC for a detailed discussion).

9-11. Harmonics

In pumping stations with adjustable frequency drives, excessive harmonic contribution to the electrical power system must be prevented. Where the adjustable frequency drive load exceeds 25% of the station load or where adjustable frequency drives must be powered by an engine-generator, a harmonic calculation should be made and suitable filtering specified. IEEE 519 explains how to make harmonic calculations and specifies harmonic current limits for systems connected to electric power utilities. Where adjustable frequency drives are to be powered by an engine-generator, consult the manufacturer regarding limits for that specific equipment.

Power factor correction capacitors must not be connected to power systems with a harmonic voltage distortion in excess of 5% without suitable filters that exclude harmonic currents from the capacitors.

Past practice in the industry has been to make the drive vendors responsible for harmonic calculations. As with short-circuit calculations discussed in Section 9-10, harmonic calculations are best performed using suitable computer software.

9-12. Construction Service

A thorough review of all submitted material is a time-consuming, unpopular, and unrewarding task, but *it is vital.*

Shop Drawing Review

The shop drawing review is the last chance to make a design change at a reasonable cost. Once construction has begun, changes are very expensive.

Allow no deviations without good reason and without a reduction in the contract price. Document all revisions, and keep the project manager and contractor fully aware of all conversations with suppliers that result in any revisions.

Obtain and review the electrical portions of submittals under other specification sections such as mechanical, process, or instrumentation.

Submittal Checklist

Check submittals for conformance to the drawings, specifications, applicable codes, addenda, and change orders. Checks of the following elements are a minimum:

- Voltage, phases, and other nameplate data
- Control (schematic) diagrams
- Wiring and interconnection wiring diagrams
- Control and alarm interfaces with other equipment
- Components
- Wiring details
- Inclusion and acceptability of required factory test reports.

Release the generator shop drawings to the general contractor only after including the following warning: "Accepted pending verification that *you* have coordinated the generator sizing with your pump and motor suppliers." If the warning is not included, the station's generator will be inadequate if the motor size is increased.

Inspection

Adequate inspection is required to assure the owner and authorities that the construction meets code, specification, and drawing requirements, so several inspectors, including city/county personnel in addition to the resident inspector or engineer, are likely to visit the job site. The designer should make spot checks, and there should be a final, detailed inspection. The following are suggestions for the resident inspector:

- Be as thorough as time allows. Have the contractor make the interiors accessible for inspection and observe the interior of as much equipment as pos-

sible. Note the condition of workmanship as well as conformance to drawings.
- Check all work against drawings, specifications, addenda, change orders, shop drawings, and codes.
- Make a written record of each deviation, whether it is to be corrected or left as is, and give the record to the project manager for transmittal to the contractor.

Tests

This section is limited to electrical system tests.

Factory Tests

Require the following factory tests:

- *Switchboard:* Test per NEMA Standard PB2.
- *Motor control center:* Test per NEMA Standard ICS.
- *Transformer, oil filled:* Test per ANSI C57.12.90.
- *Standby generator:* Run the unit at the rated load and power factor for at least 2 hr to demonstrate set performance; require the contractor to provide a load bank suitable for continuous duty, full-load operation of the engine-generator, and require complete mechanical and electrical records and fully describe witnessed testing.
- *Automatic transfer switch:* Test for proper operation in all modes.

Field Tests

The following tests are required:

- *480-V wiring:* With contactors closed, measure and record the insulation resistance to ground. Motors may be connected. Do not record or accept "infinity." Use a 500-V "megger" (megohm meter).
- *480-V motors and 230-V main pump motors:* Separately measure and record the insulation resistance to ground. Do not record infinity. Reject any measurements below 1 MΩ. Use a 500-V megger.
- *Circuit breaker solid-state trip units:* Have an independent, factory-certified testing organization field test and certify accuracy, operation, and adjustment.
- *Power sources:* Check the voltage at the service, at the lighting transformer, and at the generator terminals for amplitude and balance (loaded and unloaded). If the voltage is not within 5% of nominal or if the unbalance exceeds 1%, notify the power company, adjust generator voltage regulator, or reconnect taps as applicable.
- *Tighten all electrical connections:* Note that circuit breakers inherently protect against single phasing except in the case of a loose connection.
- *Motor phase currents:* Measure these currents for each pump motor at full load and check for overloading and imbalance. Note that a 1% voltage imbalance will cause an approximately 8% current imbalance. Excessive current imbalance will cause motor overheating. Consult with the equipment supplier when in doubt.
- *Motor and generator rotation:* Correct rotation, if necessary, by interchanging any two-line conductors on three-phase units.
- *Operational test:* Take nothing for granted. Check and test everything for proper operation: prevention of single phasing, lights, GFCI outlet circuit protection, each mode of the automatic transfer switch, standby generator alarms and shut-downs, motor space heaters, etc.
- *Motor protection list:* Record the following data and verify correctness to ensure proper motor protection for *each* motor on the job:
  (1) Unit name or designation
  (2) Motor nameplate data (horsepower, voltage, service factor, current, and temperature rating)
  (3) Circuit breaker or fuse rating
  (4) Circuit breaker setting, if any
  (5) Starter size
  (6) Overload relay manufacturer and heater catalog number
  (7) Capacitor size, if any.

Obtain a copy of the overload relay manufacturer's heater table with instructions for selecting heaters. Check each motor branch circuit for proper protection and adjustments.

9-13. References

1. *CSI Documents MP-2-1, Master Format, Master List of Section Titles and Numbers (1983 ed.)* and *MP-2-2 Section Format (1981 ed.),* Construction Specifications Institute, Alexandria, VA.
2. Kaufman, J. E., and H. Haynes (Eds.), *IES Lighting Handbook*, Illuminating Engineering Society of North America, New York (1981).

3. *Generator Set Sizing Guide*, Bulletin LE BX 6220. Caterpillar Inc., Peoria, IL.
4. *Short-Circuit Current Calculation*, General Electric Co., Schenectady, NY.
5. IEEE Standard 242-1975, *Recommended Practice for Protection and Coordination of Industrial and Commerical Power Systems*, Institute of Electrical and Electronic Engineers, Inc., Wiley-Interscience, Piscataway, NJ (1976).
6. IEEE Standard 141-1976, *Recommended Practice for Electric Power Distribution for Industrial Plants*, Institute of Electrical and Electronics Engineers, Inc., Wiley-Interscience, Piscataway, NJ (1976).

Chapter 10

Performance of Centrifugal Pumps

PAUL COOPER
GEORGE TCHOBANOGLOUS

CONTRIBUTORS
Richard O. Garbus
Robert J. Hart
Carl W. Reh
Lowell G. Sloan
Earle C. Smith

The purpose of this chapter is to introduce those fundamentals of centrifugal pump theory that are useful for background and sometimes necessary for selecting and specifying centrifugal pumps in water and wastewater pumping applications. A corresponding body of theory for positive-displacement pumps can be found in the literature [1, 2, 3].

This introduction includes: (1) the general classifications of centrifugal pumps, (2) pump application terminology and usage, (3) pump operating characteristics, (4) cavitation and net positive suction head, (5) pump characteristic curves and operating ranges, and (6) an introduction to pumping system analysis.

Standards and specifications such as ANSI/HI 9.8-1998 are commonly (and sufficiently) identified by the call number and, hence, are not referenced in Section 10-10.

10-1. Classification of Centrifugal Pumps

In colloquial usage in the United States, a "centrifugal pump" is any pump in which the fluid is energized by a rotating impeller, whether the flow is radial, axial, or a combination of both (mixed). Strictly defined (as in European practice), a centrifugal pump is a radial-flow pump only. But colloquial usage is followed here, and thus centrifugal pumps are divided into three groups:

- Radial-flow pumps
- Mixed-flow pumps
- Axial-flow or propeller pumps.

These classifications are derived from the manner in which the fluid moves through the pump (see Figure 10-1). Thus, the fluid is displaced radially in a radial-flow pump, axially in an axial-flow pump, and both radially and axially in a mixed-flow pump. The physical features of centrifugal pumps are described in greater detail in Chapter 11.

10-2. Pump Application Terminology, Equations, and Performance Curves

The basic terminology, equations, and curves for defining pump performance and solving pump problems are given in this section. The development of the

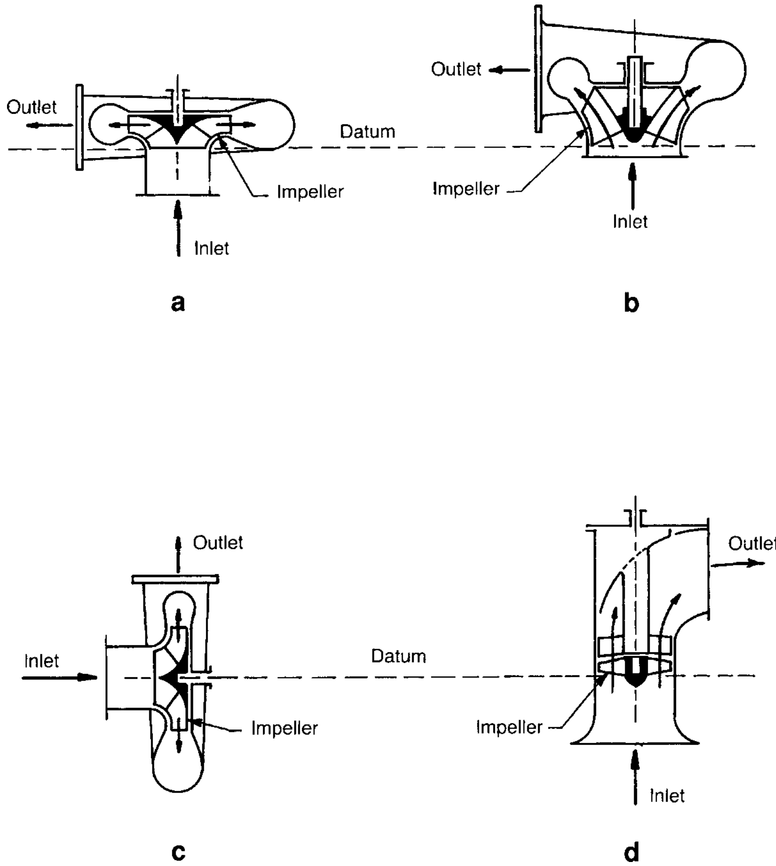


Figure 10-1. Typical flow paths in centrifugal pumps. (a) Radial flow, vertical; (b) mixed flow; (c) radial flow, horizontal; (d) axial flow.

pump performance curves, which are used to define the operating characteristics of a given pump, is reviewed briefly at the end of this section (this subject is covered in more detail in Chapter 12).

Capacity

The flow rate (capacity, discharge, or Q) of a pump is the volume of liquid pumped per unit of time, usually expressed in SI units in cubic meters per second (m^3/s) for large pumps or liters per second (L/s) and cubic meters per hour (m^3/h) for small pumps. In U.S. customary units, the capacity of a pump is expressed in gallons per minute (gal/min), million gallons per day (Mgal/d), or cubic feet per second (ft^3/s). Equivalent units of measurement are given on the inside back cover and in Appendix A.

Head

The term head (h or H) is the elevation of a free surface of water above (or below) a reference datum (see Figures 10-2 and 10-3). For centrifugal pumps, the reference datum varies with the type of pump, as shown in Figure 10-1. Head is expressed in meters (m) or feet (ft). Pressure can also be expressed as the equivalent head of water. See Appendix A.

Distances (heads) above the datum are considered positive and distances below the datum are considered negative. Each term, defined graphically in Figures 10-2 and 10-3, is expressed as the height of a water column in meters (feet) of water. H is used for total head, whereas h is used for head from the datum or for headloss. The subscripts s and d denote the pump suction and discharge, respectively. Other subscripts are defined as follows:

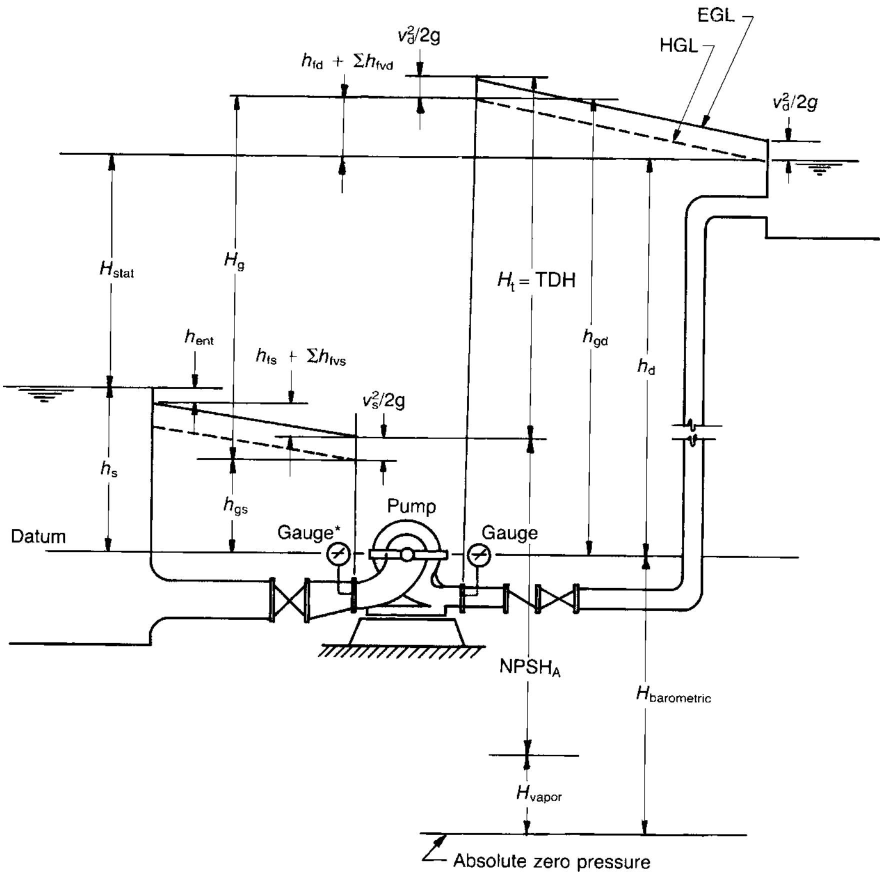


Figure 10-2. Terminology for a pump with a positive suction head. (*) The gauge is located to show theoretical pressures at the inlet and outlet flanges; see "Field Pump Tests" in Section 16-6 for practical gauge locations.

Total static head (H_{stat}): The total static head is the difference in elevation in meters (feet) between the water level in the wet well and the water level at discharge ($h_d - h_s$).

Static suction head (h_s): The static suction head is the difference in elevation between the wet well liquid level and the datum elevation of the pump impeller. If the wet well liquid level is below the pump datum, as in Figure 10-3, it is a static suction lift, so h_s is negative.

Static discharge head (h_d): The static discharge head is the difference in elevation between the discharge liquid level and the pump datum elevation.

Manometric suction head (h_{gs}): The suction gauge reading is expressed in meters (feet) measured at the suction nozzle of the pump and referenced to the pump datum elevation and atmospheric pressure.

Manometric discharge head (h_{gd}): The discharge gauge reading is expressed in meters (feet) measured at the discharge nozzle of the pump and referenced to the centerline of the pump impeller. The gauge reading is the height that a water column would attain in a vertical pipe. It is also the distance to the hydraulic gradeline (shown dashed in Figures 10-2 and 10-3).

Manometric head (H_g): The increase of pressure head, expressed in meters (feet) generated by the pump ($h_{gd} - h_{gs}$).

Friction headloss (h_{fs}, h_{fd}): The head of water that must be supplied to overcome the frictional loss in the pipe. The frictional headloss in the suction (h_{fs}) and discharge (h_{fd}) piping systems can be computed with the Hazen–Williams or Darcy–Weisbach equations (Equations 3-9 and 3-10).

Velocity head ($v^2/2g$): The velocity head is the kinetic energy in the liquid being pumped at any point in the system. The energy gradeline (shown solid in Figures 10-2 and 10-3) is always above the hydraulic or piezometric or manometric gradeline (dashed line) by $v^2/2g$. The velocity head in the discharge pipe, $v_d^2/2g$, is lost if the pipe discharges freely in air or if it discharges abruptly below the surface of

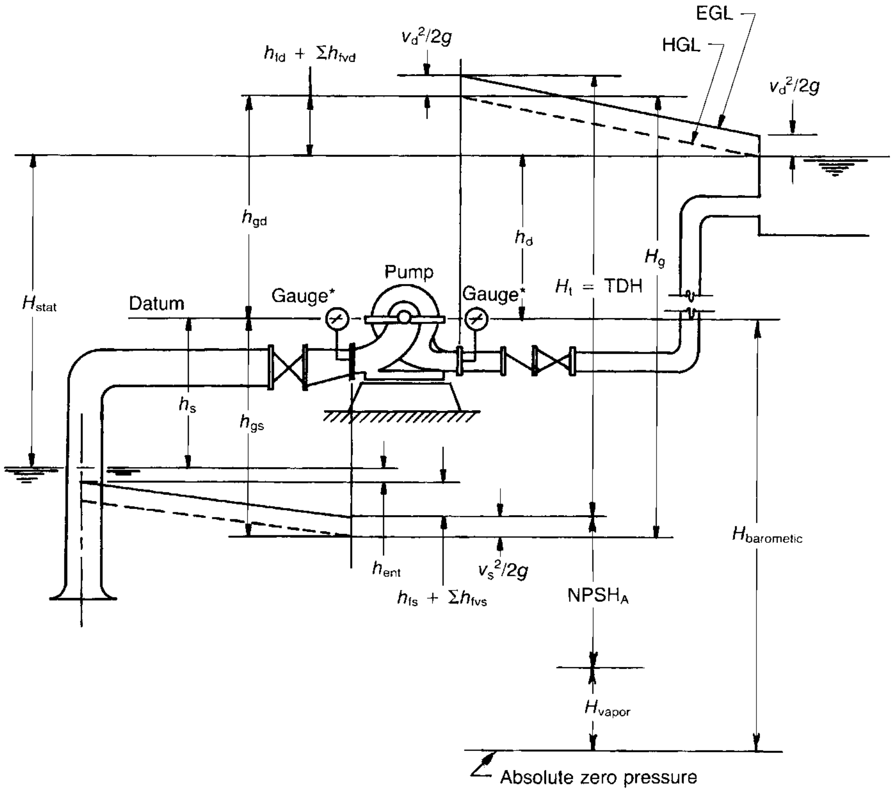


Figure 10-3. Terminology for a pump with a negative suction head. (*) The gauge location is to show theoretical pressures at the inlet and outlet flanges; see "Field Pump Tests" in Section 16–6 for practical gauge locations.

a reservoir. Some of the velocity head can be recovered if turbulence is inhibited by a gradually expanding section, but this is ordinarily impractical with the pipe velocities normally encountered (i.e., up to 2.5 m/s or 8 ft/s).

Fitting and valve losses (h_{fvs}, h_{fvd}): As a fluid flows through fittings and valves, energy is lost due to eddy formation and turbulence. Because the head lost in fittings and valves is small compared with the friction loss in long piping systems, the losses in fittings and valves are sometimes called "minor losses" and often ignored. But the lengths of pipes within a pumping station are short, and the total headloss through the fittings and valves is likely to be greater than the pipe friction loss. Regardless of the shortness of pipe length, both the frictional headloss and the "minor losses" should always be computed.

The loss of head through each individual fitting or valve is estimated using Equation 3-16 ($h = Kv^2/2g$) with K values taken from Tables B-6 and B-7 (or from the literature). The total fitting and valve losses in the piping system are determined by summing the individual losses for each fitting and valve, but, as explained in Section 3-4, the total may in reality be somewhat less or much greater than this sum.

Total dynamic head (H_t or *TDH*): The total dynamic head is the head against which the pump must work. It is determined by adding the static suction and discharge head (with respect to signs), the frictional headlosses, the velocity heads, and the fitting and valve headlosses. The expression for determining the total dynamic head for the pumps shown in Figures 10-2 and 10-3 is given by Equations 10-1, 10-2, and 10-3.

$$H_t = h_{gd} - h_{gs} + \frac{v_d^2}{2g} - \frac{v_s^2}{2g} \tag{10-1}$$

where

$$h_{gd} = h_d + h_{fd} + \Sigma h_{fvd} \tag{10-2}$$

and

$$h_{gs} = h_s - h_{ent} - h_{fs} - \Sigma h_{fvs} - \frac{v_s^2}{2g} \tag{10-3}$$

Substituting Equations 10-2 and 10-3 into Equation 10-1 and noting that $h_d - h_s = H_{stat}$,

$$H_t = H_{stat} + h_{ent} + h_{fs} + h_{fd} + \Sigma h_{fvs} + \Sigma h_{fvd} + \frac{v_d^2}{2g} \quad (10\text{-}4)$$

Some designers consider all headlosses except pipe friction to be "minor" losses, h_m, and rewrite Equation 10-4 as

$$H_t = H_{stat} + h_{fs} + h_{fd} + \Sigma h_m \quad (10\text{-}5)$$

Power

Output Power

The power output of a pump is the useful energy delivered by the pump to the fluid. In SI units, the power output is defined as

$$P = \gamma Q H = \frac{qH}{102} \quad (10\text{-}6a)$$

where P is the water power in kilowatts, γ is the specific weight of the fluid in kiloNewtons per cubic meter (kN/m^3, see Table A-8), Q is the flow rate in m^3/s, H is the total dynamic head in m, q is the flow rate in L/s, and 102 is a conversion factor for water at 15 to 20°C.

In U.S. customary units, power output is defined as

$$P = \frac{\gamma Q H}{550} = \frac{WH}{33{,}000} = \frac{qH}{3960} \quad (10\text{-}6b)$$

where P is the water power in horsepower (hp), γ is the specific weight of the fluid in pounds per cubic foot (lb/ft^3, see Table A-9), Q is the flow rate in ft^3/s, H is the total dynamic head in ft, 550 is the conversion factor from foot-pounds per second to horsepower, W is weight of water pumped in pounds per minute (lb/min), 33,000 is the conversion factor from foot-pounds per minute to horsepower, q is flow rate in gal/min., and 3960 is used to convert gallon-feet per minute to horsepower for water at 15 to 20°C (59 to 68°F).

Input Power

Pump performance is measured in terms of the flow rate that a pump can discharge against a given head at a given efficiency. The pump capacity depends on the design, and design information is furnished by the pump manufacturer in a series of curves for a given pump. Pump efficiency, E_p, is the ratio of the useful power output [water kilowatts (wkW) or water horsepower (whp)] to the power input to the pump shaft. Hence, the brake power (bkW) that must be supplied by the drive is, in SI units,

$$\text{bkW} = \gamma QH/E_p = \text{wkW}/E_p \quad (10\text{-}7a)$$

The power input (bhp) in U.S. customary units is

$$\text{bhp} = \frac{\gamma Q H}{550 E_p} = \frac{qH}{3960 E_p} = \frac{\text{whp}}{E_p} \quad (10\text{-}7b)$$

Pump efficiencies usually range from 40 to 92% and increase with the size of the pump (see Figure 10-4). Energy losses in a pump are volumetric, mechanical,

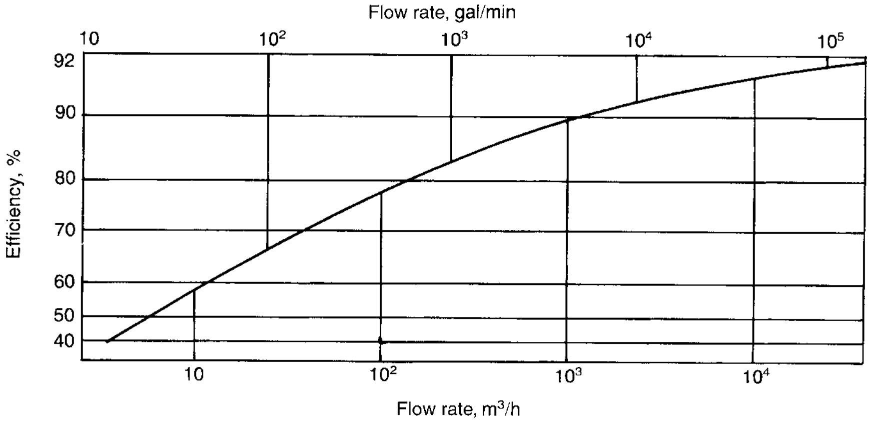


Figure 10-4. Maximum pump efficiency attainable at the best efficiency point and at optimal specific speed.

and hydraulic. Volumetric losses are those of leakage through the small clearances between wearing rings in the pump casing and the rotating element. Mechanical losses are caused by mechanical friction in the stuffing boxes and bearings, by internal disc friction, and by fluid shear. Frictional and eddy losses within the flow passages account for the hydraulic losses.

Example 10-1
Evaluation of Pump Performance

Problem: Water is pumped by a radial-flow centrifugal pump at 20°C (68°F) through a new cement-mortar-lined ductile iron piping (DIP) system such as shown in Figure 10-3, except that there are two 90-degree elbows and a gate valve in the suction piping, and a gate valve and three 90-degree elbows in the discharge piping. Other data known, measured, or computed are

- Flow rate, $Q = 0.1\,\text{m}^3/\text{s}$ ($3.54\,\text{ft}^3/\text{s}$)
- Total static head, $H_{\text{stat}} = 14\,\text{m}$ (45.93 ft)
- Static suction lift, $h_s = 2\,\text{m}$ (6.56 ft)
- Power input to pump shaft, $P = 22.8\,\text{kW}$ (30.5 hp)
- Suction piping = 3.3 m of 300-mm (11 ft of 12-in.) pipe
- Discharge piping = 47.6 m of 250-mm (156 ft of 10-in.) pipe.

Evaluate: (1) the total dynamic head, (2) the power delivered to the water, and (3) the efficiency of the pumping unit.

Solution: Because pipe is exposed and joints are flanged within the pumping station, Class 53 is needed (see Tables B-1 and B-2 for dimensions). Outside, the pipe is buried, joints can be mechanical or push on, and the pipe can be Class 50 with a net ID of 264 mm (10.4 in.), which includes the lining (see the DIPRA handbook [3]). Because most of the discharge piping can be Class 50, ignore the slightly smaller diameter of the 250-mm (10-in.) Class 53 exposed piping within the pumping station.

Total dynamic head: Find velocity, velocity head, and friction losses in suction and discharge piping using Equation 10-4.

SI Units	**U.S. Customary Units**
For the suction piping:	
ID = 312 mm (Table B-1)	ID = 123 in. (Table B-2)
$A_s = 0.0763\,\text{m}^2$ (Table B-1)	$A_s = 0.822\,\text{ft}^2$ (Table B-2)
$v_s = 1.31\,\text{m/s}$	$v_s = 4.31\,\text{ft/s}$
$v^2/2g = 0.0875\,\text{m}$	$v^2/2\text{g} = 0.288\,\text{ft}$

Find the entrance loss assuming a bell mouth entrance is used (see Table B-6).

SI Units	U.S. Customary Units
$h = 0.05 \times 0.0875 = 0.004\,\text{m}$	$h = 0.05 \times 0.288 = 0.014\,\text{ft}$

Find the friction headloss using Equations 3-9a and 3-9b.

SI Units	U.S. Customary Units
$h_{fs} = 10{,}700\,(Q/C)^{1.85} D^{-4.87}$	$h_{fs} = 10{,}500\,(Q/C)^{1.85} D^{-4.87}$
$h_{fs} = 10{,}700\,(0.1/145)^{1.85} \times (0.3121)^{-4.87}$	$h_{fs} = 10{,}500\,(1590/145)^{1.85} \times (12.3)^{-4.87}$
$h_{fs} = 4.41\,\text{m}/1000\,\text{m}$	$h_{fs} = 4.34\,\text{ft}/1000\,\text{ft}$
$h_{fs} = 4.41\,(3.3\,\text{m}/1000\,\text{m}) = 0.015\,\text{m}$	$h_{fs} = 4.34\,(11\,\text{ft}/1000\,\text{ft}) = 0.048\,\text{ft}$

The "minor" losses are found using Equation 3-16 ($h = Kv^2/2g$). Note that there is an elbow as well as a gate valve in the suction piping. Obtain K values from Tables B-6 and B-7.

SI Units	U.S. Customary Units
h (bend) $= 0.25 \times 0.0875 = 0.022\,\text{m}$	h (bend) $= 0.25 \times 0.288 = 0.072\,\text{ft}$
h (gate) $= 0.2 \times 0.0875 = 0.018\,\text{m}$	h (gate) $= 0.2 \times 0.288 = 0.058\,\text{ft}$
$\Sigma h_{fvs} = 0.040\,\text{m}$	$\Sigma h_{fvs} = 0.130\,\text{ft}$

For the discharge piping:

ID $= 0.264$ m	ID $= 10.4$ in.
$A_d = 0.0548$ m^2	$A_d = 0.589$ ft^2
$v_d = 1.82$ m/s	$v_d = 6.01$ ft/s
$v_d^2/2g = 0.169$ m	$v_d^2/2g = 0.561$ ft

Find the friction headloss from Equations 3-9a and 3-9b.

$$h_{fd} = 10{,}700\,(0.1/145)^{1.85}(0.264)^{-4.87} = 9.95\text{ m}/1000\text{ m}$$

$$h_{fd} = 9.95\,(47.6\text{ m}/1000\text{ m}) = 0.474\text{ m}$$

$$h_{fd} = 10{,}500\,(1590/145)^{1.85}(10.4)^{-4.87} = 9.82\text{ ft}/1000\text{ ft}$$

$$h_{fd} = 9.82\,(156\text{ ft}/1000\text{ ft}) = 1.53\text{ ft}$$

Find the "minor" losses using Equation 3-16.

h (check) $= 2.20 \times 0.169 = 0.372$ m	h (check) $= 2.20 \times 0.561 = 1.23$ ft
h (gate) $= 0.20 \times 0.169 = 0.034$	h (gate) $= 0.20 \times 0.561 = 0.11$
h (bends) $= 2\,(0.25 \times 0.169) = 0.084$	h (bends) $= 2\,(0.25 \times 0.561) = 0.28$
$\Sigma h_{fvd} = 0.490$	$\Sigma h_{fvd} = 1.62$

The TDH computed using Equation 10-4 is

$H_{stat} = 14.00$ m	$H_{stat} = 45.93$ ft
$h_{ent} = 0.004$	$h_{ent} = 0.014$
$h_{fs} = 0.015$	$h_{fs} = 0.048$
$h_{fd} = 0.474$	$h_{fd} = 1.53$
$\Sigma h_{fvs} = 0.040$	$\Sigma h_{fvs} = 0.130$
$\Sigma h_{fvd} = 0.490$	$\Sigma h_{fvd} = 1.62$
$v_d^2/2g = 0.169$	$v_d^2/2g = 0.561$
TDH $= 15.192$ m	TDH $= 49.833$ ft
Use 15.2 m	Use 49.9 ft

The slight discrepancies between SI and U.S. values are caused by rounding off coefficients and powers to three significant figures. All of the terms in Equation 10-4 are included in these additions. No allowance was made for the proximity of fittings or for swirling (see Section 3-4). The friction loss in the discharge piping is 0.474 m (1.53 ft), but only about 10% of it would be within the pumping station. The total pipe friction loss within the pumping station, therefore, would be about (0.1) (0.474) + 0.015 = 0.062 m (0.2 ft), whereas all the "minor" losses (due to fittings and valves) total about 0.53 m (1.75 ft)—almost 10 times as much.

Power delivered to the water. Use Equations 10-6a and 10-6b.

$$P = \gamma QH = (9.789\,\text{kN/m}^3)(0.1\,\text{m}^3/\text{s})(15.2\,\text{m}) = 14.9\,\text{wkW}$$

$$P = \gamma QH/550 = \frac{(62.32\,\text{lb/ft}^3)(3.54\,\text{ft}^3/\text{s})(49.9\,\text{ft})}{[(550\,\text{ft}\cdot\text{lb})/\text{s}]/\text{hp}} = 20.0\,\text{whp}$$

Efficiency of the pumping unit. Use Equations 10-7a and 10-7b.

$$E_p = \frac{\text{wkW}}{\text{bkW}} \qquad E_p = \frac{14.9}{22.8} = 65\%$$

$$E_p = \frac{\text{whp}}{\text{bhp}} \qquad E_p = \frac{20.0}{30.5} = 65\%$$

Pump Performance Curves

The head that a pump can produce at various flow rates and rotational speeds is established in pump tests conducted by the pump manufacturer. During testing, the capacity of the pump is varied by throttling a valve in the discharge pipe, and the corresponding head is measured. The results of these tests and other tests with different impeller diameters are plotted to obtain a series of head-capacity (H-Q) curves for the pump at some given speed (see Figure 10-5). Simultaneously, the power input to the pump is measured. The efficiency at various operating points is computed, and these values are also plotted in the same diagram. Together, these curves are known as "pump characteristic curves." However, pumps are not typically capable of operating continuously or for protracted periods at all positions along their characteristic curves. Severe damage can result from continuous operation too close to shut-off or too far to the right of the best efficiency point (BEP). In accordance with a standard adopted by the Hydraulic Institute in 1997 (ANSI/HI 9.6.3), pump manufacturers will indicate the limits of acceptable operation on pump characteristic curves. Refer to Section 10-6 for more information.

10-3. Pump Operating Characteristics

The operating characteristics of pumps depend on their size, speed, and design. Pumps of similar size and design are produced by many manufacturers, but they vary somewhat because of slight design modifications. The basic relationships that can be used to characterize and analyze pump performance under varying conditions include

- Energy transfer in pumps
- Flow, head, and power coefficients
- Affinity laws
- Specific speed

Energy Transfer in Radial Centrifugal Pumps

The transfer of energy in centrifugal pumps is based on the momentum principle, which, as applied to centrifugal pumps, can be stated as follows: "A net torque applied to a pump impeller causes a change in the angular momentum of the fluid." It is this change in angular momentum that develops head or pressure as the fluid passes through a pump impeller.

In Figure 10-6, the net torque, T, applied to the pump impeller equals the difference between the

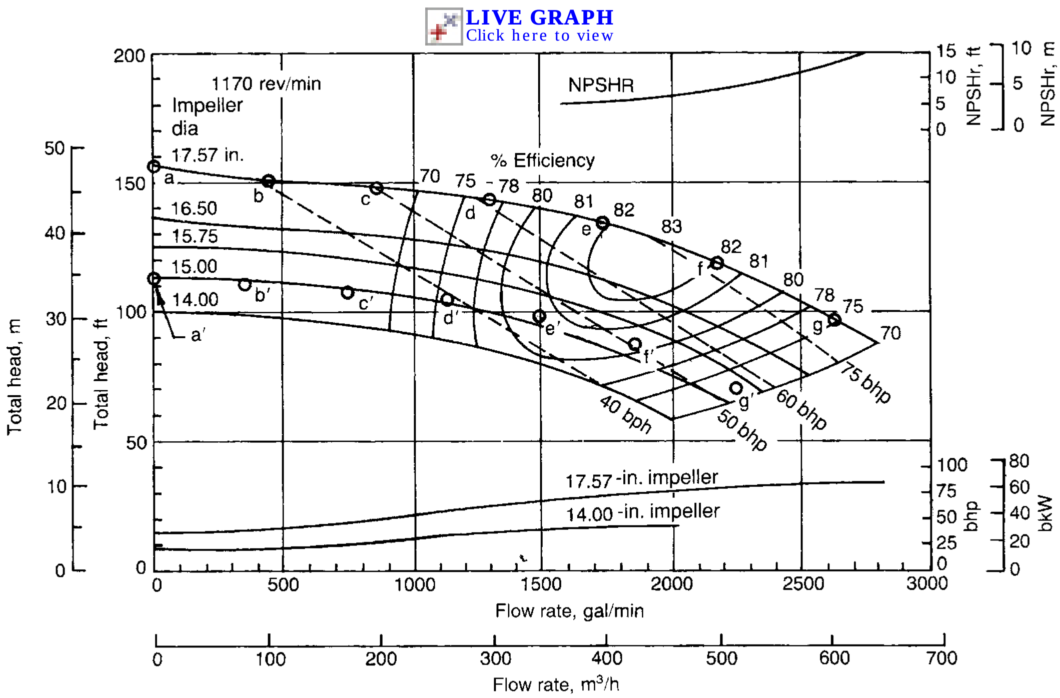


Figure 10-5. Typical pump characteristic curve. Double suction impeller.

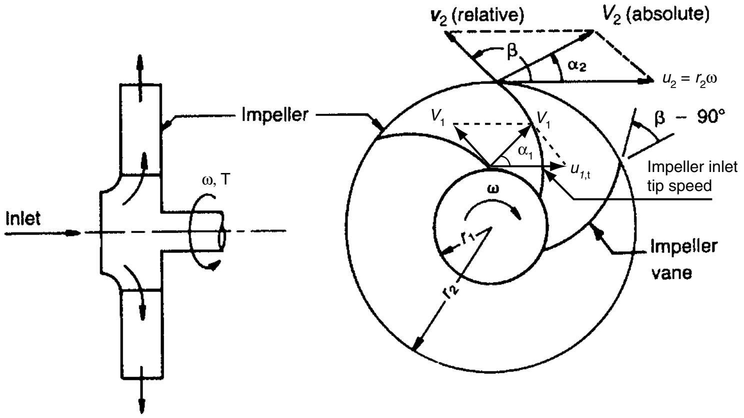


Figure 10-6. Velocity diagrams for a radial-flow pump impeller. *V*, absolute velocity; ν, relative velocity; *u*, peripheral velocity; α_1, angle to V_1 at the hub; α_2, angle to V_2 at the periphery; β, angle to ν at the periphery; ω, rotational velocity; T, torque.

moment of momentum at the inlet and outlet of the impeller. For ideal conditions, the net torque is given by

$$T = \frac{\gamma}{g} Q\,(r_2 V_2 \cos\alpha_2 - r_1 V_1 \cos\alpha_1) \qquad (10\text{-}8)$$

To obtain the power, the torque is multiplied by the angular velocity.

$$P = T\omega = \gamma Q H_{th} \qquad (10\text{-}9)$$

where P is the shaft power, T is the torque, ω is the angular velocity (rotational speed) in radians per second, and H_{th} is the theoretical head.

The theoretical head, H_{th}, added to the fluid by the pump can be obtained by equating T in Equations 10-8 and 10-9

$$\frac{\gamma}{g} Q\,(r_2 V_2 \cos\alpha_2 - r_1 V_1 \cos\alpha_1) = \frac{\gamma Q H_{th}}{\omega}$$

Substituting u/ω for r

$$H_{th} = \frac{(u_2 V_2 \cos\alpha_2 - u_1 V_1 \cos\alpha_1)}{g} \qquad (10\text{-}10)$$

For radial flow, $\cos\theta_1 = 0$ and

$$H_{th} = \frac{u_2 V_2 \cos\alpha_2}{g} \qquad (10\text{-}11)$$

A similar set of equations can be derived for mixed- and axial-flow pumps. From Equation 10-10, it can be seen that the theoretical head developed by a pump depends only on the increase of the moment of velocity within the impeller and is independent of specific weight and viscosity of the fluid (see the literature [2, 4–10] for a more complete discussion of pump theory). The actual total dynamic head, *TDH,* or simply H of the pump is less than the theoretical head, H_{th}, by the amount equal to the head losses of the fluid as it flows within the pump. Therefore, H is the product of the theoretical head H_{th} and the pump hydraulic efficiency E_{p}.

Flow, Head, and Power Coefficients

In centrifugal pumps, similar flow patterns occur in a series of geometrically similar pumps. By applying the principles of dimensional analysis, the following three independent dimensionless groups can be derived to describe the operation of centrifugal pumps and other rotodynamic machines. Note that the equations are the same in either SI or U.S. customary units. Only the value of the coefficient changes.

$$C_{\mathrm{Q}} = \frac{Q}{nD^3} \qquad (10\text{-}12)$$

$$C_{\mathrm{H}} = \frac{H}{n^2 D^2} \qquad (10\text{-}13a)$$

or, for dimensional correctness,

$$C_{\mathrm{H}} = \frac{gH}{n^2 D^2} \qquad (10\text{-}13b)$$

$$C_{\mathrm{P}} = \frac{P}{\rho n^3 D^5} \qquad (10\text{-}14)$$

where C is a coefficient and the subscripts Q, H, and P correspond to capacity, head, and power. Q is flow rate in m^3/s (ft^3/s), n is speed in radians per second (revolutions per minute or rev/min), D is impeller diameter in m (ft), H is head in m (ft), g is $9.81\,m/s^2$ ($32.2\,ft/s^2$), P is the power input in kilowatts or kW (horsepower or hp), and ρ is the density in kilograms per cubic meter (slugs per cubic foot). Although Equation 10-13b is dimensionally correct, Equation 10-13a is commonly used in the United States for both SI and U.S. customary units.

Equations 10-12 through 10-14 apply only to "corresponding points," the operating points at which similar flow patterns occur. Thus, every point on a head-capacity curve for a large pump corresponds to a point on the head-capacity curve of a geometrically similar but smaller pump that operates at the same speed. This correspondence can be corrected for different speeds provided that the speeds differ by no more than about 40%.

Affinity Laws

For a pump operating at two different speeds, the following relationships can be derived from Equations 10-12 through 10-14. (Note that neither diameter nor density changes.)

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2} \tag{10-15}$$

$$\frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \tag{10-16}$$

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{10-17}$$

where Q is flow rate, H is head, P is power, n is rotational speed, and subscripts 1 and 2 are *only* for corresponding points. *Note that Equations 10-15 and 10-16 must be applied simultaneously to ensure that Point 1 "corresponds" to or is "homologous" with Point 2. Corresponding points fall upon parabolas through the origin. They do not fall upon system H-Q curves.* These relationships, known collectively as "the affinity laws," are used to determine the effect of changes in speed on the capacity, head, and power of a pump.

The affinity laws for flow rate and head are accurate as found from actual tests for all types of centrifugal pumps, including axial-flow pumps. The affinity law for power is not as accurate because efficiency increases with an increase in the size of the pump. If the affinity law for power is used, the computed value of power can be corrected using an equation of the type proposed by Moody [2, 7, 8], which accounts for efficiency as a function of pump size, but it is often necessary to modify that equation (based on experience) as recommended by the pump manufacturer.

In applying these relationships, remember that they are based on the assumption that the efficiency remains the same when transferring from a given point on one pump curve to a homologous point on another curve. Because the hydraulic and pressure characteristics at the inlet, at the outlet, and through the pump vary with the flow rate, the errors produced by Equation 10-17 may be excessive, although errors produced by Equations 10-15 and 10-16 are very small. The application of these relationships is illustrated in Example 10-2 and in Section 15-3.

Example 10-2
Application of Affinity Laws

Problem: A pump with a normal operating speed of 705 rev/min and a 356-mm (14-in.) diameter impeller was tested at rotational speeds of 705, 625, 550, 450, and 350 rev/min. Using the head and capacity data collected at 705 rev/min, generate pump curves for rotational speeds of 625, 550, 450, and 350 rev/min and compare the curves generated with the measured data points.

	SI Units		**U.S. Customary Units**	
	Flow rate (m^3/h)	Head (m)	Flow rate (gal/min)	Head (ft)
705 rev/min				
	1596	5.91	7025	19.4
	1361	7.28	5994	23.9
	1140	8.66	5020	28.4

	Flow rate (m^3/h)	Head (m)	Flow rate (gal/min)	Head (ft)
	1002	9.39	4410	30.8
	866	10.21	3550	33.5
	363	11.28	1600	37.0
	0	13.11	0	43.0
625 rev/min				
	1420	4.57	6250	15.0
	1136	6.25	5000	20.5
	908	7.32	4000	24.0
	681	7.92	3000	26.0
	454	8.63	2000	28.3
	227	9.45	1000	31.0
	0	10.36	0	34.0
550 rev/min				
	1249	3.51	5500	11.5
	943	5.12	4150	16.8
	772	5.82	3400	19.1
	454	6.55	2000	21.5
	363	6.80	1600	22.3
	0	7.92	0	26.0
450 rev/min				
	1022	2.44	4500	8.0
	795	3.35	3500	11.0
	568	3.96	2500	13.0
	227	4.72	1000	15.5
	0	5.43	0	17.8
350 rev/min				
	568	3.13	2500	7.0
	472	2.41	2080	7.9
	363	2.59	1600	8.5
	0	3.05	0	10.0

Solution: Plot the pump head-capacity (H-Q) data obtained at 705 rev/min (shown as solid circles in Figure 10-7) and draw a smooth pump curve (shown as a solid line) through the plotted points.

Develop H-Q curves for the other rotational speeds from the smooth pump curve by applying the affinity laws for flow rate and head simultaneously. For example, the flow rate and head values at the point on the 625-rev/min curve that corresponds to point a on the 705-rev/min pump curve, with coordinate values of $1250\,m^3/h$ and 8.10 m (5504 gal/min and 26.6 ft), are determined as follows.

Using Equation 10-15

$$Q_2 = Q_1/(n_1/n_2) = 1250/(705/625) = 1108\,\mathrm{m^3/h}$$

Using Equation 10-16

$$H_2 = H_1/(n_1/n_2)^2 = 8.10/(705/625)^2 = 6.37\,\mathrm{m}$$

So, point b with coordinates of $1108\,m^3/h$ and 6.37 m (4875 gal/min and 20.9 ft) on the 625-rev/min curve *corresponds* to point a on the 705-rev/min curve. Enough other *corresponding points* are computed in the same manner to allow the entire curve to be drawn for 625 rev/min. The process is repeated for curves at 550, 450, and 350 rev/min, which pass through points c, d, and e, respectively. The curves are shown as dashed lines to indicate that they are derived mathematically from the curve at 705 rev/min. The measured values for head and capacity are plotted in Figure 10-7 as open circles. The correspondence between measured values and computed pump curves is excellent.

LIVE GRAPH
Click here to view

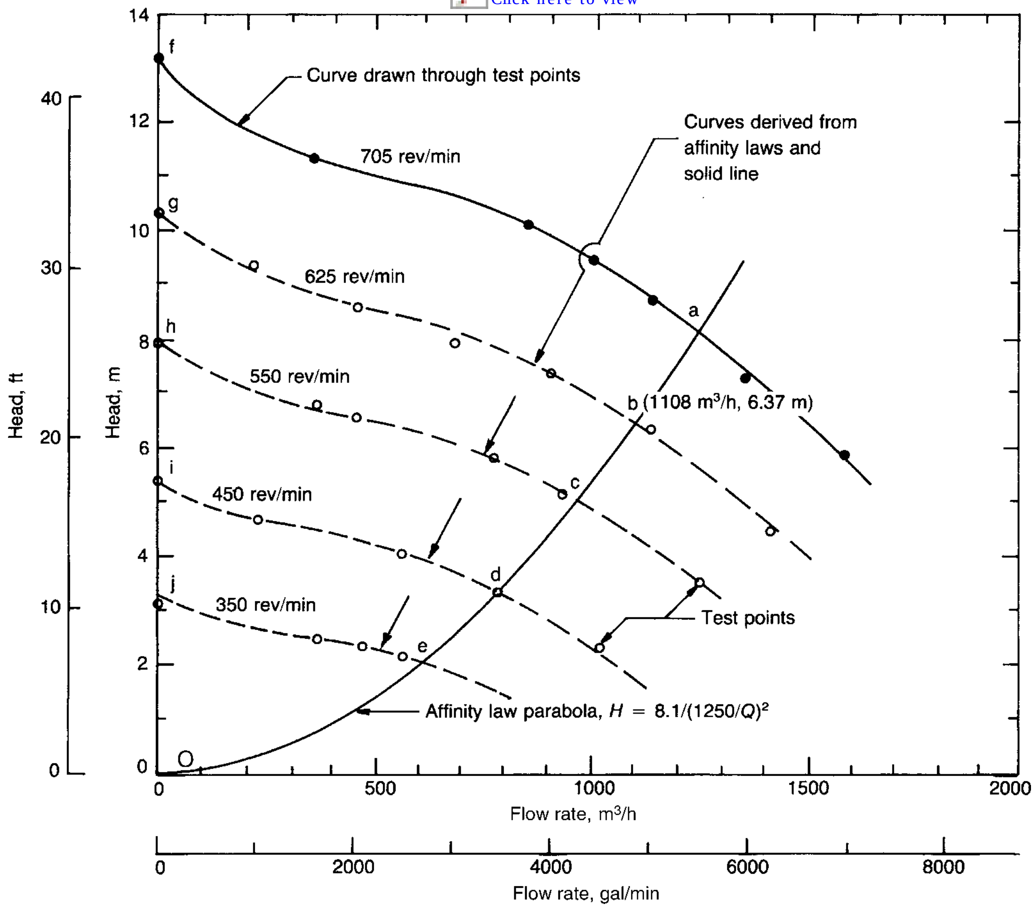


Figure 10-7. Pump curves developed from the affinity laws compared with test data.

Points f, g, h, i, and j in Figure 10-7 are unique. Because the flow is zero at these points, Equation 10-15 can be ignored and Equation 10-16 can be used alone. However, the only time that Equations 10-15 and 10-16 need not be solved simultaneously is when the flow is zero. The errors in the four points derived from point f are 0.6, 0.8, 1.7, and 5.90%, respectively. In variable-speed pumping, pumps rarely operate at less than 60% of full speed, so in practice the errors in using the affinity laws for calculating head and flow would rarely exceed 2 or 3%.

Points a, b, c, d, and e lie on a parabola that passes through the origin. The equation for the affinity parabola can be found by solving Equations 10-15 and 10-16 simultaneously to eliminate n.

$$\left(\frac{Q_1}{Q_2}\right)^2 = \left(\frac{n_1}{n_2}\right)^2 = \frac{H_1}{H_2}$$

So $H_1 = H_2\ (Q_1/Q_2)^2$, which is a parabola through point O. Hence, all *corresponding points* lie on parabolas that pass through point O, the origin. Station H-Q curves are also parabolas, but because such parabolas do not pass through the origin (unless the static head is zero), it follows that corresponding points *cannot* lie on a station H-Q curve. Expressed in the simplest terms, the foregoing statement means that *the flow from a pumping station (with a static lift) is not proportional to the speed of the pump*—a conclusion that might appear, superficially, to be at odds with Equation 10-15.

Approximate Relationships for Radial-Flow Pumps

To cover a wide range of flows with a minimum number of pump casings and impeller designs, manufacturers customarily offer a range of impeller diameters for each size of casing (see Figure 10-5). In general, radial-flow impellers are identical as cast, but the size is reduced by machining the impeller to a smaller diameter. Equations 10-12 through 10-14, written for pumps of different sizes, can be modified into equations for impellers of reduced diameter in the same pump. Dividing Equation 10-13 for an impeller of diameter D_1 by the same equation for an impeller of diameter D_2 and canceling C_H and n^2 (because density, viscosity, and rotational speed are the same) gives

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2 \tag{10-18}$$

Although the performance of many pumps can be modeled with this relationship with small errors, significant deviations can arise, because C_H may not remain constant (as assumed here) when only D is reduced without reducing all other dimensions in the same proportion, as would be required to maintain strict geometrical similarity.

The same analysis applied to Equation 10-12 gives

$$\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^3 \tag{10-19}$$

Equation 10-19 applies only to two pumps of different size (i.e., the same scale-up in three dimensions). For a single pump, the discharge area for a radial-flow impeller is a function of impeller width and volute size, and both are nearly constant even if the impeller diameter is reduced. Because area is proportional to diameter squared, Equation 10-19 can be converted to an equation for a single pump by factoring out $(D_1/D_2)^2$, which yields

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2} \tag{10-20}$$

Power is a function of head times discharge, so multiplying Equation 10-18 and Equation 10-20 gives

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^5 \tag{10-21}$$

These relationships, when used to predict the effect of changing the diameter of a radial-flow impeller, are somewhat less accurate than the affinity laws because the angle of the blade decreases slightly and the clearance between impeller and casing increases (which changes the geometry) as the diameter is reduced (see Figure 10-6). Consult the pump manufacturer before applying the modified affinity laws to more than a 5% change in impeller diameter.

It should be noted that the modified affinity laws (Equations 10-18 through 10-21) are less accurate for trimming impellers of mixed- and axial-flow pumps.

Example 10-3
Effect of Changes in Impeller Diameter and Speed

Problem: Data from Figure 10-5 for a pump with an impeller diameter of 0.4463 m (17.57 in.) operating at 1170 rev/min are tabulated as follows.

SI Units			**U.S. Customary Units**		
Point	Flow rate (m^3/h)	Head (m)	Point	Flow rate (gal/min)	Head (ft)
a	0	47.6	a	0	156
b	100	46.3	b	440	152
c	200	45.3	c	880	149
d	300	44.0	d	1320	144
e	400	41.0	e	1760	135
f	500	36.5	f	2200	120
g	600	29.8	g	2640	98

(1) Develop a new head-capacity curve for the same pump fitted with a new impeller 0.3810 m (15 in.) in diameter. (2) Compare the results with the test curve for the new impeller shown in Figure 10-5. (3) Find the increase in rotational speed that would be required for the new impeller to match the performance of the original impeller at 1170 rev/min.

Solution:

H-Q for the smaller impeller. Compute this using Equations 10-20 [$Q_2 = Q_1(D_2/D_1)$] and 10-18 [$H_2 = H_1(D_2/D_1)^2$]. Sample calculations for point d are

$Q_2 = 300\ (0.3810/0.4463) = 256\,\mathrm{m^3/h}$ $\quad Q_2 = 1320\ (15/17.57) = 1130$ gal/min

$H_2 = 44.0\ (0.3810/0.4463)^2 = 32.1\,\mathrm{m}$ $\quad H_2 = 144\ (15/17.57)^2 = 105\,\mathrm{ft}$

Compare the results to Figure 10-5. The computed values of points a to g for the new 15.00-in. impeller diameter are summarized below.

SI Units				**U.S. Customary Units**			
		Head (m)				Head (ft)	
Point	Flow rate ($\mathrm{m^3/h}$)	Computed	Test	Point	Flow rate (gal/min)	Computed	Test
a′	0	34.4	34.4	a′	0	113	113
b′	85	33.8	34.4	b′	376	111	113
c′	171	32.9	33.2	c′	751	108	109
d′	256	32.0	31.7	d′	1130	105	104
e′	341	29.9	29.0	e′	1500	98	95
f′	427	26.5	24.7	f′	1880	87	81
g′	512	21.3	—	g′	2250	70	—

Compare the computed with the tested heads in the last two columns or compare calculated points a′ to g′ in Figure 10-5 with the 15.00 curve. Minor differences (up to 0.6 m or 2 ft) are due to errors in plotting and scaling. Larger differences are due to the fact that actual losses in the pump are not considered in Equations 10-18 and 10-20.

Rotational speed for the trimmed impeller. Use Equation 10-16 ($n_2 = n_1\sqrt{H_2/H_1}$) at zero flow to estimate the new rotational speed required to obtain the original flow rate with the smaller impeller.

$n_2 = 1170\sqrt{47.6/34.4} = 1376$ rev/min $\quad n_2 = 1170\sqrt{156/113} = 1375$ rev/min

Specific Speed

Although the desired discharge or flow rate has the major impact on the size and layout of a pumping station through the sizes of the pipes and the intake and discharge bays, the geometry of each pump must also be known for establishing the detailed configuration of the station. The shapes of the impeller and adjacent stationary flow elements of a pump are determined by the "specific speed," n_s (Note: The term n_s is used generically without regard to units.), which is a combination of the pump speed, flow rate, and the total developed head. The SI value of n_s is denoted as n_g, whereas N_s is the symbol used for U.S customary units.

For a geometrically similar series of pumps operating under similar conditions, the diameter term in Equations 10-12 and 10-13a can be eliminated by dividing the square root of Equation 10-12 by the three-fourths power of Equation 10-13a:

$$n_q = \frac{C_Q^{1/2}}{C_H^{3/4}} = \frac{(Q/nD)^3}{(H_t/n^2D^2)^{3/4}} = \frac{nQ^{1/2}}{H_t^{3/4}} \qquad \text{(10-22a)}$$

where n_s is the specific speed (also called the "type number" in Europe), n is in revolutions per minute (not radians per second), Q is flow rate (strictly $\mathrm{m^3/s}$ but often $\mathrm{m^3/h}$), and H_t is the total dynamic head in m. In spite of being dimensionally incorrect, it is this form of the equation that is used in the United States. (A dimensionally correct expression would be derived by using Equation 10-13b instead of 10-13a.)

$$N_s = \frac{nQ^{1/2}}{H_t^{3/4}} \qquad \text{(10-22b)}$$

In U.S. customary units, N_s is the specific speed, n is in rev/min, Q is flow rate in gal/min, and H_t (or simply H) is the total dynamic head in ft. The relation between specific speeds for various units of flow rate and head is given in Table 10-1, wherein the numbers in bold type are those customarily used. For example, if the specific speed n_q is expressed in metric units (e.g., $Q = \mathrm{m}^3/\mathrm{h}$), the corresponding specific speed in U.S. customary units (e.g. Q = gal/min), N_s, is obtained by multiplying the metric value by 0.861.

For any pump operating at any given speed, Q and H must be taken at the point of maximum efficiency. When using Equation 10-22 for pumps with double suction impellers, one-half of the flow rate is used unless otherwise noted. However, the full flow rate is widely used in the United States. For multistage pumps, the head is the head per stage.

The pump speed, usually selected by the pump manufacturer, is based on a consideration of (1) the available speeds of the drive motor or engine; (2) critical speeds that might cause structural resonance; and (3) the limit imposed by the suction capability of the pump. The suction limit on the speed is computed from the "suction specific speed" n_{ss} as follows:

$$N = n_{ss} \times \frac{(\mathrm{NPSHA})^{3/4}}{\sqrt{Q}} \tag{10-23}$$

where N is the pump speed in rev/min, and n_{ss} is characteristic of the pump and ranges from 150 to 250 in SI units (8,000 to 13,000 in U.S. customary units), NPSHA is the available "net positive suction head" in m (SI) or ft (U.S. customary units) and is defined in Section 10-4, Q is the flow rate in m^3/s (SI units) or gal/min (U.S. customary units). Once N is known, the specific speed n_s can be computed from Equation 10-22, because Q and H are specified initially. Of course, there could be more than a single pump, and a high head requirement could dictate the need for a multistage pump or pumps in series to optimize pump efficiency and cost.

The variations in maximum efficiency to be expected with variations in size, flow rate, and specific speed are shown in Figure 10-8 for single-stage, single-entry centrifugal pumps. The progressive changes in impeller shape as the specific speed increases are shown along the bottom of Figure 10-8. The efficiency increases with the size of the pump because of a reduction in the friction losses due to the lesser relative roughness of the internal surfaces and a corresponding drop in the volumetric losses. The greatest efficiency is achieved in single-stage, single-entry pumps with a volute casing [5, 6]. In general, for a given rotational speed,

- If the n_q value is less than 30 ($N_s < 1500$), Q must be low and H must be high.
- If the n_q value is 30 to 80 (N_s = 1500 to 4000), Q and H must be of intermediate value.
- If the n_q value is more than 80 ($N_s > 4000$), Q must be high and H must be low.

In summary, the significance of specific speed is apparent from Figure 10-8. In particular, a high-n_s axial-flow pump has pipes as large as the impeller (or propeller), which feed the fluid in axially and usually conduct it away in a radial direction through a sweeping, gentle, 90-degree bend downstream from the impeller and any flow-straightening vanes. The low-n_s radial-flow pump, on the other hand, has smaller pipes and a volute collector surrounding the impeller with a tangential or radial conical diffuser exiting to

Table 10-1. Factors for Converting Values of Specific Speed Expressed in One Set of Units to the Corresponding Values in Another Set of Units. Customary units are gal/min in the United States and m^3/s in Europe. Some Europeans use m^3/h.

n_s Q H_t	rev/min L/s m	rev/min m^3/s m	rev/min $\mathrm{m}^3/\mathrm{min}$ m	rev/min m^3/h m	rev/min gal/min ft	rev/min ft^3/s ft
	1	0.0316	0.2449	1.897	1.633	0.0771
	31.623	1	7.746	60	51.628	2.437
	4.082	0.1291	1	7.746	6.665	0.3146
	0.5270	0.0167	0.1291	1	0.860	0.0406
	0.6123	0.0194	0.1500	1.162	1	0.0472
	12.975	0.4103	3.178	24.618	21.183	1

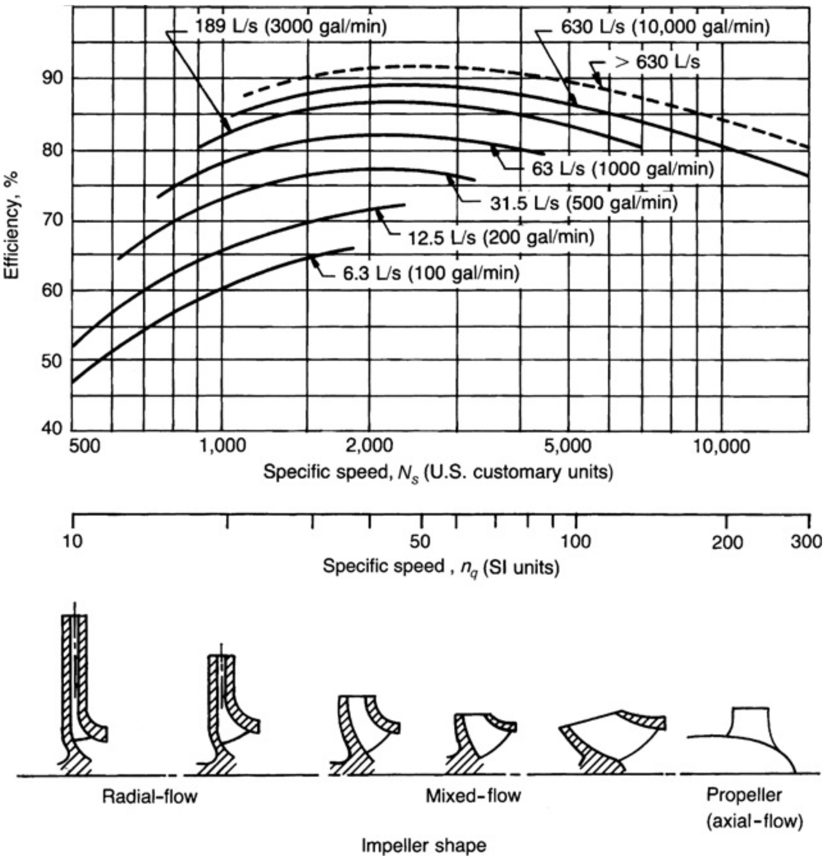


Figure 10-8. Pump efficiency as related to specific speed and discharge. After Flowserve Corp.

the downstream piping system. There are other configurations, as is apparent in Example 10-4.

10-4. Cavitation

Cavitation is the formation and collapse of vapor cavities within the pump. When cavitation occurs in pumps, it has the *potential* to cause (1) *performance degradation*, namely, loss of head, capacity, and efficiency and (2) permanent *damage* due to erosion and mechanical failure of pump components and structures. Whether these adverse effects [(1) and/or (2)] will occur—and to what extent—depends on several factors, the primary ones being these:

- The Net Positive Suction Head (NPSH) supplied at the pump inlet, (1) and (2)
- Off-Design Operation [i.e., at flow rates (Q) through the pump that are different from its BEC (design flow rate or capacity, (1) and (2)]
- The "Suction Energy" or "Energy Level" at which the pump is operated, (2) only

How these factors influence the adverse effects (1) and (2) is explained in the following subsections (see also Cooper [11]).

Example 10-4
Determination of Pump Speed and Configuration

Problem: A flow of 1 m^3/s (15,850 gal/min) must be pumped against a head of 25 m (82 ft). The available net positive suction head is 6.1 m (20 ft). Assuming a single stage and the suction specific speed n_{ss} to be 200 (10,330 in U.S. customary units), estimate the pump rotational

speed (rev/min) and discuss the corresponding pump type and configuration. Also estimate the pump efficiency.

Solution: Estimate the pump rev/min using Equation 10-23 as follows:

SI Units

$$n = 200 \times \frac{(6.1)^{3/4}}{\sqrt{1}} = 776\,\text{rev/min}$$

U.S. Customary Units

$$n = 10{,}330 \times \frac{(20)^{3/4}}{\sqrt{15{,}850}} = 776\,\text{rev/min}$$

However, if this pump is driven by an electric motor at 50 Hz (or 60 Hz), the nearest synchronous speed is 750 rev/min (or 720 rev/min); so the actual pump speed would probably be reduced accordingly from this estimate of 776 rev/min. Nevertheless, continuing with the 776-rev/min estimate, the specific speed is found as follows from Equations 10-22a and 10-22b.

$$n_q = \frac{776 \times \sqrt{1}}{(25)^{3/4}} = 69.4$$

$$N_s = \frac{776 \times \sqrt{15{,}850}}{(82)^{3/4}} = 3{,}585$$

As illustrated in Figure 10-8, this is a mixed-flow pump with a radial-discharge, overhung impeller and would achieve as much as 91% efficiency. The pump would most likely have an inlet pipe feeding axially into the impeller, a volute collector around the periphery of the impeller, and a tangential conical diffuser exiting to the downstream piping.

Alternatively (not shown in Figure 10-8), this pump could have a double-suction impeller hung between bearings with a specific speed $n_q = 49.1$ ($N_s = 2535$) per side, which is based on one-half the total flow or $0.5\,\text{m}^3/\text{s}$ (7925 gal/min). The efficiency could be as high as 92%; although one inlet passage and one volute handling both half-flows makes this pump something like the higher-n_s one above, which would not exceed 91% efficiency. The double-suction impeller would be essentially two impellers of the third type from the left in Figure 10-8 joined back-to-back and fed by a tangential inlet passage that splits symmetrically to feed flow to both sides of the impeller. The discharge pipe would come off the volute tangentially and in the same direction, making this an "in-line" pump, which is commonly used in pipelines and which is illustrated in Figures 10-2 and 10-3.

Occurrence of Cavitation and Its Effects

As fluid flows from the pump inlet port and into the impeller, it undergoes a drop in pressure because it accelerates and because the pump blades that drive the fluid have higher-than-average pressure on their driving faces and, therefore, lower-than-average pressure on their trailing faces. If the entering liquid contains dissolved air, tiny bubbles of this air come out of solution as the pressure drops. If the pump inlet absolute pressure, p_1, continues to drop and zones develop within the pump wherein the absolute pressure equals the vapor pressure, p_v, of the liquid. The liquid now boils, which causes an explosive increase in the number and size of bubbles (cavities), and the liquid's own vapor then becomes the main constituent of these cavities.

Performance Degradation and NPSH

The occurrence of cavitation on the trailing face or suction side of a pump impeller blade near its leading edge is illustrated in Figure 10-9a. The difference between p_1 and p_v is called NPSH when expressed as head and augmented by the pump inlet velocity head $V_1^2/2g$. As the NPSH becomes lower, the volume of bubbles increases and so does the likelihood that these cavities will distort the liquid flow lines as illustrated in Figure 10-9a. Sufficient distortion, in turn, causes *performance degradation*. If NPSH gets low enough, the impeller becomes filled with vapor (or vapor-locked) as shown in Figure 10-9b, and the pump can no longer generate any pressure rise or liquid head, TDH. Many of the bubbles occurring in partial cavitation collapse on the blade surfaces and create the erosion seen in Figure 10-10a and b. Such erosion is less severe in full cavitation, wherein pumping ceases.

Degradation of the performance is illustrated vividly in Figure 10-11, where three H-Q performance curves are shown, each set corresponding to a given suction lift. If the suction lift increases, the absolute pressure, p_1, at the pump suction port and the NPSH

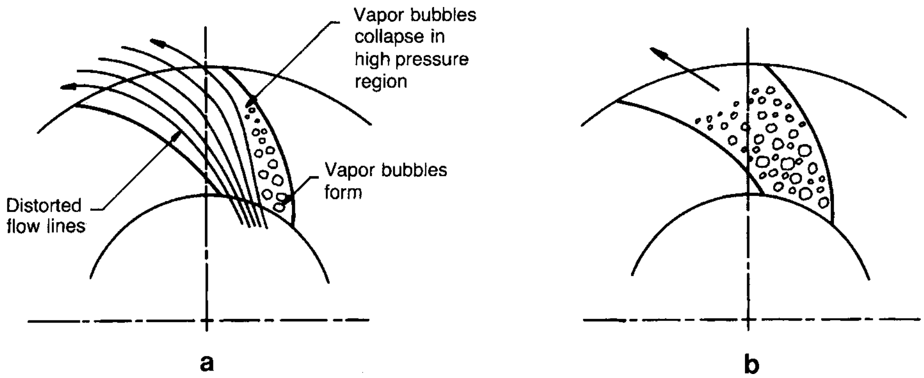


Figure 10-9. Formation of vapor bubbles in a pump impeller. (a) Partial cavitation; (b) full cavitation. After Addison [4].

decrease. It is at the lowest NPSH that the greatest degradation in performance is to be expected, and that is exactly what is portrayed in this figure. These curves are typical of centrifugal pumps of specific speeds, n_q, less than 30 ($N_s < 1500$). Here, there is a sharp drop or cutoff in the H-Q curves. Vapor bubbles begin to form at a lower flow rate than the cutoff flow rate where cavitation is fully formed, as in Figure 10-9b. For pumps with specific speeds n_q between 30 and 80 (N_s between 1500 and 4000), the pump performance curves fall more gradually until the cutoff flow rate is reached. For pumps with specific speeds n_q greater than 80 ($N_s > 4000$), there is no distinct cutoff point.

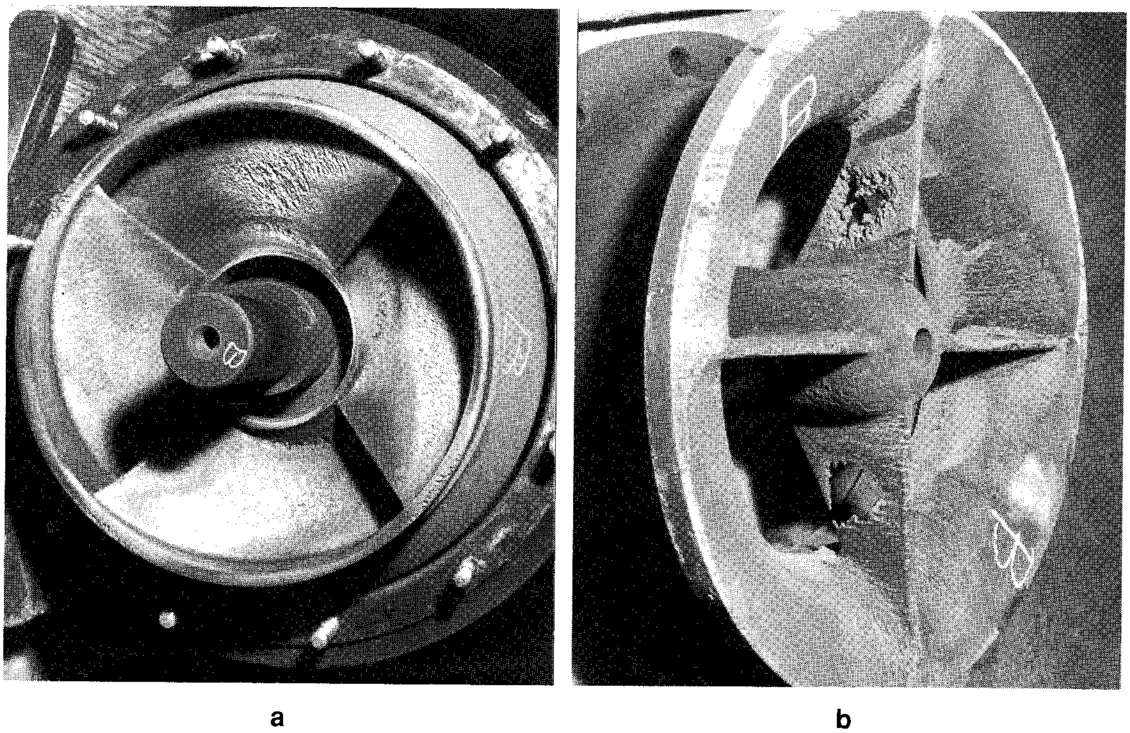


Figure 10-10. Cavitation damage in a vertical pump. (a) Impeller; (b) suction bell.

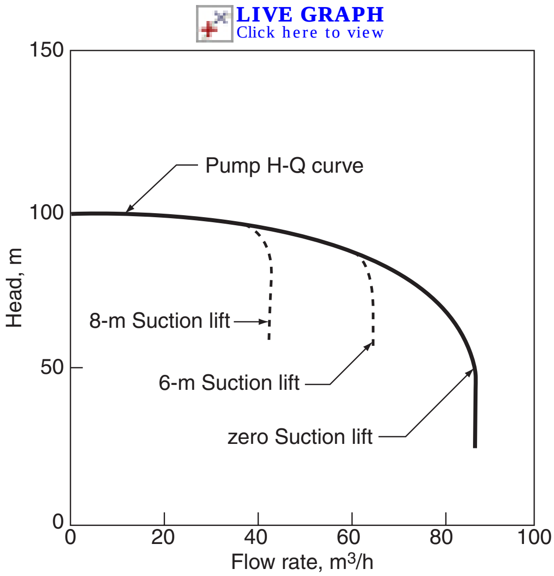


Figure 10-11. Pump performance curves at various suction lifts with cavitation. Note how rapidly the cavitation causes the head to drop.

Cavitation Damage

As the cavities illustrated in Figure 10-9a flow downstream into the higher-pressure zone of the impeller, they can no longer exist (i.e., the vapor in them must condense). This condensation occurs quite rapidly, and the bubbles collapse as soon as they are exposed to an absolute pressure, p, that exceeds the vapor pressure, p_v. The instantaneous *collapse pressure*, p_c, can be immense; in fact, it can be great and frequent enough to cause fatigue failure of the material of the blade or other surface next to which repeated collapsing of successive bubbles at the same site occurs. The resulting pitting or erosion is illustrated in Figure 10-10a, seen as expected from Figure 10-9a on the suction (visible) sides of the blades downstream of the leading edges. The damage shown in Figure 10-10b is caused by recirculation (and backward flow) in a pump that operated too far to the left of the BEP.

In addition to pitting or erosion, the collapsing vapor cavities can also cause noise and vibration. Noise is produced by the implosions of the collapse phenomenon, and the vibration is due to the imbalance and surging caused by the uneven distribution of collapsing vapor bubbles.

Erosive Damage and Suction Energy

Even though there may be extensive cavitation with many bubbles collapsing, *erosive damage does not occur* unless p_c is large enough and the material of construction weak enough. The value of p_c increases with the *suction energy* of the pump. The suction energy, in turn, can be expressed as the *inlet-tip-speed head* $u_{1,t}^2/2g$ of the pump blade tip speed, $u_{1,t}$, at the impeller inlet, namely the linear speed of the blade at its maximum diameter, as seen in Figure 10-11 and defined in Figure 10-6. Water pumps with $u_{1,t}^2/2g$ less than 30 m (100 ft) usually experience no erosion due to cavitation. The inlet-tip-speed head is about one-quarter of the head TDH of a centrifugal pump, and because the TDH is readily known, it is easier to remember that a centrifugal pump with less than 120 m (400 ft) of head should usually experience no erosion due to cavitation. However, for a mixed- or axial-flow pump, such as that shown in Figure 10-10, the TDH can be substantially less than the inlet-tip-speed head, so that it is not unreasonable to expect erosion damage at a TDH of less than 30 m (100 ft) in an axial-flow pump.

Energy Level, TDH, and Cavitation Erosion Rate

The TDH is also a measure of the *energy level* of a pump stage; and, as with suction energy, if the pump is operated above a critical energy level, cavitation erosion will occur. The TDH for this critical energy level falls with increasing specific speed, n_s, the type of impeller changing from centrifugal (or radial-outflow) to axial-flow as n_s increases, as illustrated in Figure 10-8. From the above discussion, it is apparent that high-n_s pumps exhibit more cavitation erosion at lower TDH than do low-n_s pumps [1]. (Note: As defined earlier in this chapter, the SI-value of n_s is denoted as n_q, whereas N_s is the symbol used for U.S. customary units of n_s.)

The consequences of operating a pump above the critical energy level are severe. If a pump is near the critical energy level at a given rev/min, it will experience cavitation erosion if the rev/min is increased. In a high-energy pump, the rate of penetration of cavitation erosion into the blades or other affected surfaces increases approximately as the *sixth power* of the blade inlet tip speed $u_{1,t}$, and it greatly shortens the life of the pump [1].

Off-Design Operation

If the pump is operated at flow rates different from the BEC, the locations of the cavities change. As Q is increased above the BEC, the low-pressure zones eventually switch to the opposite sides of the pump blades from those shown in Figure 10-9a, and the

resulting performance degradation for a given NPSH is greater.

When Q is much less than the BEC, the impeller is over-size for that low value of Q and it ejects highly unsteady, swirling fluid backward out of its inlet. This high-velocity, turbulent backflow is full of eddies with low-pressure centers that therefore contain bubbles. Driven by the backflow into adjacent upstream walls and stationary vanes, these bubbles collapse under the accompanying higher stagnation pressures. In Figure 10-10b, the stationary upstream guide vanes have been perforated by this violently erosive phenomenon. As the pressure increases around the bubble as it collapses, the collapse pressure, p_c, becomes greater—even greater than occurs within impellers operating at the BEP. Damage of the magnitude seen in Figure 10-10b also happens to the driving face or pressure sides of the impeller blades (the invisible sides of the impeller blades of Figure 10-10a), because these blades drive the eddying backflow just described.

Minimum Flow Limits

From the above discussion, it is evident that at low-flow, off-design conditions, intense cavitation activity accompanies the unsteady backflow. Erosion, noise, and vibration are more severe because the values of the suction energy and TDH may not be conservative enough for extended operation at low-flow, off-design conditions. These conditions have led to the specification of minimum-flow limits by pump manufacturers. As might be expected, such limits vary with energy level. The minimum flow of high-energy pumps is a considerably greater fraction of the BEC than it is for pumps of lower energy [1]. These limits are also influenced by the NPSH. For example, as described below, no cavities exist within the pump at sufficiently high NPSH, and hence, a lower minimum flow can be permitted.

Domains and Operational Criteria for NPSH

The net positive suction head, NPSH, was defined earlier in this Section 10.4 as the absolute static pressure at the pump inlet port, p_1, plus the velocity head $V_1^2/2g$ at that same location minus the vapor pressure, p_v, of the liquid. Several criteria for NPSH are used that apply to the various domains of pump operation with respect to cavitation and its effects. These domains are best described in terms of a constant-speed, constant-flow rate (discharge) test in which the pump head, TDH or H_t, is recorded as the NPSH is progressively reduced to the point of complete loss of this liquid head or pump pressure rise. The test described above is an expanded version of the pump manufacturer's "NPSH-depression" test for finding the required value of NPSH for the pump to run satisfactorily. The results of this expanded version of a typical NPSH depression test are summarized in Figure 10-12.

Difference Between Cavitation Inception and Incipient Head Reduction

In pump research laboratories, the actual cavitating fluid entering the pump blades is observed during this test through special windows with high-speed photography and strobe lights. The observer gets a view very similar to that of Figure 10-10a, except that the actual bubbly flow is also seen [1, 11]. The test begins with an extremely high value of NPSH such that no bubbles can be seen. The inlet pressure to the pump is then reduced (moving from right to left in Figure 10-12). The point at which the first bubble is either heard or observed is called the point of cavitation inception, and the NPSH value at this point is called $NPSH_i$. As indicated in Figure 10-12, no change in pump head (TDH or H_t) has occurred down to this value of NPSH. In fact, a considerable *further reduction in NPSH occurs with no head change*—down to the point where the *TDH* just begins to fall off. The value of NPSH at this point is called $NPSH_{0\%}$, because the head has fallen off by zero percent. At this point, sheet-cavities of vapor are seen trailing off the leading edges of the impeller blades if the pump is running at the BEP. At flow rates less than BEC, these sheet cavities become unsteady clouds of bubbles. Yet there has been zero reduction in head. *Until relatively recently, it was thought that the $NPSH_{0\%}$ point was the point of inception or first appearance of bubbles. Certainly, it is the point of incipient head reduction.* To avoid confusion between incipient head reduction and inception of cavity formation, however, the term $NPSH_i$ is confined here to the latter, usually much higher value of NPSH. Because $NPSH_i$ is so great, *it is unusual to find a pump whose first stage is not cavitating.*

Net Positive Suction Head Required

Obtaining repeatable results for $NPSH_{0\%}$ has proved so difficult that the idea of a measurable reduction in head that would be both repeatable and acceptable has been established at 3% of the non-cavitating

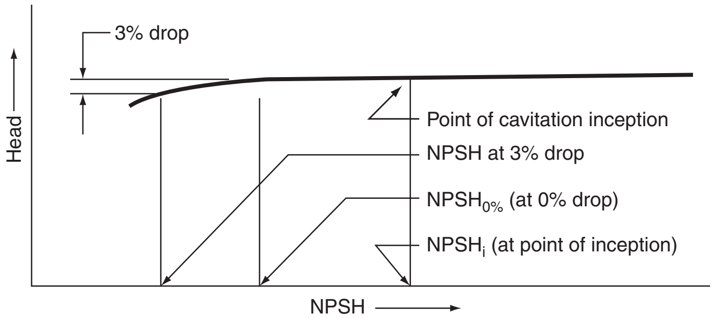


Figure 10-12. Net positive suction head criteria as determined from pump tests.

head of the pump. The value of NPSH at this point has been given the name $NPSH_{3\%}$ or NPSHR, because this is the value of NPSH that is required to maintain pump performance. At this point, the sheet cavity (or clouds) of vapor on each blade extends all the way to the leading edge of the following blade. That condition causes only 3% reduction in TDH, but as seen in Figure 10-12, any further reduction in NPSH leads to a complete breakdown in performance (i.e., total loss of pump liquid head, TDH). The NPSH at this point is sometimes called the "breakdown-NPSH."

Potential for Cavitation Damage

The potential for cavitation damage arising from erosion and vibration exists throughout the range of NPSH from $NPSH_i$ to breakdown. Vibration becomes acute if the flow rate is less than the "minimum flow" specified by the pump manufacturer. The rate of erosion is roughly described by Equation 10-24 [1, 11]:

$$P \propto B^6 (C \times N)^3 \qquad (10\text{-}24)$$

where P is rate of penetration through a pump blade or other surface, B is impeller blade inlet tip speed (see Figure 10-6), C is sheet cavity length, and N is NPSH.

So, the erosion peaks at an NPSH value somewhere between $NPSH_i$ and breakdown—usually between $NPSH_{0\%}$ and $NPSH_{3\%}$, depending on the pump design. The severity of the erosion depends on the energy level of the pump as represented by the term, B^6 in Equation 10-24.

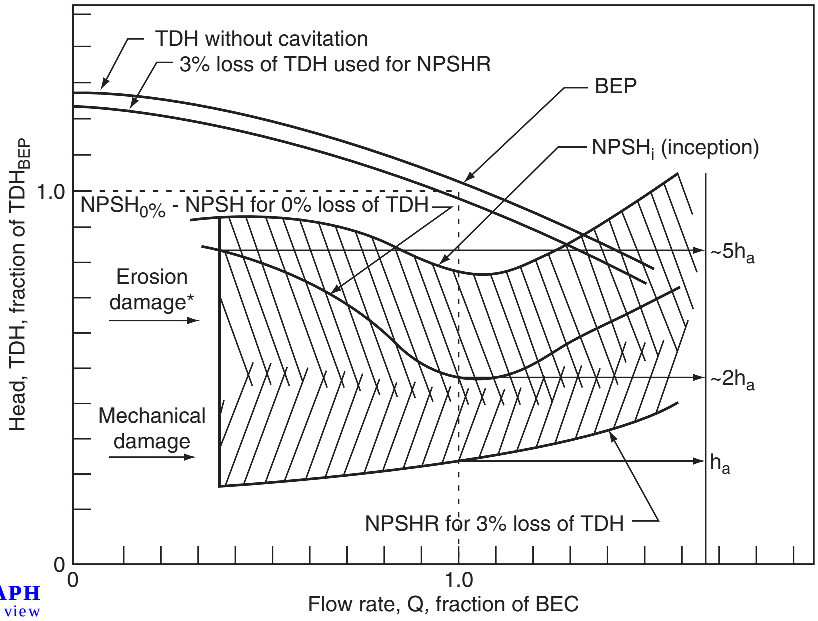
The limits of application for Equation 10-24 are within about ±25% of the BEC. In this range the cavity trailing off the leading edge of each blade appears to laboratory observers as a "sheet" of finite, measurable length C. At much lower flow rates, recirculating flow breaks this sheet into clouds of bubbles, and a distinct value of C cannot be determined. For flow rates below about 110% of the BEC, the cavity exists on the suction or trailing face of the blade. At higher flows it is on the pressure or driving face. For the typical usage (flow rate <110% BEC), the following can be said about the value of C:

- At NPSHi, C is zero.
- At $NPSH_{3\%}$ (almost at breakdown), C is essentially the distance from the leading edge of one blade to that of the next blade.

In other words, at breakdown, this sheet-like cavity stretches into the passage between two adjacent blades. Determination of C at intermediate values of NPSH in the above range is best made through actual visualization in pump development laboratories. As that is inconvenient for users, estimate C by interpolating between the two extremes in this NPSH range, or (for an approximate equation) see page 2.91 in Karassik et al. [1].

Domains versus Flow Rate (Discharge)

Applying the foregoing explanations to the entire range of flow rate of a pump at constant speed yields the curves in Figure 10-13. The potential for damage is seen as covering the range up to the *inception-NPSH* line; $NPSH_i$ is typically five or more times the value of NPSHR [11]. As the energy level increases, so does the NPSH at which the damage is significant. For example, in power-generating stations, the boiler feed pumps are often "high-energy pumps" and need a value of NPSH to be supplied that is close to this $NSPH_{0\%}$-curve to have an acceptable life in resistance to cavitation erosion and mechanical damage. The supplied value of NPSH is called the *available NPSH* or NPSHA.



LIVE GRAPH
Click here to view

Figure 10-13. Effect of NPSH on TDH and pump damage. NPSHR at BEP for 3% loss of *TDH* is termed h_a. Courtesy of DuPont Engineering [12] altered. *Note: NPSH to suppress erosion and mechanical damage increases with pump energy level or suction energy. No damage occurs if NPSH > $NPSH_i$.

Air Injection

Injection of small quantities of air into the pump inlet reduces the hydraulic shock imposed on the mechanical equipment. Although air injection may improve mechanical stability by acting as a shock absorber, there are distinct disadvantages and seldom does it permit long-term operation, because: (1) oxygen content increases corrosion, (2) gas increases the NPSHR, and (3) air causes noise and some vibration.

Taking Precautions

In conclusion, serious erosion can occur as a result of blindly accepting data from pump catalogs, because of the current standard (a 3% drop in head) used by most pump manufacturers. In critical installations where continuous duty is important or where the pump is to be operated at a reduced speed, the manufacturer should be required to furnish information on the potential for cavitation erosion, minimum flow rate, and NPSHR test results. Typically, NPSHR is plotted as a continuous curve for a pump (see Figure 10-5). When impeller trim has a significant effect on the NPSHR, several curves are plotted.

Determining the Available Net Positive Suction Head, NPSHA

The NPSH "available" (NPSHA) in the actual installation is calculated using Equation 10-25.

$$NPSHA = H_{bar} + h_s - H_{vap} - h_{fs} - \Sigma h_m - h_{vol} - FS \tag{10-25}$$

where:

H_{bar} is the barometric pressure in m or ft of water column corrected for elevation above mean sea level (see Table A-6 or A-7). Note that storms can reduce barometric pressure by 1.7%.

h_s is the static head of the intake water surface above the eye of the impeller. If the water surface is below the eye, h_s becomes a negative number.

H_{vap} is vapor pressure of the fluid at the maximum expected temperature. See Table A-8 or A-9 for the equivalent height of the water column.

h_{fs} is pipe friction in m or ft between the suction intake and the pump.

Σh_m is the sum of minor pipe friction losses such as entrance, bend, reducer, and valve losses. These "minor" losses are significant.

h_{vol} is the partial pressure of dissolved gases such as air in water (customarily ignored) or volatile organic matter in wastewater (customarily estimated to be about 0.6 m or 2 ft).

FS is a factor of safety used to account for uncertainty in hydraulic calculations and for the possibility of swirling or uneven velocity distribution in the intake.

There is no term for velocity head in Equation 10-25, because velocity head is part of the absolute dynamic head.

As shown in Figure 10-13, the NPSHA should be greater than the NPSHR by a factor of safety that depends mainly on the energy level of the pump. Although the 3% loss in TDH that defines the NPSHR is typically insignificant, the shock on the equipment can reduce the life of bearings, seals, and impellers. Surprisingly large variations in NPSHR are sometimes observed in tests of supposedly identical pumps. Small pumps are more sensitive to casting imperfections than large ones. Additionally, *when the pump operates at any point but the BEP, the NPSH may have to be increased substantially to avoid cavitation, so the need for a generous factor of safety is apparent*. In the past, it was customary to consider that 0.6 m or 2 ft (but no less than 20% of the NPSHR) was adequate, and some very knowledgeable engineers continue to recommend such safety factors for water and wastewater pumps. *More conservative engineers warn that anything less than 1.5 m (5 ft) or less than 1.35 times the NPSHR is risking damage to the pump*. Consult the pump manufacturer to decide which rules to follow, and be certain the pump manufacturer understands the application and the potential variation in operating conditions to be imposed upon the pump. Be aware that to eliminate cavitation and its effects entirely, the NPSHA must, depending on the operating flow rate relative to the BEP, be two to five times the NPSHR. The effect of NPSH on erosion and mechanical damage is illustrated in Figure 10-13. Note that NPSH to suppress erosion and mechanical damage increases with pump energy level or suction energy. It is assumed that little damage can occur if NPSHA > $NPSH_i$.

Calculating the Required Net Positive Suction Head, NPSHR

Normally the NPSHR of a pump is found from the manufacturer's tests; however, there is a body of

Example 10-5
Calculation of Net Positive Suction Head Available (*NPSHA*)

Problem: Estimate the *NPSHA* for the pump system shown in Figure 10-3. Use the data given in Example 10-1. Assume the water temperature is 40°C (104°F) and the elevation is 500 m (1600 ft).

Solution: Use Equation 10-25 with data from Appendix Tables A-6 to A-9 and calculations in Example 10-1.

SI Units

H_{bar} (Table A-6)	= +9.74
h_s (Example 10-1)	= −2.00
H_{vap} (Table A-8)	= −0.76
h_{ent}	= −0.004
h_{fs}	= −0.015
$\Sigma h_m = 0.022 + 0.018$	= −0.040
Subtotal	= +6.921
Safety factor[a]	= −1.5
NPSHA	= +5.421
Use	+5.4

U.S. Customary Units

H_{bar} (interpolate Table A-7)	= +32.0
h_s (Example 10-1)	= − 6.56
H_{vap} (interpolate Table A-8)	= −2.48
h_{ent}	= −0.014
h_{fs}	= −0.048
$\Sigma h_m = 0.072 + 0.058$	= −0.130
Subtotal	= +22.768
Safety factor[a]	= − 5.0
NPSHA	= +17.768
Use	+18

[a]Note that according to some engineers, the safety factor should be at least 1.5 m (5 ft) and also not less than 35% of NPSHR. According to others, 0.6 m (2 ft) and also not less than 10% of NPSHR is adequate for water and wastewater.

theory and empirical correlations that are used to make estimates when the test results are not available [1, 8]. The method of Rutschi [14], which is presented here, provides a correlation for Thoma's cavitation parameter, σ, the ratio of the NPSHR to the pump head TDH or H_t [8, 9, 10, 13]:

$$\sigma = \frac{NPSHR}{H_t} \tag{10-26}$$

There is disagreement in the literature as to whether NPSHR ($= NPSH_{3\%}$) or $NPSH_{0\%}$ should be used in the definition of Thoma's cavitation parameter. NPSHR is used here because, as seen in the next example, applying Rutschi's method yields an NPSH result that is only 4% greater than the manufacturer's NPSHR value. Moreover, as indicated in Figure 10-12, $NPSH_{0\%}$ tends to be more nearly twice NPSHR and is difficult to establish repeatedly.

With σ equal to the ratio $NPSHR/H_t$, it follows from the definition of specific speed, n_s (Example 10-4), and suction specific speed, n_{ss} (Equation 10-23), that $\sigma = (n_s/n_{ss})^{4/3}$. As Wislicenus [10] demonstrated, n_{ss} tends to be constant over the entire range of n_s. Thus Rutschi [14] developed a correlation for Thoma's cavitation parameter that depends on n_s. Pump efficiency is also involved, as low pump efficiency means greater internal losses and pressure drops and therefore greater NPSHR (lower n_{ss}). Plotted in Figure 10-14, Rutschi's relationship is as follows:

$$\sigma = \frac{K(n_q \text{ or } N_s)^{4/3}}{10^6} \tag{10-27}$$

where the values of K are given in Table 10-2.

The accuracy implied by the results of Example 10-6 may be misleading. First, the data points in Figure 10-14 deviate by as much as 25% from the line of best fit. Second, the K coefficients in Table 10-2 vary more widely than do the efficiencies. (A change of 30% in efficiency reduces K by half.) Finally, the efficiency of pumps (upon which K is so dependent) can easily vary by 2%. Although the calculations cannot be trusted to yield errors of less than 30 or 40%, they nevertheless provide both insight and approximate results.

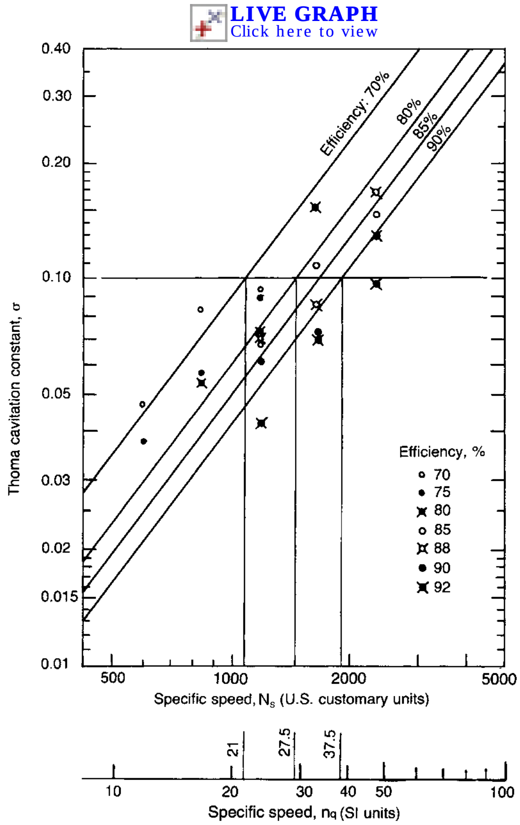


Figure 10-14. Cavitation constant, σ, versus specific speed of pumps. After Rutschi [14].

Table 10-2. Values of K for Equation 10-27[a]

Pump efficiency (%)	SI units[b]	U.S. customary units[c]
70	1726	8.99
80	1210	6.30
90	796	4.14

[a]For double suction pumps, use the same formula with Q equal to half of the actual value.

[b]Use with n_q values in m^3/s, m, and rev/min.

[c]Use with N_s values in gal/min, ft, and rev/min.

Predicting NPSHR at Off-BEP Flow Rates

If the pump operates at low head at a flow rate considerably greater than the BEC, Equation 10-28 is approximately correct:

$$\frac{NPSHR \text{ at operating point}}{NPSHR \text{ at BEP}} = \left(\frac{Q \text{ at operating point}}{Q \text{ at BEP}}\right)^n \tag{10-28}$$

Example 10-6
Determination of NPSHR and NPSHA

Problem: At a total dynamic head of 36.5 m (120 ft) and a rotational speed of 1170 rev/min, the pump described in Figure 10-5 delivers a flow of 500 m^3/h (2200 gal/min). Assume that the pump is to operate at sea level at a temperature of 15°C (59°F) and that the suction losses are 2.0 m (6.6 ft).

(1) Estimate the required NPSHR, (2) compare the computed NPSHR to the pump manufacturer's NPSHR value given in the figure, and (3) estimate the allowable suction head.

Solution:

(1) *NPSHR*. Equate Equations 10-26 and 10-27 to

$$NPSHR = \frac{H_t K (n_q \text{ or } N_s)^{4/3}}{10^6}$$

and calculate the specific speed using Equations 10-22a and 10-22b. Because the pump curves shown in Figure 10-5 are for a split-case pump with a double-suction impeller, one-half of the flow rate is used in computing n_s. Use K values of 1140 and 6.0 at 82% efficiency (see Table 10-2).

SI Units

$$n_q = 1170\left(\frac{250}{3600}\right)^{1/2}(36.5)^{-3/4}$$

$$= 20.8$$

$$NPSHR = \frac{36.5 \times 1140\ (20.8)^{4/3}}{10^6} = 2.38\,\text{m}$$

U.S. Customary Units

$$N_s = 1170(1100)^{1/2}(120)^{-3/4}$$

$$= 1070$$

$$NPSHR = \frac{120 \times 6.0\ (1070)^{4/3}}{10^6} = 7.88\,\text{ft}$$

(2) *Compare NPSHR to the pump manufacturer's NPSHR*. The computed value is 4% greater than the value of 2.28 m (7.4 ft) given in Figure 10-5.

(3) *Estimate the allowable suction head*. Rewriting Equation 10-25 in terms of the suction head yields

$$h_s = NPSHA - H_{\text{bar}} + H_{\text{vap}} + h_{\text{vol}} + h_{\text{fs}} + \Sigma h_{\text{m}} + FS$$

where, for this example, h_{vol} is assumed to be zero, and $h_{\text{fs}} + \Sigma h_{\text{m}}$ are given as 2 m (6.6 ft). Substituting the more conservative value of NPSHR (the computed value in this example) for NPSHA, the value of h_s is

$$h_s = 2.38 - 10.33 + 0.17 + 2.0 + 1.5$$

$$= -4.28\,\text{m}$$

$$h_s = 7.88 - 33.9 + 0.58 + 6.6 + 5.0$$

$$= -13.8\,\text{ft}$$

where the exponent n varies from 1.25 to 3.0, depending on the design of the impeller. In most water and wastewater pumps, n lies between 1.8 and 2.8. The NPSHR at the BEP increases with the specific speed of the pumps. For high-head pumps, it may be necessary either to limit the speed to obtain the adequate NPSH at the operating point or to lower the elevation of the pump with respect to the free water surface on the suction side.

Prevention and Control of Cavitation

The adverse effects of cavitation can be avoided most easily by simply raising the NPSHA sufficiently to ensure that no vaporization takes place within the pump. For most pumps this increase would be far more than necessary; and as NPSHA would then have to exceed the inception-NPSH or $NPSH_i$, this figure is so high as to be as much head as many

pumps generate. Furthermore, as discussed previously, for sufficiently low pump head the energy level is so low that the cavitation does little or no damage. *The real requirement is for just enough NPSH*—as suggested by the explanation accompanying Figure 10-13—to ensure adequate pump life against cavitation damage for the energy level of the pump involved. A low-energy pump [e.g., a centrifugal pump with a head of less than 120 m (400 ft)] can theoretically get along indefinitely with NPSHA being no greater than NPSHR. To guard against uncertainties in NPSHA and/or NPSHR and a possible loss of performance, a factor of safety in the form of some added NPSHA is good practice.

Preventing both performance degradation and damage, therefore, begins with (1) raising NPSHA, (2) reducing NPSHR, and (3) changing the pump design. Some ways of accomplishing these prevention approaches are:

Raising NPSHA

- Decrease the suction lift (see Figure 10-3) or increase the suction head, h_s (see Figure 10-2).
- Decrease the suction losses (e.g., with larger pipes, more gradual turns, etc.).
- Where possible, lower the vapor pressure.

Reducing the NPSHR

- Reduce the impeller rotating speed (rev/min)—possible if the pump is too large and the flow rate is being throttled at the current speed to drop both head and flow rate to those required.
- Avoid operation at or near the manufacturer's specified "minimum flow."
- Inject air at the inlet of the pump. However, as explained above, that may be of doubtful value.
- Decrease the maximum flow rate (which reduces $NPSH_i$) to prevent cavitation.
- Use a variable-speed drive that reduces rev/min when possible (e.g., when throttling flow rate, less pump head is needed so the speed can be reduced). Also, because the BEC (BEP flow rate) decreases directly with rev/min, a lower speed brings BEC closer to the actual, lower Q.
- Change the pump or the impeller to one designed for a lower NPSHR, although such a change is often accomplished by increasing the impeller inlet diameter or blade inlet tip diameter. Such an increase also raises the suction energy and could cause more damage at much lower flow rates should the pump be required to operate at such reduced flows.
- Add a booster pump or an inducer upstream from the main pump impeller. Illustrated in Figure 10-15, an inducer is an auxiliary, low-head, axial-flow propeller mounted just upstream from the impeller that supplies some of the NPSH required by the impeller. An inducer differs from a typical propeller in that it has long, tightly wrapped helical blades (usually two or three) that require much less NPSHR than does an impeller. The inducer has a very high suction-specific-speed (n_{ss}) capability—n_{ss} (see Equation 10-22a) being from 300 to 600 in SI units of rev/min, m^3/s, and m (or 15,000 to 30,000 in U.S. customary units of rev/min, gal/min, and ft) [1, 9, 13]. Inducers normally run with long cavities on the suction sides of their blades, and it is the unique behavior of these cavities that is connected with the high n_{ss} capability of inducers. The difficulty with inducers is that the range of flow rate Q is quite limited unless the suction energy is quite low. In particular, for Q lower than the low end of this range, the cavities evolve into surging clouds of vapor, backflowing at unequal rates and so imposing fluctuating loads (both radial and axial) on the inducer, coupled with excessive erosion. The resulting wear would require replacement of the inducer, which may be preferred to replacing the more expensive impeller. However, the violence of the surging might first cause mechanical failure of the whole machine, especially if it is a high-energy pump. Passive methods of eliminating this off-design "cavitation surge" have been devised but are not widely available [1].
- Take advantage of the favorable vaporization characteristics of some liquids. Compared with what occurs when cold water vaporizes, the volume

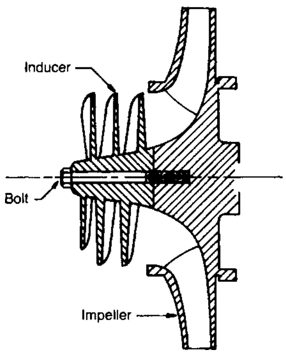


Figure 10-15. Typical inducer (axial-flow impeller) attached to a conventional pump impeller. After Turton [9].

of the cavities formed within the pump is far less for high-temperature water. Reductions in NPSHR below that specified by the pump manufacturer for room-temperature water are possible for high-temperature water [1, 8].

Changing the Pump Design

- Specify a material for the impeller (and adjacent vanes—Figure 10-11—and other parts exposed to the collapsing cavities) that is more resistant to cavitation erosion, such as stainless steel, titanium, etc. [1]. See Table 10-3.
- Utilize an axial inlet nozzle that is free of elbows or other shapes that change the flow direction appreciably, as this would eliminate distortion of the flow entering the impeller eye, which subjects the blades to varying angles of attack or "incidence" of the incoming fluid, the larger incidence on the blades creating larger cavities.
- Redesign the impeller blades to minimize the local reductions of pressure that produce cavities and thus reduce the inception NPSH of the pump ($NPSH_i$ in Figures 10-12 and 10-13). NPSHR is not reduced, but if $NPSH_i$ is reduced the damaging action of bubbles at NPSH > NPSHR is reduced or eliminated [1, 11]. This design approach is a relatively recent development found in some high-energy pumps.

10-5. Pump Characteristic Curves

In most pump curves, the total dynamic head H in m (ft), the efficiency E in percent, and the power input P in kW (hp) are plotted as ordinates against the flow rate Q in m^3/s or m^3/h (gal/min or Mgal/d) as the abscissa.

Nondimensional Pump Characteristic Curves

Nondimensional pump curves are obtained by expressing head, flow rate, power input, and efficiency as percentages of the corresponding values at the best efficiency point, and such curves are useful for (1) comparing the hydraulic properties of pumps belonging to the same type, and (2) assessing the performance of pumps of different specific speeds for various applications [6]. The general shape of these curves varies with the specific speed, the form and number of blades on the impeller, and the form of the casing used. As shown in Figure 10-16, the slope of the head-capacity curve becomes steeper as the specific speed increases. Notice the shape of the power curves in Figure 10-16a. As head decreases below the BEP, the power required increases. Thus, overloading the motor is most likely to occur in pumps with low specific speeds and with flat head-capacity (H-Q) curves. Because of the shape of the characteristic curves in Figure 10-16, mixed-flow and axial-flow

Table 10-3. Relative Resistance to Cavitation and Costs for Various Metals. Courtesy of Fairbanks Morse Pump Co.

		Impeller cost[a], $	
Material	Relative cavitation resistance	Original	Replacement
Cast iron	1	1450	3000
Cast iron with 3% Ni	1+	2030	4200
Tin bronze (gun metal)	1.2	5800	12,000
Aluminum	1.7	4350	9000
Carbon steel	3.4	2900	6000
Manganese bronze	4.5	5800	12000
Ni-resist	4.6	8700	13,000
Monel	6.8	10,150	21,000
CB7 Stainless steel	8.8	5800	12,000
CF8M ss	14	4800	9900
Titanium	18	29,000	60,000
Aluminum bronze	28	5800	12,000
CA15 ss 410BHN	35	4800	9900

[a]Approximate cost in 2004 for a 75-kW (100-hp) wastewater pump impeller about 400 mm (16 in.) in diameter.

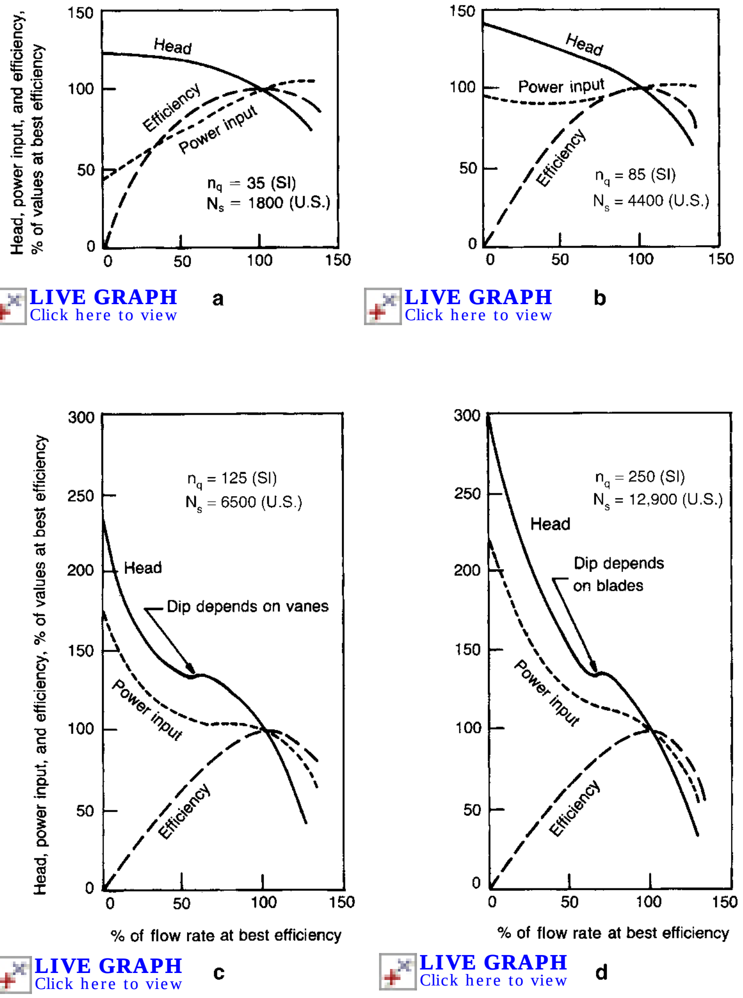


Figure 10-16. Typical dimensionless characteristic curves for centrifugal pumps. (a) Radial flow; (b) mixed flow; (c) mixed flow; (d) axial flow.

pumps are non-overloading in their operating range; that is, the power required by the pump does not increase as the flow rate increases above the BEC. However, depending on the impeller blade design, a dip (and consequent instability) can occur as shown in H-Q curves for pumps of high specific speeds. Prolonged operation of high–specific speed pumps in this head-flow rate range should be avoided.

As the specific speed increases, the effects on the power input are marked. With radial-flow pumps, where the specific speed, n_q, is about 35 (N_s about 1800), the energy input decreases with a decrease in flow. For a mixed-flow pump with a specific speed of about 85 (N_s about 4300), the power input is nearly constant. In pumps with specific speeds, n_q, greater than about 125 (N_s greater than 6400), the power adsorbed by the fluid rises steeply as the flow rate is decreased to zero. Thus, with mixed- and axial-flow pumps, some means of unloading the pump (such as a bypass pipe with a pressure-activated valve) is needed as the head rises so as to avoid overloading the pump driver. Sometimes, it is more cost effective to oversize the motor than to install an unloading device. A comparative cost analysis should be made if there is doubt.

Stable and Unstable Pump Curves

The head-flow rate curves for radial-flow centrifugal pumps can be either stable (Figure 10-17a) or unstable (Figure 10-17b), whereas the head-flow rate

curves for mixed-flow and axial-flow pumps are always stable (except for a small area of operation). Unstable pump curves are usually limited to specific speeds, n_q, of less than 20 ($N_s < 1000$). With stable pump curves, there is only one flow rate for each value of head, whereas two flow rates are possible for a given value of head in unstable pump curves, as can be seen from Figure 10-17b.

The use of pumps with unstable pump curves can lead to pumping instabilities in systems with significant capacitance or where pumps are operated in parallel. In such situations, a static and dynamic

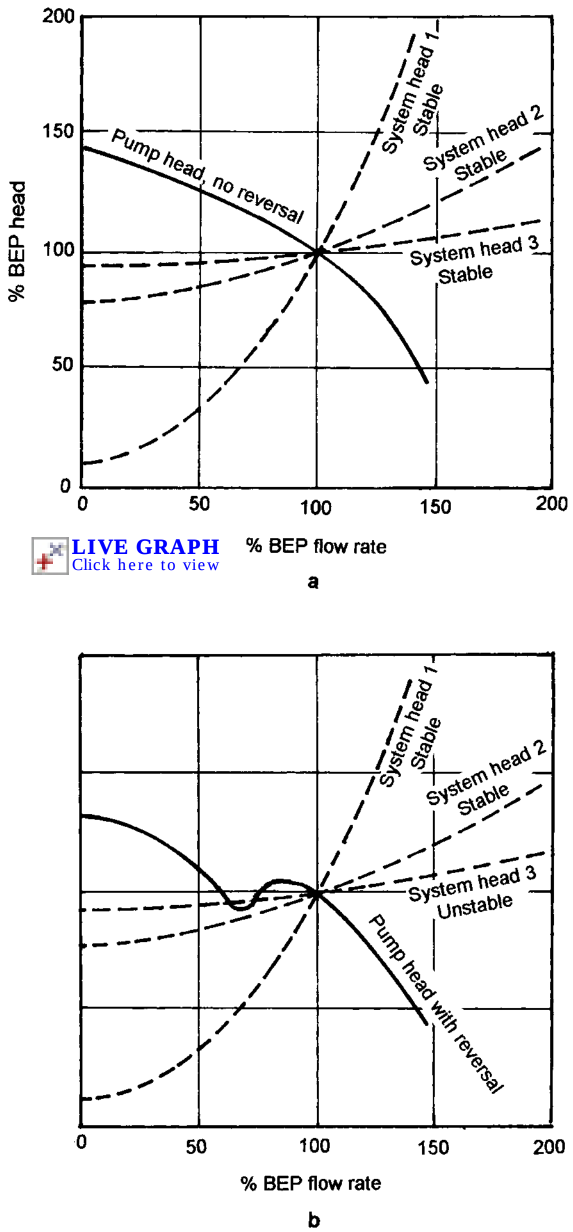


Figure 10-17. Typical pump H-Q curves. (a) Stable and (b) unstable.

analysis of the entire pumping system should be conducted to ascertain whether the system is stable under all conditions. If this analysis cannot be done or gives doubtful results, it is best to avoid the use of pumps with unstable H-Q curves entirely. In single-stage pumps, stable pump head-capacity curves can be obtained by reducing the number of impeller blades, changing the exit blade angle, and changing the configuration of the blades [6].

10-6. Pump Operating Regions

A pump operates best at its BEP. Not only is the efficiency maximum, but radial loads on the impeller and the problems of cavitation are minimized. In 1997, the Hydraulic Institute issued ANSI/HI 9.6.3-1997, American National Standard for Centrifugal and Vertical Pumps Allowable Operating Region. This standard provides guidance on how to specify pumping equipment to achieve optimum life from the equipment. The document establishes two operating regions, the Preferred Operating Region (POR), defined as the region where "... [T]he service life of the pump will not be significantly affected by hydraulic loads, vibration or flow separation." According to the document, for most centrifugal pumps, this region is nominally bounded between 70 and 120% of BEP flow rate, BEC. For vertical (column-type) pumps, the POR is the same as defined for centrifugal pumps for specific speeds, n_q,less than 87 ($N_s = 4500$) and between 80 and 115% of BEP for higher specific speed pumps. The standard explains that operation outside these limits may be possible, but at the risk of increased vibration, damaging hydraulic forces, cavitation effects, and reduced service life. This latter region, which is to be defined by the individual pump manufacturer, is termed in the standard as the Acceptable Operating Region (AOR). Many manufacturers are now indicating the AOR in their

Example 10-7
Estimating the Size of the Pump Required

Problem: Water is to be pumped at a flow rate of 0.2 m^3/s (3166 gal/min) to a height of 30 m (98.4 ft) at a rotational speed of 1150 rev/min. Determine (1) the type of pump, and (2) the impeller diameter.
Solution:

(1) *Type of pump*. Calculate the specific speed using Equations 10-22a and 10-22b and relate the type of pump to the specific speed.

SI Units

$$n_q = 1150(0.2)^{1/2}(30)^{-3/4} = 40.1$$

U.S. Customary Units

$$N_s = 1150(3.166)^{1/2}(98.4)^{-3/4} = 2071$$

From Figure 10-8, a pump with a mixed-flow (e.g., screw or helicoidal) type of impeller is needed.

(2) *Impeller diameter*. Referring to Figure 10-16, the normalized shut-off head for a pump with a specific speed, n_q of about 40 (N_s about 2000) is estimated to be about 1.4 times the value at the BEP. At shut-off head, the head developed by a pump is proportional to $u^2/2g$, where u is the peripheral speed of the impeller (see Figure 10-6). Assuming that the head at the BEP is 30 m (98.4 ft), the shut-off head is

$$\begin{aligned} H &= u_2^2/2g = 1.4 \times 30 = 42 \text{ m} \\ u &= \sqrt{42 \times 2 \times 9.81} = 28.7 \text{ m/s} \\ u_2 &= \pi D n/60 \\ D &= 60u/\pi n = 60 \times 28.7/(3.14 \times 150) \\ &= 0.477\,\text{m impeller diameter} \end{aligned}$$

$$\begin{aligned} H &= u_2^2/2g = 1.4 \times 98.4 = 137.8 \text{ ft} \\ u &= \sqrt{137.8 \times 2 \times 32.2} = 94.2\,\text{ft/s} \\ u_2 &= \pi D n/60 \\ D &= 60u/\pi n = 60 \times 94.2/(3.14 \times 150) \\ &= 1.56\,\text{ft} = 18.8\,\text{in. impeller diameter} \end{aligned}$$

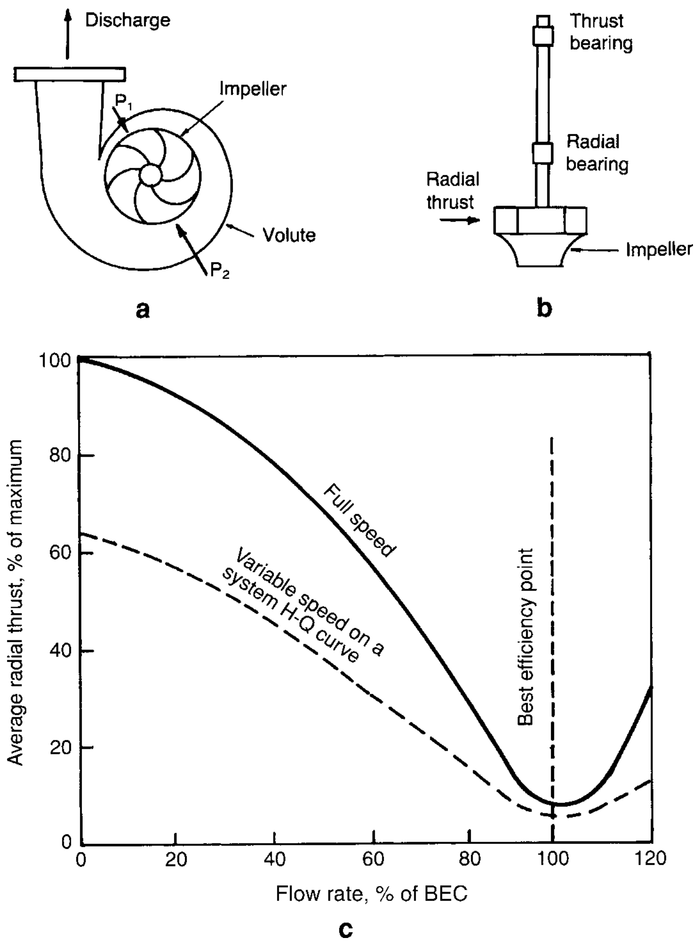
catalogs, but only a few to date have also displayed the POR. The standard should be used by the specifying party to define the requirements for pump selections to individual applications. The following rules are suggested:

- Routine, day-in, day-out (i.e., daily water demand, or dry weather flow) operating conditions should be located in the pump's POR
- Peak operating conditions (i.e., peak fire flow, peak wet weather flow) can be located in the pump's AOR because these infrequently experienced episodes will not result in significant equipment deterioration.

The advent of this ANSI/HI standard represents a departure in the process of promoting the capabilities of and specifying pumping equipment. If pump manufacturers follow through with the standard and provide POR and AOR limitations within their literature and catalogs, the users will have a greater appreciation for the capabilities and limitations of individual equipment selections and will be able to judge more readily the relative merits of competing designs.

Radial Thrust

Pump volutes are so designed that at optimum flow rate (at the BEP) the average pressure around the OD of the impeller approaches uniformity. At other flow rates, the pressures around the OD of the impeller vary considerably. The unbalanced radial forces are represented as the radial thrust, $P_2 - P_1$, in Figure 10-18a, where P is the pressure times the projected area of the impeller. Radial thrust always occurs in



LIVE GRAPH
Click here to view

Figure 10-18. Development of radial thrust in a radial-flow centrifugal pump shaft. (a) Plan of pump; (b) impeller and shaft; (c) radial thrust versus flow rate.

centrifugal pumps, but it is of greatest concern in end-suction overhung shaft designs as shown in Figure 10-18b.

The magnitude of radial thrust depends on such operating conditions as flow rate, head, and rotational speed. Radial thrust can be astonishingly large. For example, tests of a 350-mm (14-in.) pump rated at 0.5 m^3/s (7800 gal/min) at 50-m (163-ft) head showed that the radial thrust at shut-off was 36 kN (8200 lb) at 1160 rev/min and 24 kN (5400 lb) at a variable speed when operated along the station's system curve (see Figure 10-18c).

Be aware of the potential for damage by radial loads that may be too great for the load-carrying capacity of standard shafts and bearings. Pumps have occasionally been destroyed in a few days or even a few hours through excessive wear in bearings, seals, shaft sleeves, and wearing rings. Bearings, shafts, and casings must be designed to resist the anticipated radial loads. Radial loads can be approximated by the method given in ANSI/HI 1.3-2000. Standard pumps can often be customized with larger shafts and bearings. One practical means of dealing with radial thrust is to define several critical operating points and require the manufacturer to furnish a pump with a specified life at such points (see Appendix C).

Cavitation

When the pump flow rate increases beyond the BEC, the absolute pressure at the pump inlet required to prevent the possible adverse effects of cavitation increases so that cavitation is a potential problem if there is insufficient NPSH margin. The NPSHR curves are the basis for selecting the maximum permissible flow rate, but the engineer must be aware that the amount of margin, in terms of percentage of NPSHR, increases dramatically as flow rate exceeds approximately 120% of BEC or falls below 85% of BEC. When the pump flow rate decreases toward zero at the shut-off head, the radial load increases and the recirculation of the pumped fluid within the impeller becomes a problem. This recirculation can cause vibration and could combine with cavitation to ruin the impeller (see Section 10-4). In wastewater pumps, recirculation can result in plugging caused by recirculated flow tumbling rags and stringy material in the pumped flow.

Pumps should not be operated at or near the shut-off head for extended periods of time because such operation also leads to heating of the liquid and can cause severe wearing-ring rub and other problems.

Fixed Efficiency Loss

For a given impeller diameter, the operating range of a pump can be established by (1) setting a limit on the minimum acceptable efficiency, and (2) setting upper and lower limits on the allowable speed changes. Alternatively, if the diameter of the pump impeller is to be changed, the operating range of the pump can be established by (1) setting a limit on the minimum acceptable efficiency, and (2) setting upper and lower limits on the allowable impeller diameters.

The operating range of a pump based on these criteria is illustrated in Figure 10-19.

Pump Operating Range

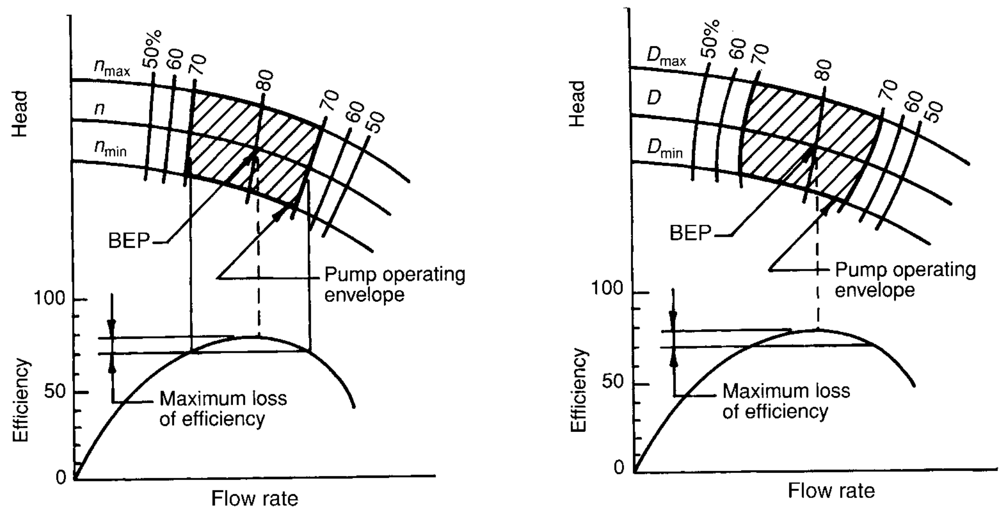
An alternative approach used to establish the pump operating range is to set limits on the pump flow rate as a percentage of the BEC. An operating range of 60 to 125% of the BEC is often recommended for pumps with specific speeds, n_q, less than 40($N_s < 2000$) [15] (see ANSI/HI 9.6.3). Pump operating envelopes are defined using these values, as illustrated in Figure 10-20, for changes in speed as well as for changes in the impeller diameter.

10-7. Elementary Pump System Analysis

In any pump–force main system, the head developed by the pump must equal the total dynamic headloss in the system and, of course, the e flow rate in pump and pipe are equal. This relationship determines the "pump operating point," a single point on both the H-Q curve of the pump and the H-Q curve of the piping system. There are several ways of finding the pump operating point, including computer analysis and several graphical methods, but the simplest (for simple problems) and the most universally used method is given in this section.

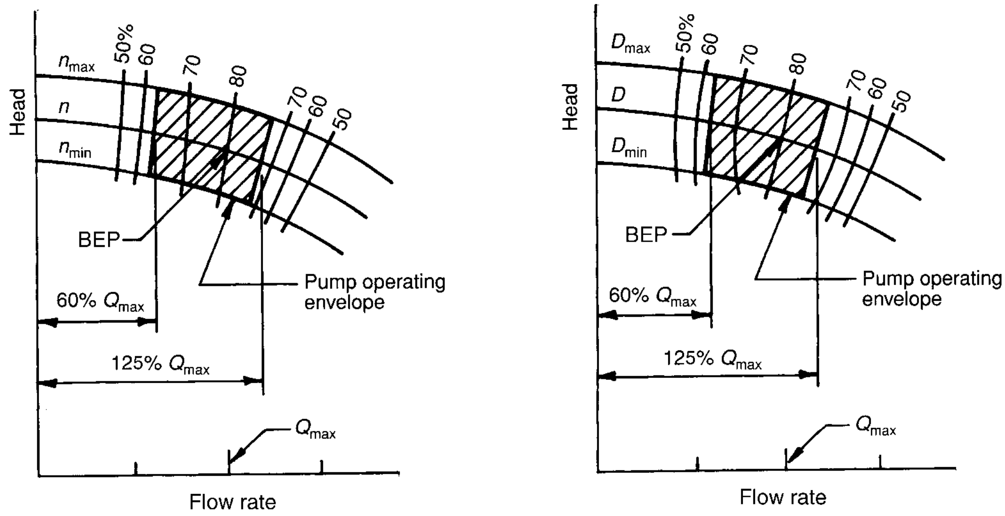
System Head-Flow Rate (H-Q) Curves

The transmission pipeline H-Q curves in Figure 10-21 are plotted over a range of flows from zero (where the only head is due to static lift) to the maximum expected (which also includes all friction, fitting, and valve losses). Because transmission pipeline or force main head-capacity curves are approximate functions of $v^2/2g$, the curve is approximately a parabola with its apex on the zero Q line. Actually,



a b

Figure 10-19. Pump operating envelopes based on a fixed efficiency loss. (a) Change of rotational speed; (b) change of impeller diameter.



a b

Figure 10-20. Pump operating envelopes based on the percentage of flow rate at the BEP. (a) Rotational speed; (b) impeller diameter.

headloss is a function of $Q^{1.85}$—not Q^2—in the Hazen–Williams equation. In the Darcy–Weisbach equation, the value of the friction factor, f, changes with the Reynolds number. However, the use of a parabola is close enough for practical purposes.

New pipe is expected to be very smooth for some indeterminate time. But bacterial slime, other deposits, or deterioration may reduce the H–W coefficient C to as little as 120 for pipes lined with cement-mortar (see Table B-5 and Section 3-2). Under rare conditions, the coefficient C may fall to 100 or less.

A simple envelope of transmission pipeline H-Q curves is shown in Figure 10-21.

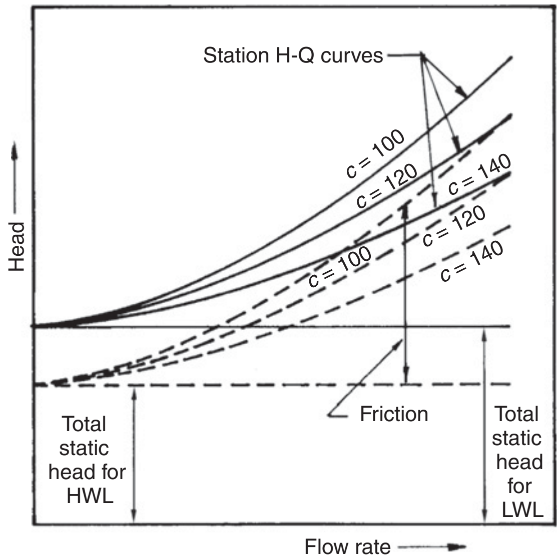
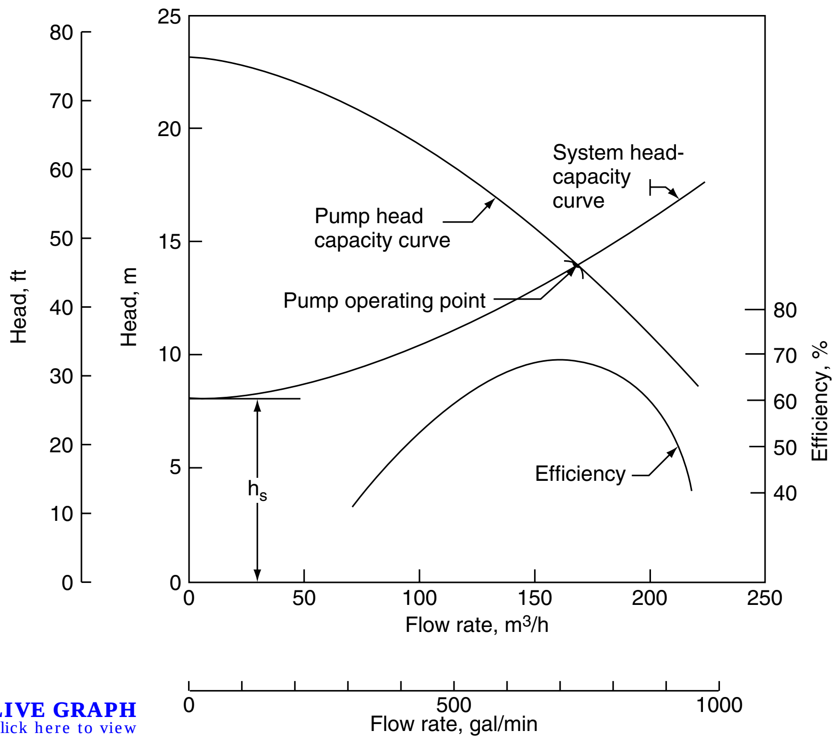


Figure 10-21. Typical envelope of transmission pipeline head-capacity (H-Q) curves.

Single-Pump, Single-Speed Operation

The point on any specific system head-flow rate curve at which a single-speed pump must operate is determined by superimposing the pump H-Q curve on the transmission pipeline H-Q curve as shown in Figure 10-22. The point of intersection is the pump operating point. As the pipe ages and the pipeline H-Q curve rises, the operating point moves up and to the left along the pump H-Q curve.

As pumps start and stop (or as they change speed) because of the change in water level, the path of operating points is extended still further along the pump curve. Pumps and impellers must be chosen for a range of operating conditions. The efficiency curve is usually rather broad at the top. Hence, choose an impeller that operates from slightly left to slightly right of the BEP. If the range in operating points is too great to fit the efficiency curve nicely, choose an impeller that fits in the early years when the pipe is smooth. Then a new impeller of larger diameter can be installed when conditions warrant it. Pay attention to the motor sizing.



LIVE GRAPH
Click here to view

Figure 10-22. Determining the operating point for a single-speed pump with a fixed value of h_s.

Single-Pump, Variable-Speed Operation

The point on any head-flow rate curve at which a variable-speed pump must operate is determined as described previously for the single-speed pump. The affinity laws are used, as shown in Example 10-8, to determine the rotational speed at any other desired operating point on the system curve. Again, the operating point moves along the pump H-Q curve, although not so far, because the change between HWL and LWL is less than with single-speed operation.

Example 10-8
Application of a Variable-Speed Pump

Problem: The pump curve shown in Figure 10-23 is for a pump operated at 1150 rev/min that pumps 130 m^3/h (572 gal/min) at a head of 13.2 m (43.3 ft), which is shown as point a. If the pump must discharge 150 m^3/h (660 gal/min) at a head of 15.5 m (50.8 ft), which is shown as point b, what is (1) the required new operating speed, and (2) the efficiency of the new operating point?

Solution: (1) *Operating speed.* Extend the system H-Q curve from point a to an intersection with the desired flow rate at point b. The new pump curve must pass through point b. Note that Equation 10-15 ($Q_1/Q_2 = n_1/n_2$) cannot be used to find the pump speed because points a

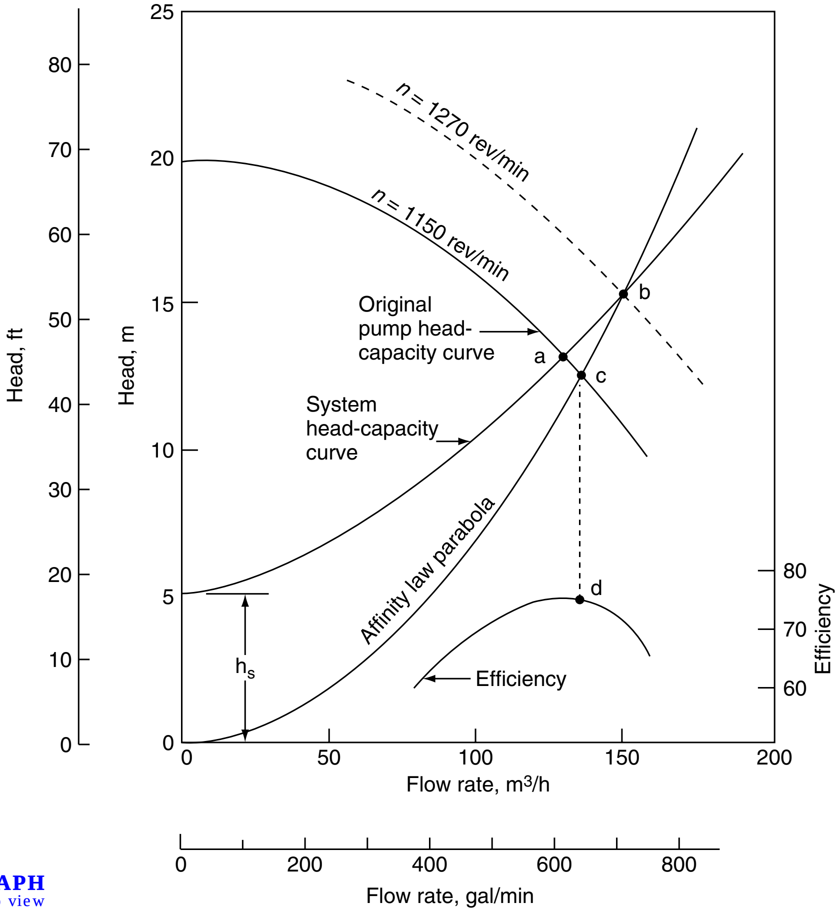


Figure 10-23. Determining the pump operating points for a single variable-speed pump and a system curve with a fixed value of static lift (h_s).

and b are not corresponding points (see Example 10-2 and the text following Equation 10-17). One way to solve the problem is to create an "affinity law parabola" by solving Equations 10-15 and 10-16 simultaneously to eliminate n:

$$\frac{Q_1^2}{Q_2} = \frac{n_1^2}{n_2} = \frac{H_1}{H_2}, \quad H_1 = H_2(Q_1/Q_2)^2$$

which defines a parabola passing through the origin and point b in Figure 10-23. Every point on the parabola is a corresponding point. Because point b corresponds to point c, Equation 10-15 can be used to find speed at point b after Q is scaled at point c.

SI Units modified to m³ / h	**U.S. Customary Units**
$Q_c = 136\,\text{m}^3/\text{h}$ (by scaling)	$Q_c = 598\,\text{gal/min}$ (by scaling)
From Equation 10-15 ($n_b = n_c Q_b/Q_c$),	
$n_b = 1150\ (150/136) = 1270\ \text{rev/min}$	$n_b = 1150\ (660/598) = 1270\,\text{rev/min}$

By scaling H at points b and c, the speed can also be found from Equation 10-16

$(n_b = n_c\sqrt{H_b/H_c})$

$n_b = 1150\sqrt{15.4/12.6} = 1270\,\text{rev/min}$	$n_b = 1150\sqrt{50.5/41.3} = 1270\,\text{rev/min}$

(2) *Efficiency*. The efficiency of the new operating point is found by projecting a line from point c downward to point d on the efficiency curve, which gives an expected efficiency of 75%.

Multiple Pump Operation

In pumping stations where several pumps are installed, two or more pumps are usually operated in parallel. Occasionally, pumps are operated in series. The elements of the methods used to develop the combined H-Q curve for multiple pumps are outlined in the following subsections. A more extensive explanation for multiple pumps is given in Section 10-8.

Parallel Operation

When two or more pumps are discharging into the same header or manifold, the total flow is found by adding the individual flows at a given head. A minor error is introduced, however, because the system curve does not include headlosses in the suction and discharge piping. One method for eliminating the error is to subtract from the pump curve those station headlosses in the suction and discharge piping up to the intersection of the last pump discharge and the manifold (header). In the inset of Figure 10-24, the headlosses from a to b to c are subtracted from the H-Q curve for pump P1. Similarly, the losses from a to d to c are subtracted from the H-Q curve for pump P2. The modified pump curves are shown as dashed lines

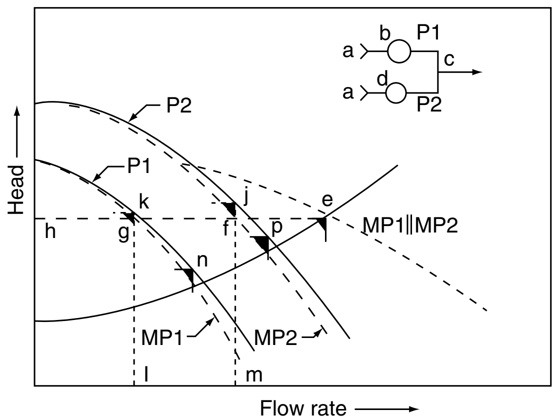


Figure 10-24. Operation of two pumps in parallel.

labeled MP1 and MP2 in the figure. Adding the abscissas (or flow rates) of MP1 and MP2 produces the pump curve MP1 || MP2, where the symbol "||" means "in parallel with." Thus, he = hg + hf. The intersection of MP1 || MP2 with the system curve for the manifold and force main downstream from point c is the operating point (point e) for the pumps when both are running.

To find the flow rate contributed by each pump, draw the horizontal line, eh, which intersects the modified pump curves at f and g. The discharge from P1 is hg; from P2 it is hf. To find the total head at which each individual pump operates, project a vertical line from f to j and from g to k. The actual operating point of P1 is point k and the head on the pump is kl. The operating point of P2 is point j and the head is jm. With one pump off, the other operates at point n (for P1) or point p for P2. The pump specifications must be written for pump P1 operating either at point k or point n and for pump P2 at point j or point p.

Series Operation

Pumps are operated in series, as shown in Figure 10-25, when the head requirement is greater than can be obtained with a single pump. The combined head capacity curve, P1 + P2, is found by adding ordinates—that is, the heads developed at the same flow rate . Thus, ad = bd + cd. The operating point is point c for pump P1 and point b for pump P2, and the operating point for the system is point a.

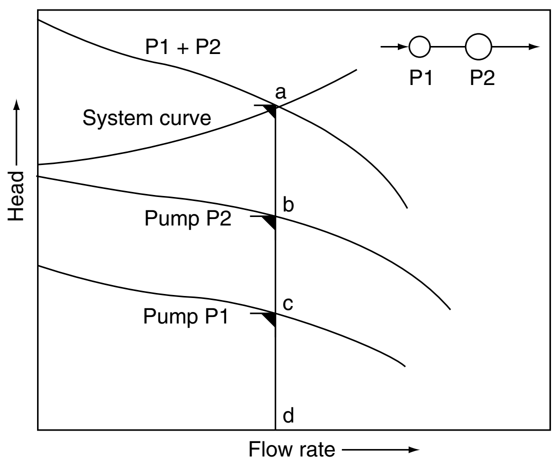


Figure 10-25. Operation of two pumps in series.

10-8. Practical Pumping System H-Q Curve Analysis

Review the fundamentals of pumping system analysis before reading this section in which additional details add practicality to this subject. The ultimate objective in analyzing a pumping system is to describe the required range of operating conditions in sufficient detail to be able (1) to understand the application completely, and (2) to describe the application to equipment suppliers in sufficient detail for them to apply their products intelligently.

As noted in Section 3-2, hydraulic calculations are never precise, regardless of the care taken in assessing headlosses in valves, fittings, and piping. The head on pumps is influenced by a number of variables including:

- Variations in static lift
- Variations in fluid viscosity
- Variations in pumping rate
- Deviations from assumed pipe roughness coefficients.

The last two variables have been the cause of many failed or poorly performing pumping stations. Engineers tend to focus obsessively on maximum flow and head for pump selection and acceptance tests, with insufficient emphasis on daily operation at minimum flow rates. Maximum design flow rates rarely occur, whereas, during dry weather or average demand days, low flow rates occur daily, and these usually impose the most severe operating conditions on pumping equipment—particularly in variable speed applications. As headloss calculations are inaccurate, prudent engineers develop system curves based on both the *worst* and *best* expected combination of headloss factors. Actual pumping conditions will lie somewhere between these two extremes, so pumps must be capable of operation within their AOR at any combination of head and flow between the two extremes. Furthermore, pumps should operate within their POR during the most common flow rates and heads—usually at or close to runout.

When considering operating regimes, visualize what is happening in the system and do not ignore characteristics that may alter system requirements. These include:

- Combination valves that may open to alter static lift at lower pumping rates
- Operation of other pumping stations that may be connected to the same discharge pipeline.

Constant-Speed (C/S) Pumps in Parallel

A typical schematic diagram of a pump and piping system for a pumping station with three duty C/S pumps (plus a standby pump) that discharge into a manifold and thence into the force main is shown in Figure 10-26. The system is divided into three basic components: force main, suction piping, and discharge piping. Head-capacity (H-Q) curves are drawn for each component. The H-Q curves for the pumps are then superimposed. In Figure 10-27, dynamic headlosses in suction and discharge piping are combined into Curve B+C. The pump, together with its suction and discharge piping (between manifold and wet well), is herein termed a "*pumping unit*."

Station Curves

In Figure 10-27, curves A, A′, and A″ are drawn first. They are combinations of static head plus the dynamic headlosses in and from the manifold to the point of discharge. The curves must encompass the entire envelope of operating conditions, including both HWL and LWL in the wet well, forebay, or supply main, and the extremes of friction losses in the force main. (See Section 3-2, Tables B-5 to B-7, and Figures B-2 and B-3.) Curve A represents losses at the greatest static lift (LWL in the wet well) and the *worst* condition for pipeline resistance losses—in this example for Hazen–Williams $C = 120$. Curve A′ represents the lowest static lift condition (HWL in the wet well) and similar pipeline losses. Curve A″ represents the lowest static lift condition with the most favorable (*best*) pipeline resistance losses—assumed here to be $C = 145$. The worst and best are the boundary conditions that describe the range of system head characteristics. Minor losses in the force main caused by valves and fittings can, perhaps, be ignored if the pipe is very long.

Curve B+C is the sum of friction and turbulence losses from the suction intake to the manifold. (Losses on the inlet side of the pump are calculated and accumulated separately, because they are needed to calculate NPSHA.) Here, turbulence losses in pipe fittings and valves are more significant than pipe friction losses. Typical loss coefficients are given in Tables B-6 and B-7. Curve B+C is drawn for the maximum expected losses.

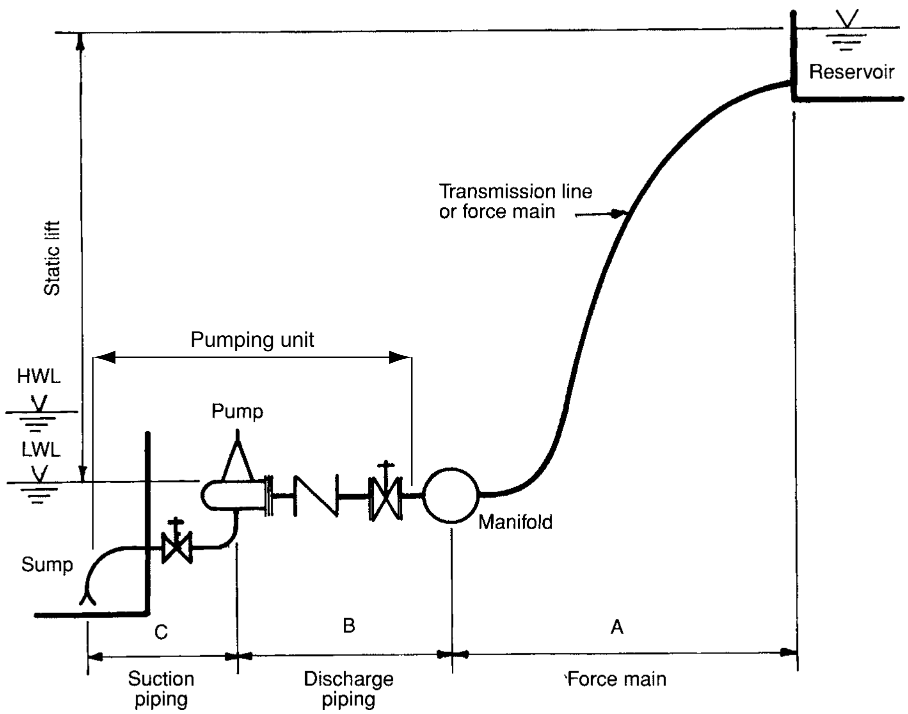


Figure 10-26. Schematic diagram of pump and piping system. One of four (3 duty) pumps is shown.

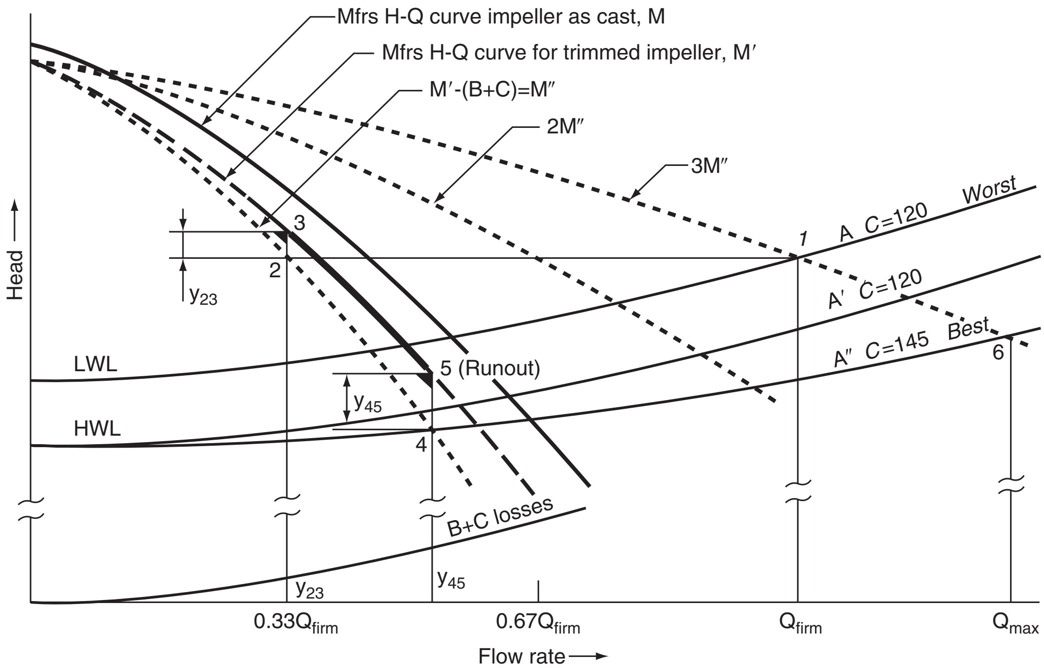


Figure 10-27. System H-Q curve analysis for three duty C/S pumps per Figure 10-26.

Analysis

Under the *worst* conditions, three pumps and their associated piping, B+C (the *pumping units*) discharge the firm pumping capacity, Q_{firm}, at the maximum head of Point 1, so each *pumping unit* discharges $Q_{firm}/3$ at an equal head, Point 2. The headloss from sump to manifold (Curve B+C) is y_{23}, so the pump must operate at Point 3, y_{23} above Point 2. As Point 3 represents the greatest head on the pump (unless some valve is closed), it is called the "warranted point," the only point for which both head and flow rate are warranted by the manufacturer. Note that the pumps will rarely—if ever—operate at Point 3, because Q_{firm}, the maximum flow that can occur, is always chosen to exceed expected peak flow rates, and furthermore, peak flows rarely persist for more than an hour or so.

Choose a pump with an H-Q curve (here, Curve M) that includes Point 3—that is, above and to the right of Point 3. With the proper impeller trim, Curve M becomes Curve M′, passing through Point 3. (Review Example 10-3 to ensure the affinity laws are applied properly for trimming impellers.) If the pump and its associated piping (C+B) is considered to be a *pumping unit*, its H-Q curve is M″ passing through Point 2 and, by extension, to Point 4 on the *best* curve (Curve A″). The abscissa of Point 4 represents the maximum flow rate that can pass through the pump, and the pump operating point is Point 5 (called runout), y_{45} above Point 4. When the system is new, the pump will normally operate at or near runout whenever the pump is activated by the HWL signal. For the life of the station, all operating points fall between Points 3 and 5.

Make further efforts to select a pump for which Points 5 and 3 are within the POR. If that range is too large, make sure that Point 5 is within the POR and Point 3 within the AOR.

What happens during the highest flow rate to the station? From the above, the H-Q curve for a *pumping unit* is Curve M″. When two pumps operate, the curve is 2M″, and three pumps produce Curve 3M″. The maximum station flow rate that can occur is labeled Q_{max} with the head at Point 6. If the same construction for Points 1, 2, and 3 is developed for Point 6, the operating point for the pump falls between Points 3 and 5.

Now refer to the manufacturer's NPSHR curve and determine whether there is sufficient NPSHA for a comfortable margin—not less than 130% of NPSHR if the operating point is within ± 15% of

BEC. Outside of that boundary, the margin should be much greater—as much as 180% of the NPSHR. The alternative is either to lower the planned setting for the pumps or seek another pump performance curve.

Always bear in mind that pump performance is worse, in terms of vibration, potential cavitation, and mechanical damage to the pump, at operating conditions different from the best BEP. As a general rule, off-the-shelf pumps perform best if operating conditions lie between about 70 and 115% of BEC.

Specifications for Purchase Orders

The conditions for both Points 3 and 5 *must* be published in the purchase order or project specifications, because that is the only legally defensible method for communicating performance requirements to the equipment manufacturer. When specifying Point 3, list head, flow, and NPSHA; however, when specifying Point 5, list only the head and NPSHA because every pump manufacturer has a pump curve of a different shape and the intersects for these curves with Curve A″ can vary.

Hydraulic Profile

The next step *(which should not be omitted)* is to plot the hydraulic profile for the system curves against the profile for the pipeline under the intended modes of operation. Although extreme accuracy is not critical, it is important to assess the system as it is intended to function. Try to visualize how the system will behave under all operating conditions. Questions to be asked when performing this task are:

- Are all operating conditions properly defined by the system curves?
- What happens in the pipeline under each operating condition?
- Are there any peculiarities in the operation of the system that have not been considered in the construction of the system curves?
- Has *everything* that might influence pump operating conditions (and, hence, pump performance) been considered?

An example of where one can go astray is seen in Figure 10-28, in which a knee is shown in the pipeline

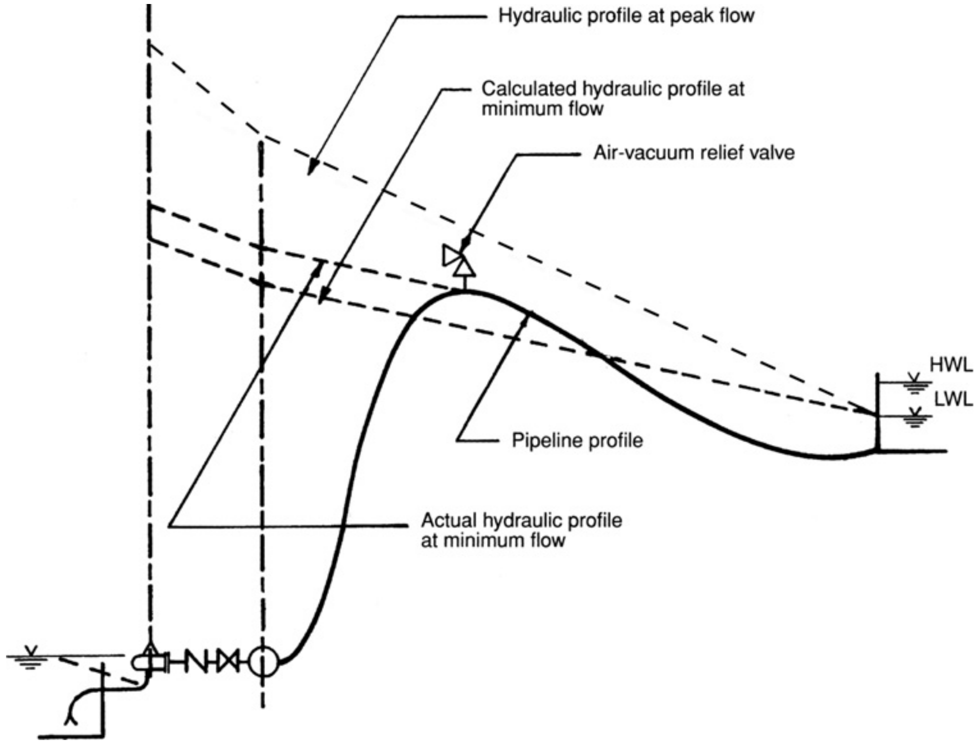


Figure 10-28. Hydraulic profiles for Figure 10-27.

profile. In this example, the operating point (Point 3 in Figure 10-27) for three pumps is properly defined. But when only one pump is running, the operating point (Point 5 in Figure 10-27) is wrong because the hydraulic profile in Figure 10-28 is below the high point in the pipeline. The air-vacuum valve will open, and flow will stop until the pump increases the pressure enough to send water over the high point. So by plotting hydraulic profiles, it can be seen that the transmission pipeline H-Q curve must be raised, as shown in Figure 10-29, to accommodate a new static head. The true flow rate is less than the flow rate obtained from Figure 10-27, so the system will not perform as intended. Blunders of this sort are unlikely to inspire client confidence.

Variable-Speed (V/S) Pumps in Parallel

The important difference between V/S and C/S pumping is that V/S pumps operate over a wide speed range, and care is required to make certain the pumps always operate near their BEP at all speeds (see Section 15-4). Other differences are: (1) storage is neither required nor desirable for V/S pumping; (2) fewer pumps are needed because of the V/S feature, so piping and space requirements are reduced; and (3) in properly designed systems, the difference between LWL and HWL in V/S pumping is small, slightly less than the diameter of the influent pipe discharging into the wet well, so the average static lift is less than that for C/S pumping.

Full-Speed Operation

For full-speed operation with all duty pumps (two in this example) operating, the construction for Points 1, 2, and 3 is the same as that for C/S pumps.

The HWL for a single operating *pumping unit* is at the mid-depth of the influent pipe, Point 4 on Curve M″, because for any water level higher, the pump controller is programmed to start a second pump. (For reasons, see Figure 12-22 and the accompanying text.) Point 5 (Runout) is y_{45} above Point 4.

Low-Speed Operation

The pump controller is programmed to operate a single pump at variable speed on a straight line

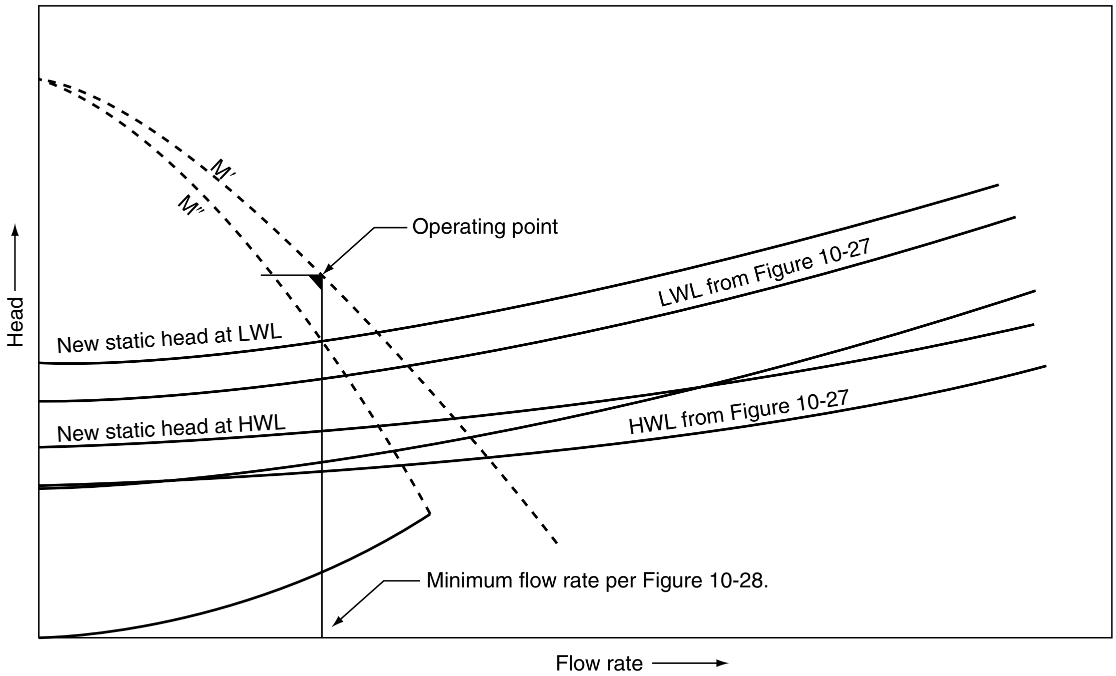


Figure 10-29. System curve analysis for one pump operating at minimum flow as per Figure 10-28. Compare to Figure 10-27.

basis for water levels between the invert and the half-depth of the influent pipe. Above mid-depth, the controller turns two pumps on, each operating at 50% capacity. (The straight-line basis is only an approximation, as can be seen by examining Figure B-4, but the errors are insignificant.) As the water level falls below the mid-depth, it is wise to let the controller continue to operate two pumps until the total flow rate is, say, 35% of Q_{firm} so as to prevent frequently switching pumps off and on as the water level fluctuates around mid-depth.

Point 6 for 0.35 Q_{firm} should be located 0.35 d (where d is the diameter of pipe) above the invert (0.65 d below the soffit) for the straight line approximation. From Figure B-4, it should be located 0.42 d above the invert (0.58 d below the soffit). In Figure 10-30, it is located 0.6 d below the soffit. (Note that as water level rises, the line representing water level falls in Figure 10-30 because the head falls.) Point 7 is y_{67} above Point 6.

For a single pump at minimum flow rate, $0.18Q_{firm}$, Point 8 is 0.18 d above the invert (on a straight line basis), so Point 9, at y_{89} above Point 8, appears to fall on Line A (but in reality, it does not).

All Pump Speeds

Obviously, the pumps must be able to operate at Points 3, 5, 7, and 9. They must also be able to operate anywhere within the cross-hatched region.

Other intersections can be tried, but the others all fall within the cross-hatched area.

Communicating with Manufacturers

It is the custom of pump manufacturers to guarantee no more than one operating point, so the guaranteed operating condition (Point 3 in Figure 10-30) should represent the required head and capacity at the *worst* assumptions for losses. Certain other requirements must also be met, and the following is a good example for accomplishment:

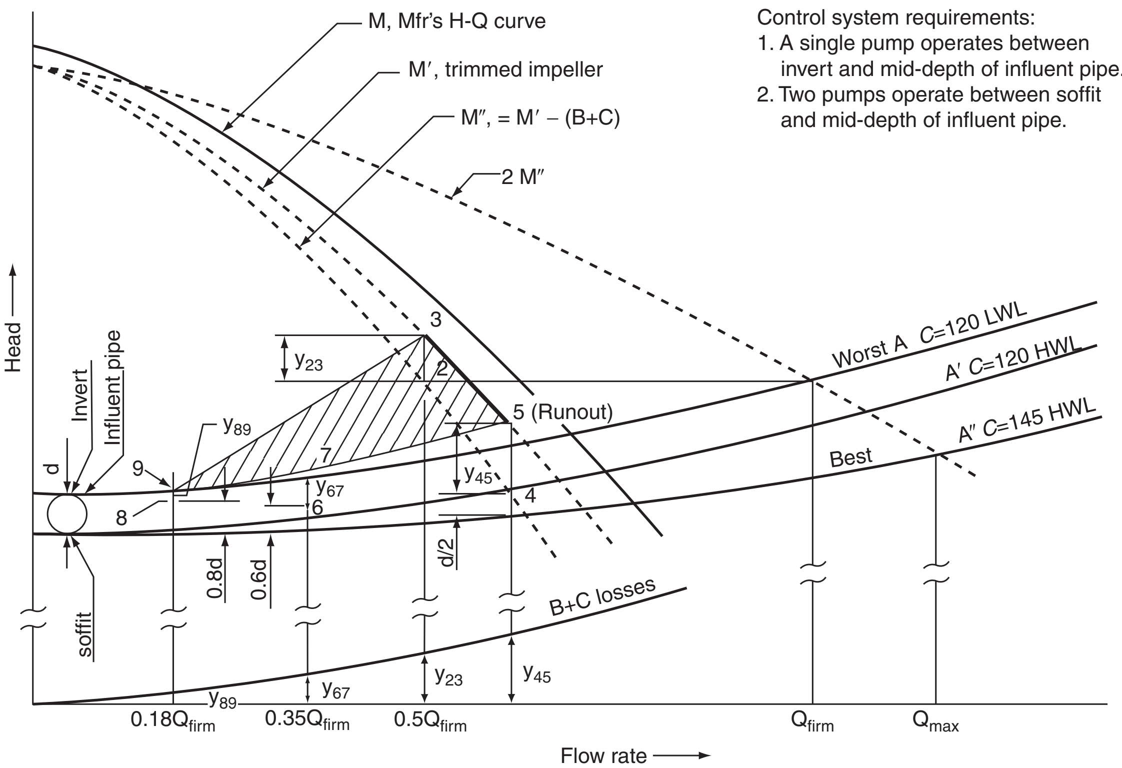


Figure 10-30. System H-Q curve analysis for two duty V/S pumps per Figure 10-26.

- Condition A. The required flow rate at maximum head is specified. (Point 3 in Figures 10-27 and 10-30.)
- Condition B. The minimum head (Point 5 in Figures 10-27 and 10-30) is specified. The flow rate is *not* specified because each pump manufacturer's curve unique, and to specify a second rate of flow would probably limit the choice to only one manufacturer. This operating head should lie within the pump's POR at the maximum operating speed. The maximum flow rate (runout) occurs at this minimum head.
- Condition C. The required efficiency is specified at the maximum operating speed, but the point at which it is attained is not specified. (It is unreasonable to expect and virtually impossible for a manufacturer to meet specified head, flow, *and* efficiency at any one point.)

In most instances, runout (or near runout) represents the most service time, so it should not be overlooked when developing the requirements for pump selection. If the pumps are not selected for runout, it is likely they will be subject to severe cavitation, damaging vibration, and costly repetitive repairs. The probable peak flow at the probable lowest head (Point 5 in Figures 10-27 and 10-30) will be a far more frequent operating condition than Point 3). Hence, select pumps so that the head and flow rate at Point 5 falls within the POR as defined by ANSI/HI 9.6.3.

The message is clear: *System hydraulic information should be developed for a range of assumptions and all potential operating conditions. Make certain that the all operating conditions are known and considered by the pump manufacturer for pump selection.*

Deep Wet Wells

Storing the peaks of storm water runoff in long tunnels far (~ 30 m or 100 ft) below ground surface is becoming popular. The tunnels vary from about 2.5 to 4.9 m (8 to 16 ft) in diameter, and hence the static head can vary by the same amount. The total dynamic head varies still more, and such large variations pose severe problems for designers, who must be very careful and thorough in selecting pumps and designing systems.

Computer Modeling

There are a number of digital computer programs that can be used to calculate pump performance curves and pump operating points. But none is known to be as flexible and to follow the principles stated herein as closely as PUMPGRAF2® [16].

10-9. Complex Pumping System H-Q Curves

The trend in engineering today is to depend on the newer and more advanced computer programs such as Cybernet® [17] or KY Pipe [18] to determine operating points in complex piping systems. Hence, the graphical method for solving complex problems described by Frey [19] in the first edition is omitted in this edition.

10-10. References

1. Karassik, I. J., J. P. Messina, P. Cooper, and C. C. Heald (Eds.), *Pump Handbook,* 3rd ed., McGraw-Hill, New York (2001).
2. Warring, R. H., *Pumps: Selection, Systems, and Applications,* Trade and Technical Press, Morden, Surrey, UK (1979).
3. DIPRA, *Handbook of Ductile Iron Pipe,* 6th ed., Ductile Iron Pipe Association, Birmingham, AL (1984).
4. Addison, H., *Centrifugal and Other Rotodynamic Pumps,* 3rd ed., Chapman & Hall, London (1966).
5. Anderson, H. H., *Centrifugal Pumps,* 3rd ed., Trade and Technical Press, Morden, Surrey, UK (1980).
6. Lazarkiewicz, S., and A. T. Troskolanski, *Impeller Pumps,* Pergamon Press, Warsaw, Poland (1965).
7. Norrie, D. H., *An Introduction to Incompressible Flow Machines,* Edward Arnold Publishers, London (1963).
8. Stepanoff, A. J., *Centrifugal and Axial Flow Pumps,* 2nd ed., Wiley, New York (1957).
9. Turton, R. K., *Principles of Turbomachinery,* L. E. & F. N. Spon, London (1984).
10. Wislicenus, G., *Fluid Mechanics of Turbomachinery,* Vol. 1, 2nd ed., Dover, New York (1965).
11. Cooper, P., "Cavitation: How does it happen?" *Pumps & Systems*, 11,16–21 (June 2002).
12. DuPont Engineering, P.O. Box 80840, Wilmington, DE 19880–0840.
13. Grohmann, M., "Extend pump application with inducers," *Hydrocarbon Processing,* 59, 121–124 (December 1979).
14. Rutschi, K., "Die fleiderer-Saugzahl als gutegrad der Saugfahigkeit von Kreiselpumpen," *Schweizerische Bauzeitung,* No. 12, Zurich (1960).
15. Metcalf and Eddy, Inc., *Wastewater Engineering: Collection and Pumping of Wastewater,* G. Tchobanoglous (Ed.), McGraw-Hill, New York (1981).
16. Wheeler, W., PUMPGRAF2®. For a free copy of this computer program with instructions, send a formatted 1.4-MB, $3\frac{1}{2}$ -in. diskette and a stamped, self-addressed mailer to 683 Limekiln Rd., Doylestown, PA 18901–2335.

17. Cybernet®, Haestad Methods, Inc., 37 Brookside Rd., Waterbury, CT 06708.
18. KY Pipe, Dept. of Civil Engineering, University of Kentucky, Lexington, KY 40506.
19. Sanks, R. L., G. Tchobanoglous, B. E. Bosserman, D. Newton, and G. M. Jones, Frey's Method in Section 10–8 in *Pumping Station Design* (1st ed.), Butterworth-Heinemann, Newton, MA (1989).

Chapter 11

Types of Pumps

RICHARD O. GARBUS

CONTRIBUTORS
Thomas L. Angle
Kirk Blanchard
Rodney L. Cusworth
Kundan Desai
Mayo Gottliebson
David A. House
Joseph K. Jackson
R. Russell Langteau
Colin Martin
Stephen G. Miller
Reynold Nickel
Lowell G. Sloan
Earle C. Smith
Robert E. Starke

The types of pumps commonly used in public utilities for the pumping of water, wastewater, and sludge are described in this chapter, together with details such as impellers, casings, seals, and wear rings. A summary of pump sizes and applications is given in Section 11-11, and a general comparison of advantages and disadvantages of the pumps is presented in Section 25-6. A more complete listing of pump parts is given by the Hydraulic Institute [1]. For pump dimensions and performance data, refer to the pump manufacturers' catalogs. Examples of pump selection are presented in Chapters 12, 17, 18, 19, 26, and 29.

References to standards are given in abbreviated form, such as ANSI B73.1, as that is sufficient to identify the publication.

11-1. General Classifications of Pumps

Pumps may be classified in many different ways. Classification by general mechanical configuration is used in this chapter. Figure 11-1 (an abridged version of the more complete listing given by the Hydraulic Institute [1]) is a chart of the pumps discussed here arranged according to their design. A slightly different version of Figure 11-1 is given in ANSI/HI 9.1-9.5-1994, a later edition of the standards, but it offers no advantage for this chapter. An overview of pumps is given in the following subsections. The pumps are described in more detail in Sections 11-3 through 11-10. There are two basic groups of pumps: (1) kinetic, and (2) positive displacement.

Kinetic Pumps

Kinetic pumps impart velocity and pressure to the fluid as it moves past or through the pump impeller and, subsequently, convert some of that velocity into additional pressure. Kinetic pumps are subdivided into two major groups: (1) centrifugal (or volute), and (2) vertical (or turbine) pumps.

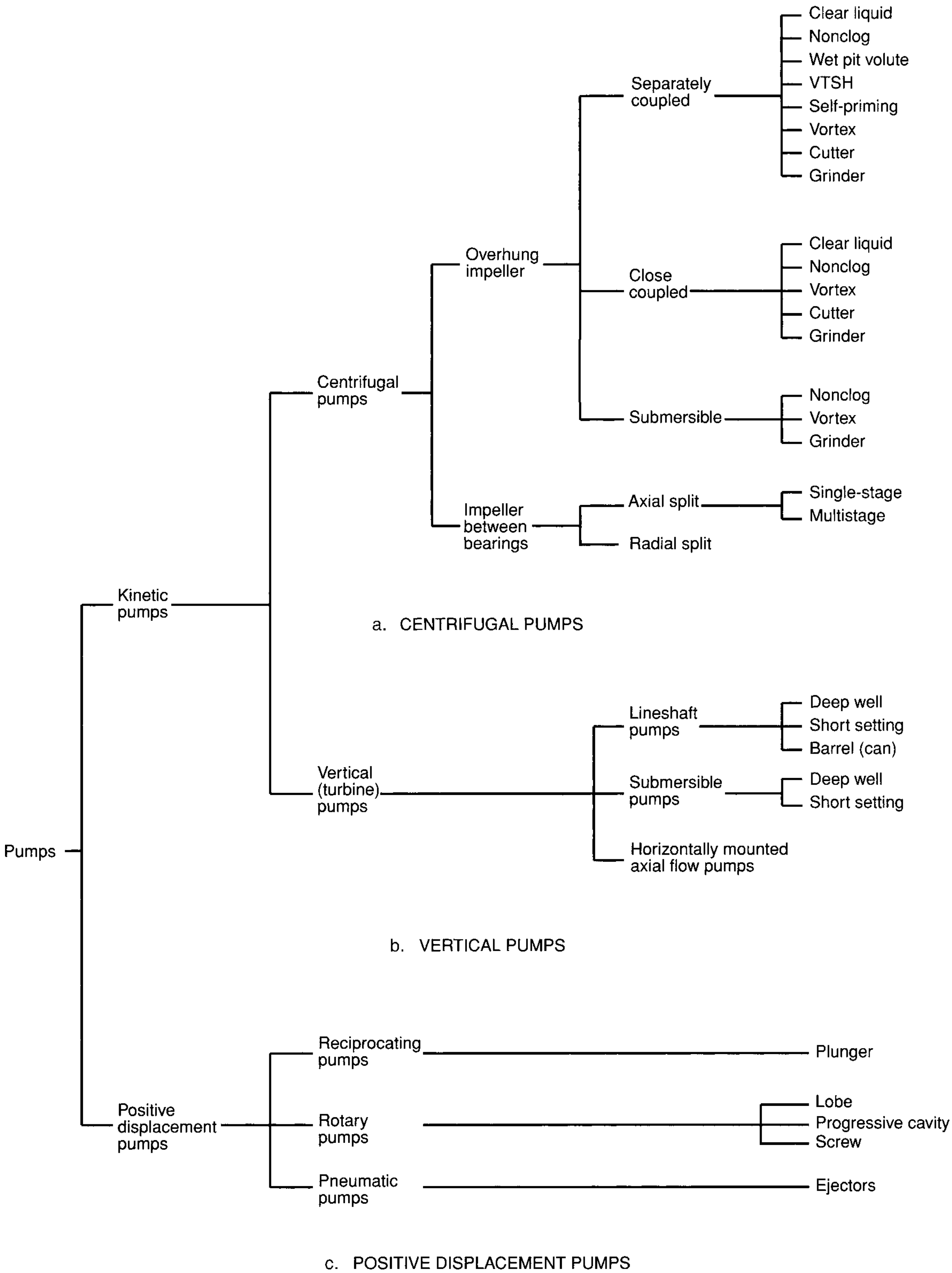


Figure 11-1. Classification of pumps. Courtesy of Hydraulic Institute Standards [1].

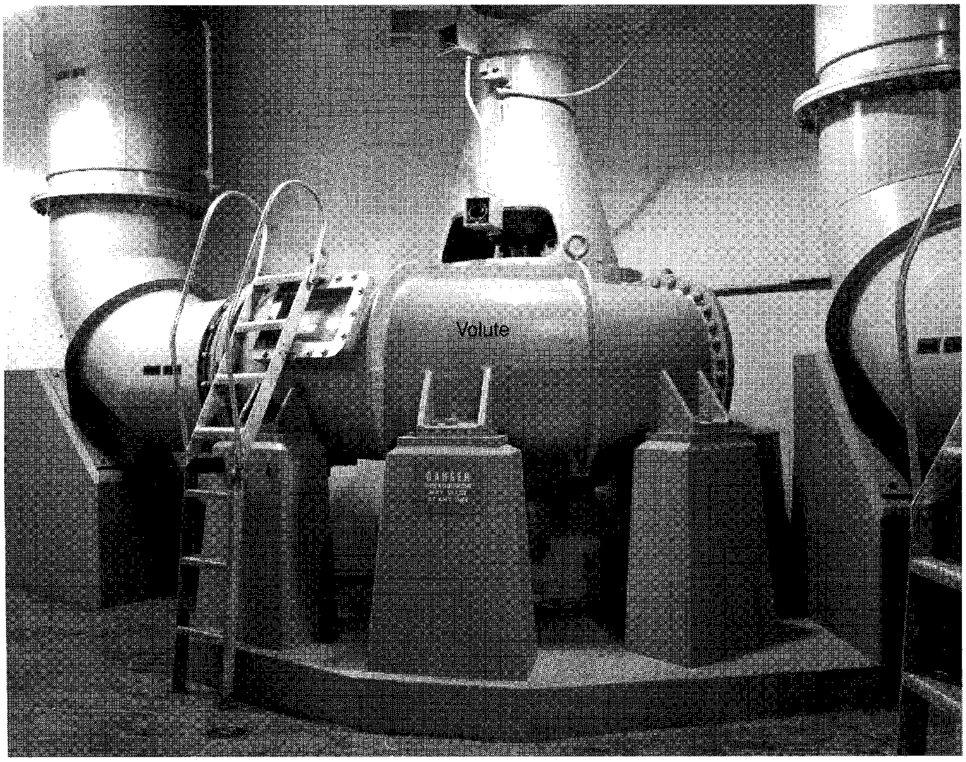


Figure 11-2. A centrifugal pump in a wastewater pumping station. The outer shell is the volute or casing. Photograph by George Tchobanoglous.

Centrifugal Pumps

All centrifugal pumps have one common feature: they are equipped with a volute or casing (see Figure 11-2). The function of the volute is to collect the liquid discharged by the impeller and to convert some of the kinetic (velocity) energy into pressure energy.

Vertical Pumps

Vertical pumps are equipped with an axial diffuser (or discharge bowl) rather than a volute. The diffuser performs the same basic functions as the volute (see Figure 11-3).

Positive Displacement Pumps

In positive displacement pumps, the moving element (piston, plunger, rotor, lobe, or gear) displaces the liquid from the pump casing (or cylinder) and, at the same time, raises the pressure of the liquid. The three major groups of positive displacement pumps are (1) reciprocating pumps, (2) rotary pumps, and (3) pneumatic pumps.

Reciprocating Pumps

In a reciprocating pump, a piston or plunger moves up and down. During the suction stroke, the pump cylinder fills with fresh liquid, and the discharge stroke displaces it through a check valve into the discharge line. Such pumps can develop very high pressures. Plunger pumps are the only reciprocating type of interest here (see Figure 11-4).

Rotary Pumps

The pump rotor of rotary pumps displaces the liquid either by rotating or by a rotating and orbiting motion. Lobe pumps (Figure 11-5) and progressing cavity pumps (Figure 11-6) are the two rotary pump categories discussed in this chapter.

In an Archimedes screw "pump" (a water lifter, really), a helical screw (Figure 11-7) rotates slowly and conveys water at atmospheric pressure trapped between the flights of the screw up a trough or a pipe.

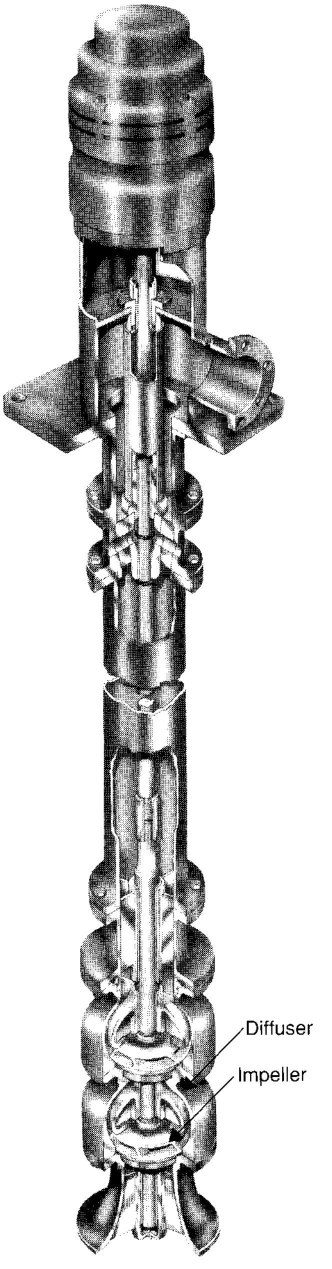


Figure 11-3. A vertical turbine pump. Courtesy of Johnston Pump.

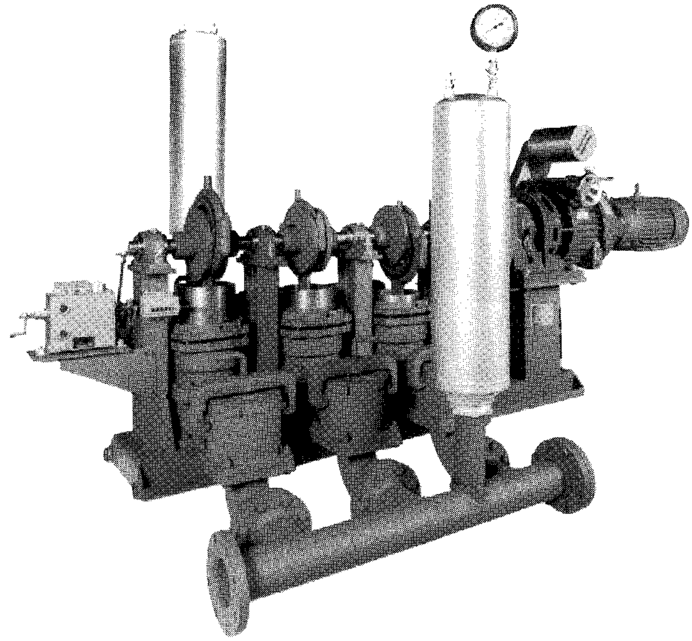


Figure 11-4. A plunger pump. Courtesy of Komline–Sanderson Engineering Corp.

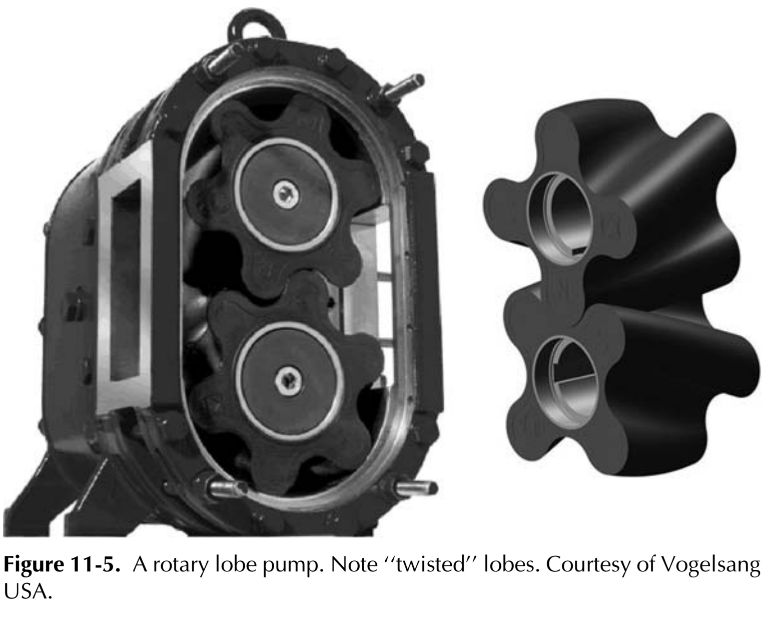


Figure 11-5. A rotary lobe pump. Note "twisted" lobes. Courtesy of Vogelsang USA.

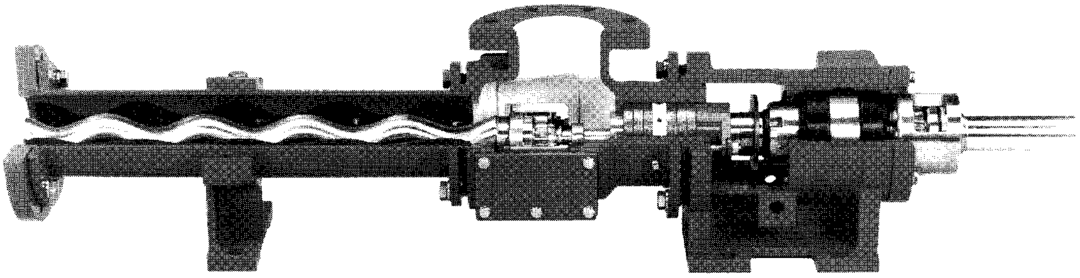


Figure 11-6. A progressing cavity Moyno™ pump. Courtesy of Robbins & Meyers, Inc.

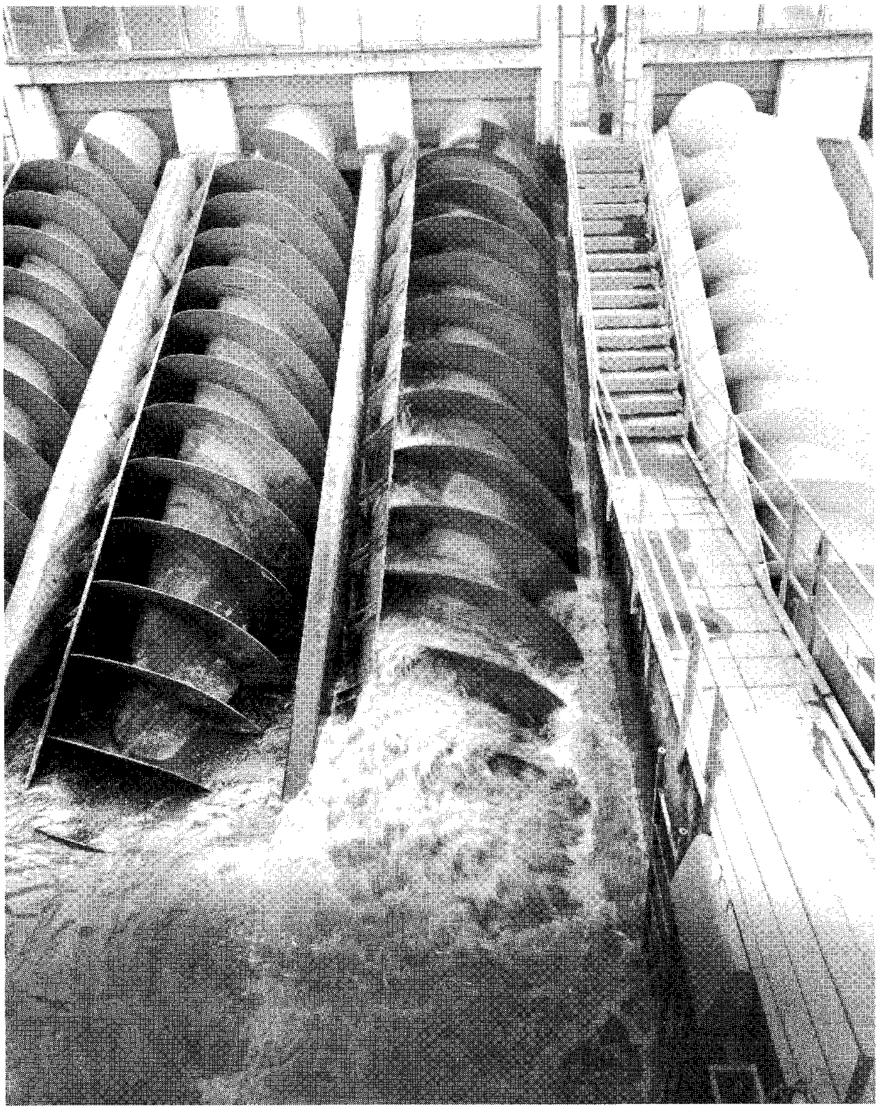


Figure 11-7. An open screw (Archimedes) Spiralift™ pump. Courtesy of Zimpro/Passavant, Inc.

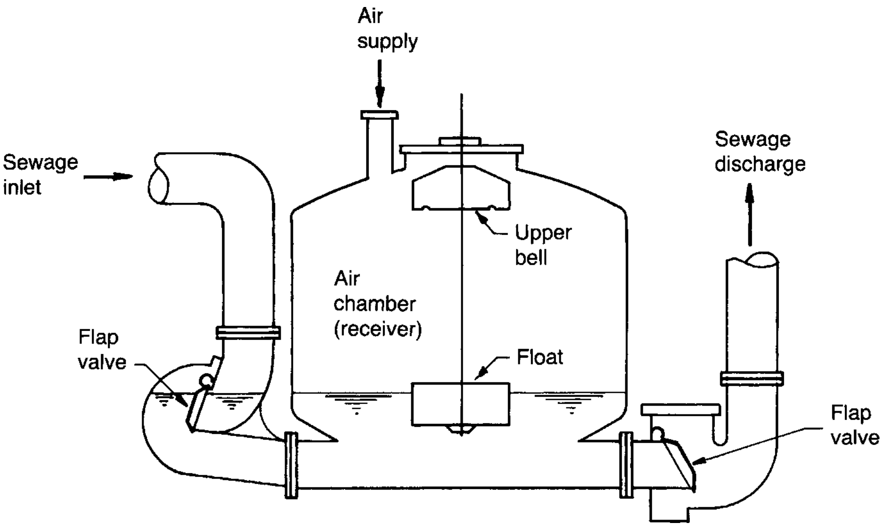


Figure 11-8. A pneumatic ejector. Adapted from Yeomans Chicago Corp.

Pneumatic Pump

Compressed air is used to move the liquid in pneumatic pumps. In pneumatic ejectors, compressed air displaces the liquid from a gravity-fed pressure vessel through a check valve into the discharge line (see Figure 11-8) in a series of surges spaced by the time required for the tank or receiver to fill again.

The reduced density of a column of an air–liquid mixture is used to raise the liquid in an air lift pump (see Figure 11-9). But because the water is not pressurized, it is not really a pump. It is therefore not included in Figure 11-1, and it should be called a "water-lifting device."

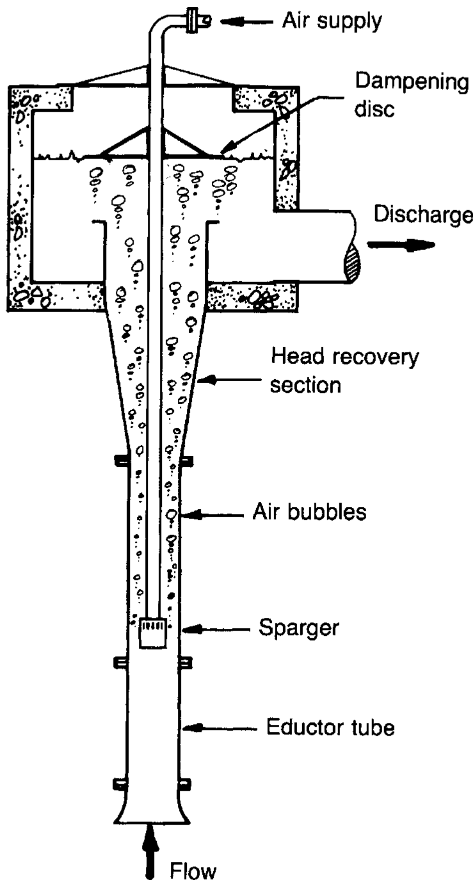


Figure 11-9. An air lift pump. Adapted from Walker Process Corp.

11-2. Classification of Centrifugal Pumps

Centrifugal pumps are subdivided into several categories, as shown in Figure 11-1a.

Overhung-Impeller Pumps

In an overhung-impeller pump, the impeller is mounted at the end of the pump shaft in a cantilever fashion. Both bearings are arranged on the same side of the impeller (see Figure 11-10).

Separately Coupled Pumps

Separately coupled pumps (also called "frame-mounted pumps") have their own shafts, which are coupled to and driven by the driver shaft (see Figure 11-10).

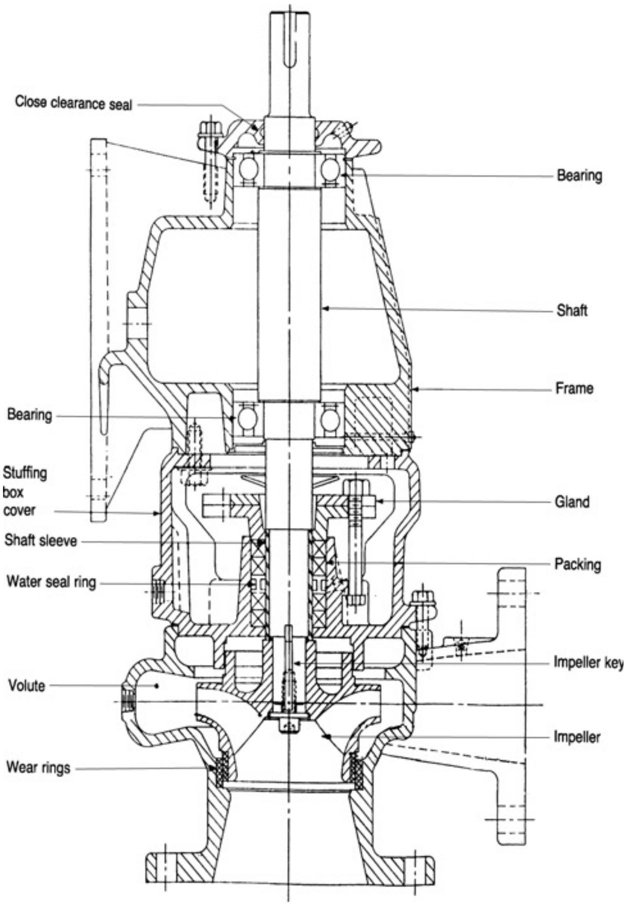


Figure 11-10. An overhung-impeller, separately coupled, end-suction, clear-liquid pump. Courtesy of Fairbanks Morse Pump Corp.

Close-Coupled Pumps

The impeller of a close-coupled pump is mounted on the driver shaft. There is no separate pump shaft, and no coupling is required between the driver and the pump (see Figure 11-11).

Submersible Pumps

Submersible pumps are close-coupled pumps driven by a submersible motor and designed for submerged installation in a wet well (see Figure 11-12).

Impeller-between-Bearings Pumps

In impeller-between-bearings pumps, the bearings are mounted on each side of the impeller (see Figure 11-13). Only the axial-split-case design is of interest here.

Axial-Split-Case Pumps

Axial-split-case pumps have a casing that is split along the (usually horizontal) centerline of the shaft. The impellers can be readily exposed for inspection and service by removing the upper half of the casing (see Figure 11-13).

11-3. Construction of Centrifugal Pumps

A representative centrifugal pump and its basic components are shown in Figure 11-10. The function of the components of the pump and the different options available with these components are discussed in the following subsections.

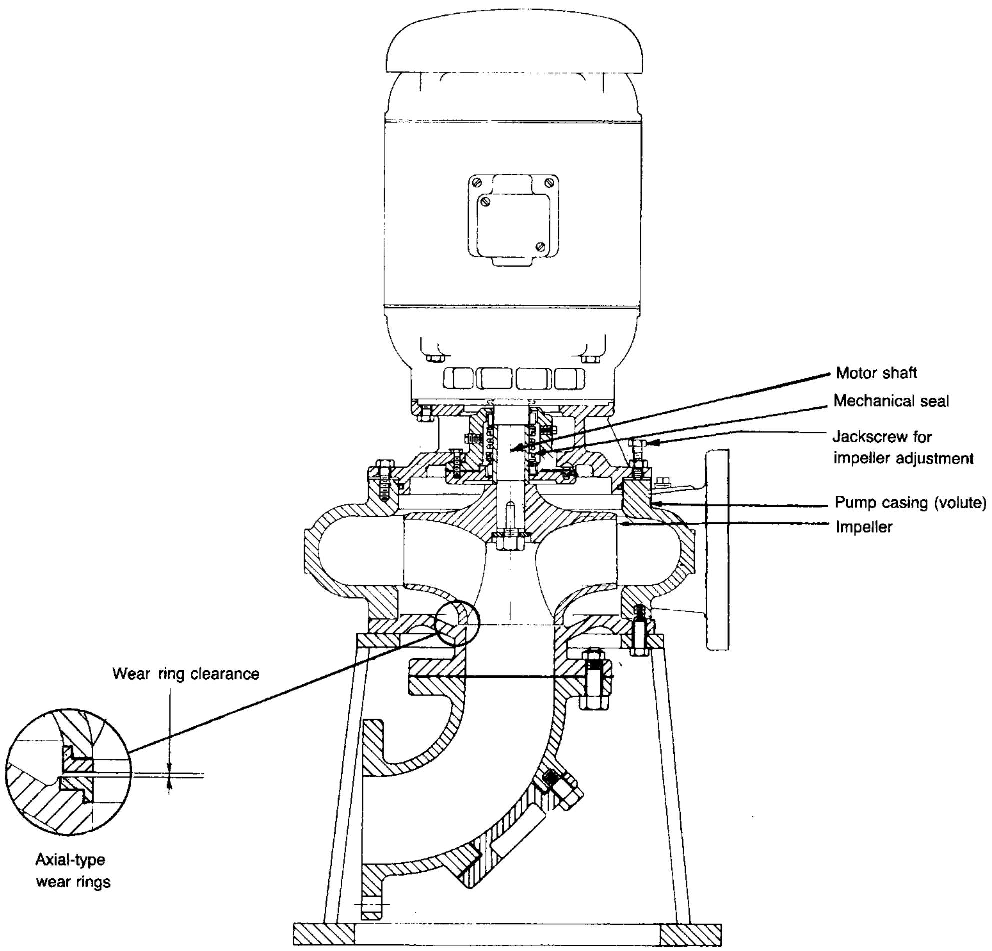


Figure 11-11. An overhung-impeller, close-coupled, end-suction pump. After Fairbanks Morse Pump Corp.

Impeller

The impeller increases the velocity of the liquid and raises its pressure. The impellers of centrifugal pumps may be of the radial- or mixed-flow type. The pump characteristics are determined by the impeller type. The relationship between impeller type, specific speed, and pump characteristics is discussed in Chapter 10 and illustrated in Figure 10-8.

Centrifugal pump impellers may be enclosed, semiopen, or open (Figure 11-14). Enclosed impellers are by far the most common. Semiopen impellers are used less frequently; they have, in theory, slightly better efficiency and are easier to cast, but they require close tolerance of axial clearances between the blades and the front cover. The third type, the open impeller, requires close clearances on both sides simultaneously. Because such tolerances are difficult to maintain, open impellers are seldom used in centrifugal pumps.

The pressure of the liquid at the impeller discharge is higher than at the impeller inlet, so the liquid tends to circulate back to the impeller suction. Some provision must be made to contain this recirculation. This is the function of the impeller and casing wear rings in enclosed impellers, and the wear plates in semiopen impellers.

The impellers are subjected to radial and axial thrust forces. Radial thrust is caused by unequal pressure distribution in the volute due to volute asymmetry. The theory of radial thrust is presented in Section 10-6.

The axial thrust is caused by the pressure differential between the front and back sides of the impeller. Axial thrust in multistage, high-pressure pumps can reach high values. These pumps are sometimes equipped with hydrodynamically balanced impellers, as shown in Figure 11-14e. This type of impeller has a balancing wear ring on its back shroud which, combined with

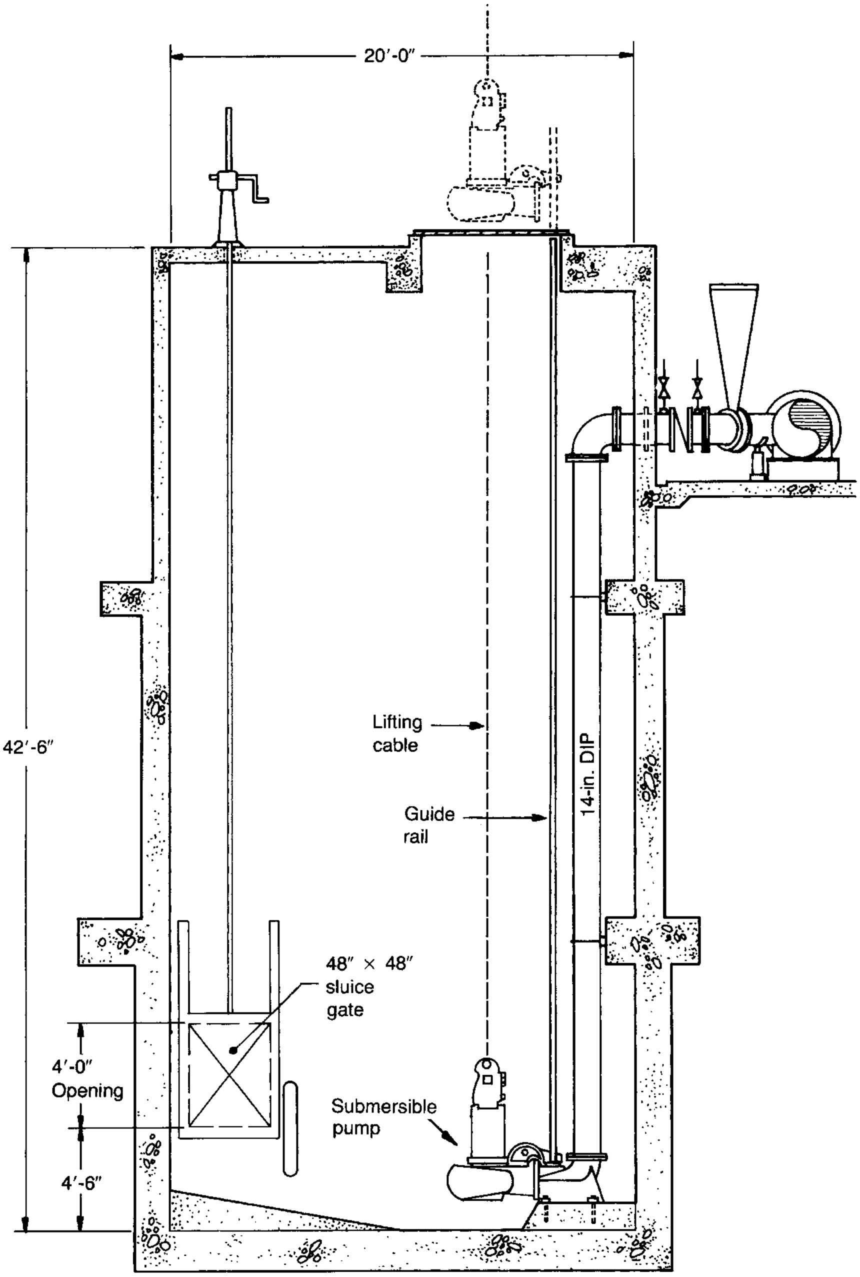


Figure 11-12. Submersible pump (pull-up design) in the Lancaster Reclamation Plant, Los Angeles County Sanitation District No. 14.

pressure relief (or pressure balancing) holes, equalizes the pressure on both sides of the back shroud and reduces the axial thrust. Although the back shroud wear rings reduce the axial thrust, they also have disadvantages: (1) they complicate the pump design, and (2) the pressure relief holes cause continuous recirculation of the pumped liquid with a resulting loss of efficiency. Consequently, hydrodynamically balanced impellers are rarely used in the pumps discussed in this book. Instead, the thrust bearings carry the full axial load which, from practical experience, has been shown to be the simplest technical solution.

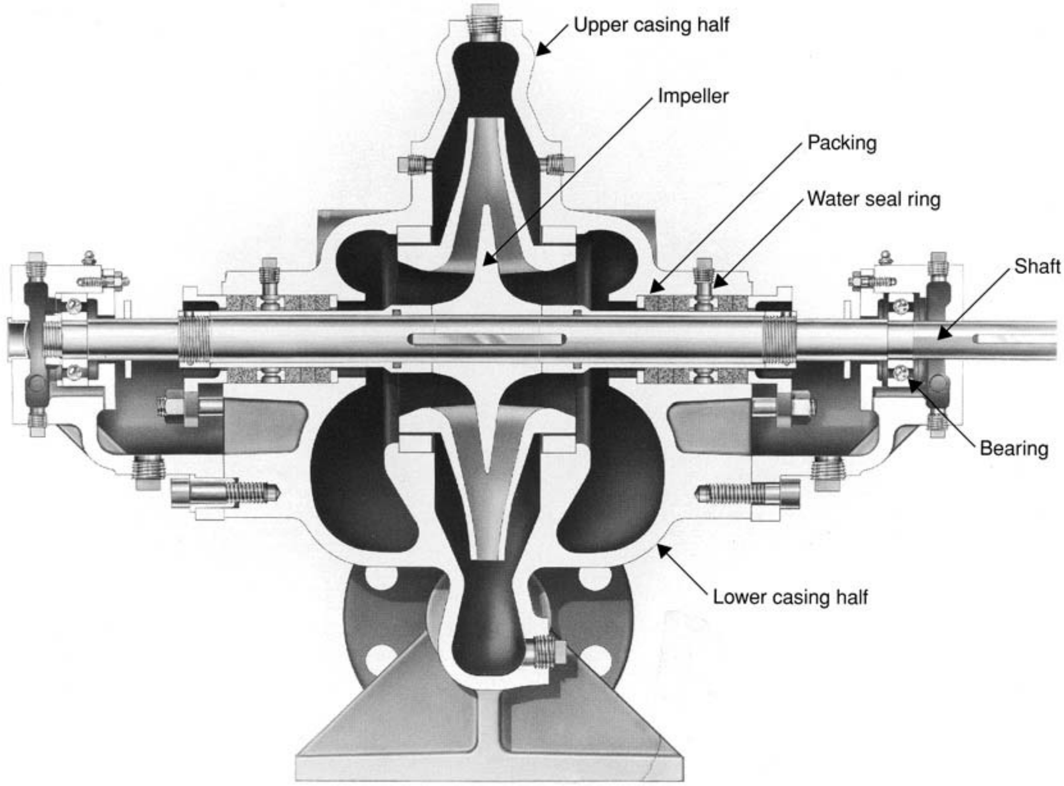


Figure 11-13. An impeller-between-bearings, axial-split, single-stage pump. Courtesy of Dresser Pump Division.

Wear Rings

The wear rings control the liquid recirculation between the impeller discharge side and the inlet side. The wear of wear rings is caused by grit or other abrasive matter in the pumped liquid. Rubbing of wear rings also causes wear, but in well-designed pumps the shaft deflection does not exceed wear ring clearance unless the pumps run outside of their design operating range (or close to shut-off). The wear rings may be separate or integral. Separate rings are machined independently from the impeller and the inlet cover and then pressed or fitted in place and locked by mechanical devices such as set screws, pins, or locking compounds.

The phrase "integral wear rings" means that no separate wear rings are installed, but that a close clearance is arranged between the impeller and the inlet cover. Enough material is usually provided in the cover and on the impeller so that after the wear ring clearance exceeds the recommended limits, both can be remachined and separate replacement wear rings can be installed.

Wear rings can be of axial, radial, or some special design. Axial or "face-type" wear rings rely on the impeller setting relative to the suction head to maintain the wear ring clearance. They require an axially adjustable impeller or an axially adjustable suction cover (see Figures 11-11 and 11-15b). Axial wear rings have the advantage of being adjustable for wear in the field without the need for pump disassembly or wear ring replacement. They do, however, complicate the pump design. They are used primarily on pumps smaller than about $0.25\,m^3/s$ (4000 gal/min).

Radial-type wear rings are not sensitive to the exact axial impeller location and, hence, do not require axial adjustment of the impeller (see Figures 11-10, 11-14a, 11-15a, and 11-15c). They cannot be readjusted for wear, however, so they must be replaced when leakage caused by wear becomes objectionable.

Some pumps designed for gritty liquids and very long wear ring life are equipped with flushed wear rings (see Figure 11-15c). Flushing liquid, free of abrasive particles and injected under pressure into the wear ring clearance, prevents the entry of the grit.

L-shaped wear rings (Figure 11-15d) are another option designed to increase the length of the leakage path and to reduce the leakage rate. For gritty applications, they may be combined with the external flush

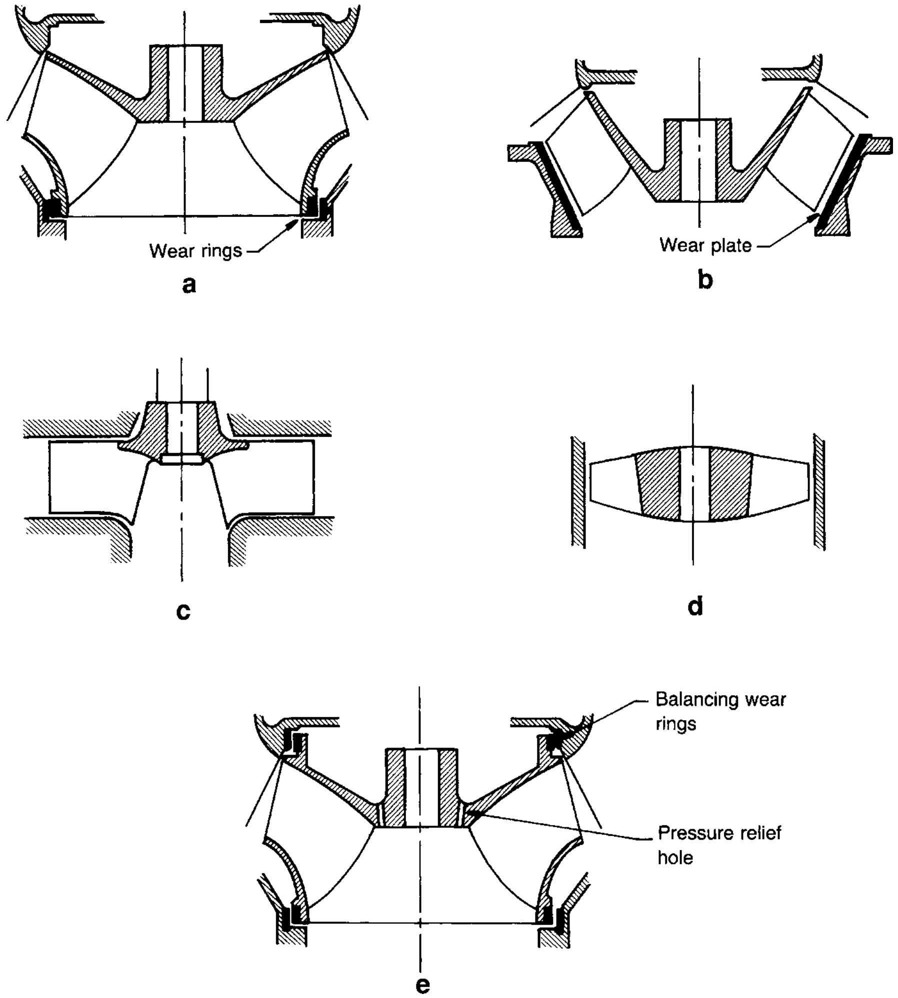


Figure 11-14. Impeller designs. (a) Enclosed impeller; (b) semiopen impeller; (c) open impeller, radial-flow; (d) open impeller, axial-flow; (e) hydrodynamically balanced impeller. After Fairbanks Morse Pump Corp.

option to conserve on flushing water requirements. They do require the same axial adjustment as the face-type rings.

Because the wear rings may contact each other under extreme operating conditions, the wear ring material must be both nongalling and compatible with the pumped liquid. A difference in ring hardness of Brinell hardness number 50 on the C scale (50 BHN) is generally recommended, especially for stainless-steel rings. Front head rings are usually made of the harder material.

Wear Plates

Semiopen impellers have no wear rings but rely on the close clearance between the vane tip and the suction cover to control the leakage around the vane tips (see Figure 11-14b). When gritty liquids are pumped, wear of the blade tips and of the casing is to be expected. Hence, the suction cover is often equipped with a wear plate, which has a contour matching the blade tip contour and can be axially readjusted to compensate for wear. The wear plates are usually made of abrasion-resistant material such as Type 416, heat-treated stainless steel (e.g., ASTM A 743-CAIS, 350 BHN).

Pump Shaft

In transmitting power from the driver to the impeller, the pump shaft must support all radial and axial

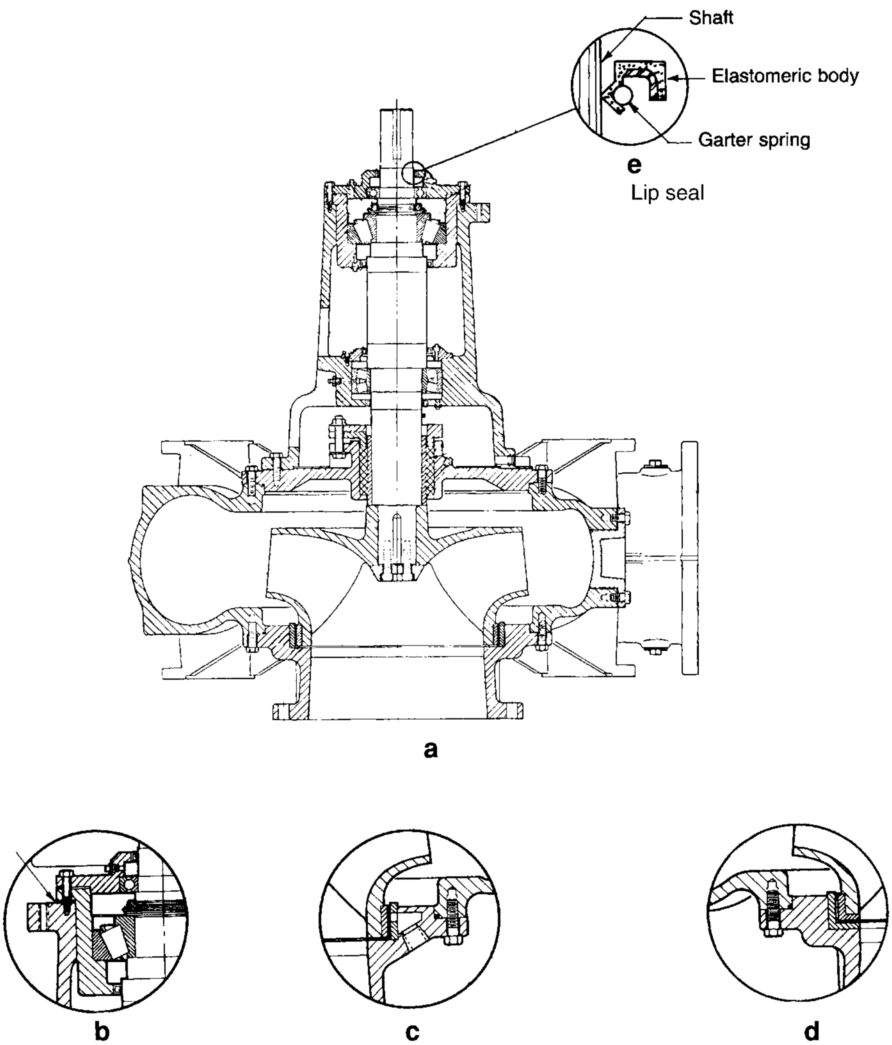


Figure 11-15. A solids-handling pump. (a) Pump; (b) impeller axial adjustment; (c) wear ring external flush; (d) L-shaped wear rings; (e) lip seal. Courtesy Fairbanks Morse Pump Corp.

impeller forces. It is subject to torsion, bending, and tension. Good shaft design is needed to ensure that the endurance limit of the shaft material is not exceeded under any anticipated operating conditions. Furthermore, the deflection of the shaft must be controlled to avoid wear ring rubbing and damage to the packing or the mechanical seals. A deflection of 0.05 mm (0.002 in. or 2 mils) at the shaft seal is often given as a recommended limit. Deflection at radial wear rings should be compatible with wear ring nominal clearance.

Shaft Sleeve

Most shafts are equipped with a replaceable shaft sleeve to protect the shaft from wear in the packing area (see Figure 11-10). Typical sleeve material is heat-treated Type 416 stainless steel, 300 to 350 BHN. The sleeve must be sealed to the shaft to stop leakage of the pumped liquid, and either O-rings or anaerobic sealing and locking compounds (which harden in the absence of oxygen) are used for this purpose.

Positive locking devices (such as pins, keys, or notches) are sometimes used to prevent sleeve rotation on the shaft, but the use of sealing compounds is also a successful locking method. Shaft sleeves are not necessary if a nonfretting mechanical seal is used with the added advantage of increased shaft strength.

Pump Casing

The pump casing or pump volute encloses the impeller and contains the discharge and, sometimes, the suction nozzles (see Figure 11-10). The casing collects the high-velocity flow from the impeller and converts some of the velocity energy into pressure. The casing may be one piece or of axial- or radial-split design.

Suction Cover

The suction cover (or front head) encloses the suction opening of the casing and also contains the suction nozzle of the pump. On some designs, the suction cover is an integral part of the pump casing (see Figure 11-10).

Stuffing Box Cover

The stuffing box cover (also called the "back head" or "adapter") encloses the inboard opening of the casing and also contains the stuffing box (see Figure 11-10).

Frame and Bearing Housings

Separately coupled pumps with overhung impellers have a frame that contains the pump bearings (see Figure 11-10). The bearings may be mounted directly in the frame or in bearing housings mounted in the frame. Bearing housings facilitate the assembly of the bearings and are used in larger pumps. With horizontal pumps, the frame frequently supports the entire pump (see Figure 11-10).

Bearings

The pump bearings carry the shaft and absorb all of the radial and axial forces acting on the shaft. A great majority of the centrifugal pumps for pumping stations are equipped with antifriction bearings (rollers, tapered rollers, or balls in single or multiple rows). Journal bearings are used predominantly for high-speed and high-load applications (e.g., boiler feed pumps and multistage, high-pressure chemical pumps) and are of little interest to pumping station designers.

The thrust bearing of the cantilever-type pump carries not only all of the thrust load but also its share of the radial load. The thrust bearing must be capable of supporting axial loads in both directions because thrust reversal can be expected with most pumps, especially during the starting process. In some designs, a separate (usually smaller) thrust bearing is installed for the thrust reversal. Typically, the thrust bearing is pressed on the shaft and locked in the bearing housing (or in the frame); thus, the thrust bearing determines the axial location of the shaft and the impeller. The radial bearing is also pressed on the shaft, but is free to slide in the frame or in the bearing housing. The bearings are thereby protected from excessive axial loads generated by unequal thermal expansion of the shaft and frame.

The antifriction bearing design is of little consequence as long as the bearings have been selected by the manufacturer to deliver adequate bearing life and the design provides for proper bearing installation and lubrication. The bearing life can be expressed in statistical terms only. Commonly, the L-10 bearing life rating is used. It is defined by the AFBMA as the number of operating hours at a given load that 90% of a group of bearings will complete before the first evidence of fatigue develops. Average life is statistically three to five times the L-10 life (depending on whether the rating is for degassed or nondegassed steel). Average life is defined as the operating hours at which 50% of the group of bearings fail and the rest continue to operate.

The L-10 life must be specified and computed for a selected pump operating condition. The bearing life is a function of bearing load and speed. The impeller radial loads (and, to a lesser degree, the axial loads) vary considerably between the best operating point (BEP) and shut-off of the pump (see Figure 10-18). As a result, *the computed L-10 life at shut-off for a typical, single-volute pump can be as little as one-hundredth or less of the life at BEP*.

A frequent specification is 40,000 h of L-10 life at the operating point for pumps in continuous, 24-h service. Specification of bearing life at shut-off or for the entire head-capacity range of the pump (which is the same as shut-off) should be avoided. Such a specification leads to oversized shafts and bearings and does not ensure the computed bearing life because, at shut-off, the bearings are exposed to shock loading, the effect of which cannot be readily predicted or

prevented. Operation at shut-off can be damaging to the shaft, impeller, and other structural components of the pump as well and, therefore, operation at shut-off for prolonged periods of time should be avoided.

L-10 life in excess of 100,000 h is of no practical value. It can be taken only as an indication that bearing fatigue failures are unlikely. Bearings fail anyway (as shown through experience) due to other causes such as corrosion caused by water contamination or deteriorated lubricants, rolling surface damage due to abrasive particles, and lack of lubricant. However, if bearing isolators (particularly the non-fretting vapor block type) are used, bearings are effectively protected against these contaminants and life is extended indefinitely.

Grease is the most common lubricant for pump bearings. It requires relatively little attention and service and usually gives long bearing life. Oil-bath lubrication is used much less frequently because retaining the oil in the bearing housing poses some technical difficulties, especially on vertical pumps. Force-fed oil lubrication is used very infrequently. Although it ensures the lowest bearing operating temperatures, provides the best protection from bearing contamination, and gives the longest bearing life, it requires expensive auxiliary equipment (which also needs service and attention). In the experience of pump operators, the benefits gained from force-fed lubrication do not justify the additional expense.

Bearing Seals

The bearings must be protected from contamination by abrasive particles or corrosive liquids (including water). Consequently, the bearing frame must be sealed. The following are typical seal arrangements.

Close-Clearance Seals

Close-clearance seals consist merely of a close clearance between the shaft and the frame or bearing housing (see Figures 11-10 and 11-15). They are very simple, have an unlimited life, and require no maintenance. However, they give only limited protection from water spray (e.g., hosing down the pumps) or from condensation during shut-down. Still, close-clearance seals are very successful on continuously operating and well-maintained pumps where the normal bearing operating temperature prevents any water condensation and grease acts as a barrier to outside contaminants.

Elastomer Seals

Elastomer (lip-type) seals are the most frequently used seals in current designs. They give good protection from contamination and moisture, but they also generate friction heat and cause higher bearing-operating temperatures with potentially reduced lubricant life (Figure 11-15e).

Labyrinth Seals

Labyrinth seals offer good bearing protection and generate no heat. They require more space and are relatively expensive. At present, they are offered as optional equipment only. They are suitable only for horizontal shafts.

Pump Shaft Seals

To contain the liquid in the pump, the pump shaft must be sealed against the pressure in the pump. The two basic types of seals are packings and mechanical seals, and both have advantages. The final selection of seal type depends on installation requirements and owner preferences, but packing is recommended for water and wastewater service in any installation where leakage from the stuffing box is not objectionable or on vertical turbine pumps that require dismantling for seal removal.

Packing

Packing is the simplest and most frequently used seal. The packing is usually made of braided synthetic fibers impregnated with special lubricating compounds. The packing rings require continuous lubrication and cooling, accomplished by using the pumped fluid or an external clean water source, so a properly adjusted packing must continuously leak and drip. All packings require periodic inspection and occasional adjustment of the gland (see Figure 11-10).

Packings are frequently equipped with a water seal ring (see Figure 11-10) so that a buffer liquid from an outside source (potable water or chlorinated final effluent for wastewater pumps) or from the pump discharge (for clean water pumps) can be injected into it. This construction ensures that the packing gets an adequate supply of lubricating and cooling liquid under all operating conditions, particularly when the pressure on the pump side of the packing drops below the atmospheric pressure. The injection of clean liquid is also used to prevent contaminants

and gritty materials in the pumped liquid from entering the packing and causing accelerated shaft sleeve wear. If potable water is used as the buffer liquid, an air gap must be provided between the fresh water source and the packing to prevent contamination of the water supply with the pumped liquid.

Although most types of packing in wastewater service must be flushed with clear water to avoid long-term scoring or abrasion of shaft sleeves due to grit in the flow stream, some types of braided, graphite-lubricated and impregnated or Teflon®-impregnated, non-asbestos packing have proven to work effectively in such service without any external clear water flushing.

Mechanical Seals

Mechanical seals have the advantages of no leakage and no need for periodic maintenance, but they are expensive and require care in installation. Some are subject to catastrophic failure. For mechanical seal replacement, the pump must be partially disassembled, which is costly. Mechanical seals should therefore not be specified indiscriminately. Note that API682 has an excellent discussion of mechanical seals.

A typical, simple, single-face mechanical seal is shown in Figure 11-16. Mechanical seals rely on very close clearance between the stationary and rotating seal faces for sealing. Both faces are lapped flat to within approximately a micron. They are lubricated and separated from each other by an extremely thin film of the pumped liquid. The most frequently used sealing face materials are ceramic for the stationary seal and carbon (sintered graphite) for the rotating seal. The secondary seals that prevent leakage around the sealing faces are rubber or synthetic elastomer compound O-rings or bellows.

Double-face mechanical seals (Figure 11-16b) are installed when the pumpage contains abrasives (e.g., wastewater containing gritty material). Clean liquid is injected into the cavity between both seal faces at a recommended pressure of 70 kPa (10 lb/in.^{2}) above the sealed liquid pressure. It lubricates and cools both seals and prevents intrusion of the pumped liquid between the inner seal faces. A wide variety of seal designs, arrangements, and materials is available for corrosive or toxic liquids, high pressures, and high surface velocities, but these exotic designs are not required in water and wastewater service.

SpiralTrac™ Seals

SpiralTrac™ [2] greatly reduces flush by using the rotating flow to create an axial flow component to sweep particulates out of a stuffing box or away from a mechanical seal—essentially by centrifugal action. They prevent the scoring of shaft sleeves and substantially increase the life of bearings, packing, and mechanical seals.

Separately Coupled Pump Design

A pump designed for a separately coupled driver is shown in Figure 11-10. The pump itself is totally self-sufficient and directly mounted on a base or a foundation. The driver is either separately mounted on the same base or foundation, or it can be mounted directly on the pump frame. A flexible coupling is usually used to connect the pump to the driver.

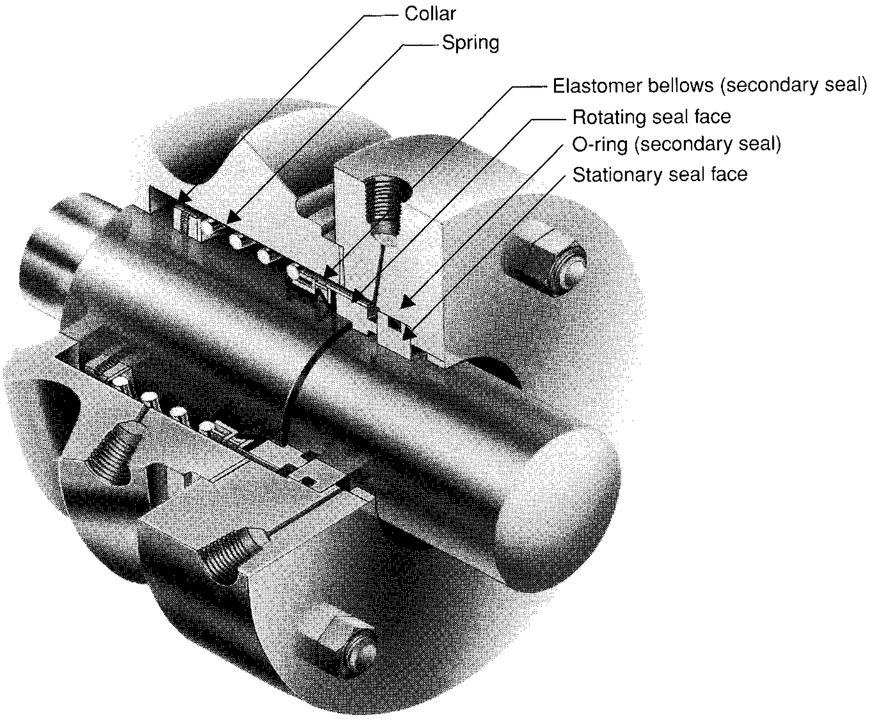
Close-Coupled Pump Design

Close-coupled pumps have the pump impeller installed on the motor shaft and the pump casing bolted directly to the motor (see Figure 11-11). Compactness and low initial cost are the prime objectives of this concept. The motor shaft of close-coupled pumps must resist all impeller radial and axial thrust forces. Hence, the motor bearings must carry the pump impeller loads in addition to normal motor electromagnetic and torsional loads. Standard flange-mounted motors are used for the close-coupled pumps when the shafts have adequate strength and rigidity and when the bearing life meets the specified requirements. Special motors with bigger bearings and shafts must be supplied when the loading exceeds standard motor capabilities.

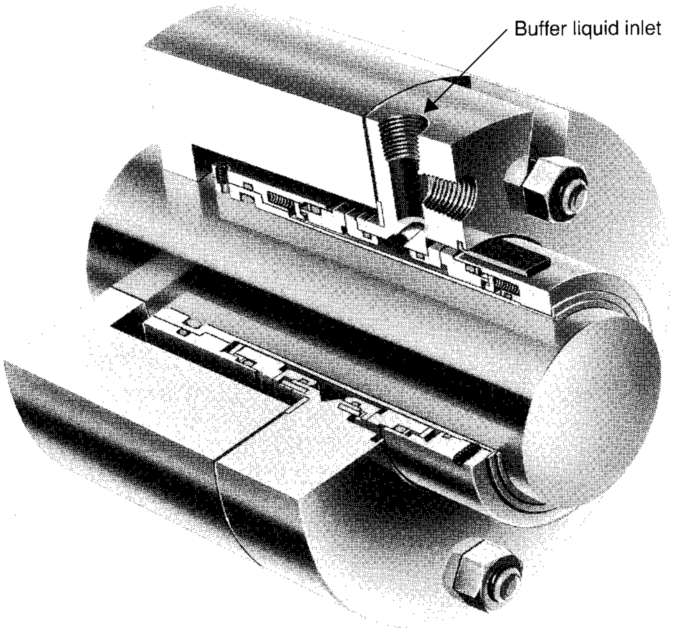
Submersible, Close-Coupled Pump Design

Submersible, close-coupled pumps operate while immersed in the pumped liquid. The submersible motors are single-purpose machines designed specifically for this application (see Figures 11-12 and 11-17). The motor housing is hermetically sealed to prevent the intrusion of the pumped liquid. The motor is usually equipped with two mechanical seals, and the space between the seals is filled with oil. To compensate for the thermal expansion of the oil, the oil chamber must contain a pressure-limiting device, which is usually an air cushion or a sealed bladder. *The pressure-limiting device is important. Several serious accidents have occurred in which a pump exploded when lifted from the wet well soon after being stopped.* The cause of the explosion was the expansion of the oil due to stored heat that was no longer dissipated by water.

Next Page



a



b

Figure 11-16. Mechanical seals. (a) Single-face mechanical seal; (b) double-face mechanical seal. Courtesy of John Crane, Inc.

Chapter 12

Pumps: Intake Design, Selection, and Installation

GARR M. JONES
ROBERT L. SANKS

CONTRIBUTORS
Stefan M. Abelin
Richard (Rick) Arbour
Virgil J. Beaty
Joel E. Cahoon
John Clements
Roger Cronin
Thomas Demlow
John L. Dicmas
Rick A. Donaldson
Erik B. Fiske
Paul R. Gallo
Richard O. Garbus
James G. Gibbs, Jr.
Mayo Gottliebson
Alan W. O'Brien
Arthur Molseed
Ned W. Paschke
Duane M. Petersen
Douglas L. Schneider
Arnold R. Sdano
Otto Stein
Charles E. Sweeney
Alan Vause
William Wheeler
Theodore T. Williams

This chapter deals with wet well design, pump piping, and selection of pumping equipment. Developments within the last decade have had a profound effect on the design process, especially on the design of intakes of all types, and also on the considerations that should be addressed in pump selection. This chapter differs greatly from that in previous editions of this book because of these changes. While the changes affecting pump selection are many and significant, the changes associated with the design of pump intakes are the most far-reaching. For this reason, the chapter begins with a detailed discussion of pump intakes and their design. Based on experience, when the rules for the arrangement and geometry of intake

design as set forth in ANSI/HI 9.8 are not properly addressed, the results will most likely be unsuccessful. Further information on poor intakes and why they fell short of expected performance is located in Chapter 27. The pump selection process has been altered and improved by: (1) the advent of more informative data on pump performance and design, (2) more useful Hydraulic Institute standards, and (3) improved knowledge of pump cavitation phenomena.

Most of pages 351 to 357 in the second edition are omitted in this, the third edition. Those pages describe problems inherent in some common types of wet wells, and give reasons for them. They also indicate the hydraulic phenomena the designer should have considered and that should have led to changes to avoid the problems. Nevertheless, they are valuable and the reader should be familiar with the lessons learned as described in those pages.

References to a standard or code are given in abbreviated form such as ANSI/HI/9.8-1998. Specific sections may be referenced as, for example, ANSI/HI 9.8.2.2.3-1998.

12-1. Design of Pump Intakes

Pump inlet conditions are among the most overlooked and misunderstood aspects of pumping station design, yet they probably constitute the single reason most responsible for the success or failure of a pumping station. Pump performance is entirely dependent on the quality of the effort expended to ensure that adequate conditions exist at the pump intake. Regardless of the type of intake (whether pressurized, sump, or forebay), care must be exercised to avoid poor hydraulic conditions at the impeller. The pump intake design must satisfy the requirements for proper approach conditions by avoiding the following:

- Poor velocity distribution at the entrance to the pump
- Excessive swirling in the pump intake piping
- Air entrainment in the pumped flow
- Inadequate NPSHA at the pump inlet, especially at the most frequent operating conditions
- Unstable approach conditions in multiple pump operation
- Vortices.

Poor inlet conditions have a dramatic effect on pump and system performance in several ways, including:

- Pump cavitation and vibration that results in costly and frequently repeated repairs
- Loss of pump capacity, head, efficiency, and inability to achieve installation design objectives.

Considerable effort has been expended in recent years to improve the knowledge base for the design of pump intakes. Much of this effort is reflected in the requirements established by the publication of ANSI/HI 9.8-1998, "Pump Intake Design." The following paragraphs provide a background for the requirements in that document and additional material on developments since its publication.

12-2. Pump Intake Design Standards

In the past, designers of pump intakes relied more or less on recommendations by the British Hydromechanics Research Association [1] and early Hydraulic Institute Standards such as those published before 1998 [2]. Such publications, however, were confined to clean water, with discussions on solids-bearing waters limited essentially to the weak advice that flat floor areas should be minimized, that designs proven to be successful should be followed, or that pump manufacturers should be consulted.

In 1994, the Hydraulic Institute appointed a committee of pump manufacturers and for the first time invited users, engineering consultants, and researchers to expand and improve the standards for clean water and, also for the first time, to consider designs for solids-bearing waters in detail. A new and entirely different standard resulted and, after consensus by public canvassing, was adopted as the American National Standard for Pump Intake Design, ANSI/HI 9.8-1998 by both the American National Standards Institute and the Hydraulic Institute. Every designer of pumping stations should have and understand ANSI/HI 9.8-1998 plus all of the other pertinent ANSI/HI standards. They are available in hard copy or as CDs (see www.pumps.org, topic e-store).

Normative standards such as ANSI/HI 9.8 establish a criterion of care and a benchmark for engineers. A normative standard is one that establishes a level of performance acceptable to all groups. Engineering, application, and user groups at large have had the opportunity for representatives from their groups to comment on the standards, so they reflect (1) the opinions of all participants, and (2) the experience and defensible information on the subject. The formalized procedures used by organizations such as ANSI, ASME, ASTM, and NFPA are designed to make certain that all interested parties can be heard and that the product of the process (the standards)

represents the best available information at hand, devoid of bias toward any particular product or point of view. Individual committees are required to document their deliberations and to invite public comment on draft standards as a part of the process. Responses to public comment are documented and made a part of the record. The results, although not necessarily representing the cutting edge of available technology, do represent the minimum in acceptable performance, and that minimum should be considered when any project is developed. The owner can certainly decide whether the standard should be used, but the engineer has an obligation to inform the owner of the perils faced by nonconformance, including the prospect that (1) the installation may not perform as desired, and (2) warranties may be voided.

This book is complementary to and compatible with these standards, and the presentation in this chapter not only follows ANSI/HI 9.8-1998 but also reflects: (1) experience gained through more than 45 years of designing pumping station wet wells, (2) information developed at the U.S. Corps of Engineers Waterways Experiment Station in Vicksburg, Mississippi, (3) investigations at Montana State University at Bozeman, ENSR Hydraulic Laboratory in Redmond, Washington, Alden Research Laboratory in Holden, Massachusetts, and northwest hydraulic consultants in Seattle, Washington, and (4) full-scale tests at the Fairbanks Morse Pump Corp. plant in Kansas City—all of which have led to several publications [3, 4, 5, 6, 7].

Pump Inlet Conditions

Undesirable features noted in many sump designs include:

1. A free fall (no matter how short) from the inlet conduit into the sump or pool below with the consequent entrainment of air in the liquid and (with wastewater) the release of odors. The air bubbles, easily captured by currents and carried into the pumps, cause loss of capacity and damage to the equipment. Air discharged into pipes promotes sulfuric acid production, and in unprotected concrete pipes the combination leads to collapsing sewers.
2. Piping with excessive velocities that cause unreasonable headloss and can lead to vibration problems due to turbulence in fittings and valves.
3. Abrupt changes in flow direction upstream from the pump inlet connection. In sumps, abrupt changes usually cause vortices. In intake manifolds and pump inlet piping, abrupt changes in direction may cause flow to become asymmetrical and thus overload pump shafts and bearings. Abrupt changes in flow direction are acceptable when the pump manufacturer (1) supplies the fitting as a part of the equipment, or (2) does not take exception to the presence of the fitting. Pump operation is probably not adversely affected in such circumstances.
4. Sump or inlet piping geometry that permits differential velocities and, thus, rotation of the fluid. With the slightest rotation, the spin increases as the water approaches the pump suction inlet. Swirling in the suction pipe may reduce the local NPSHA in the core to zero and thereby cause cavitation, noise, and rapid wear even though the average NPSHA is adequate. Swirling at the impeller changes the angle of attack of the flow to the impeller blades and shifts the pump curve, often drastically.
5. Horizontal velocities in sumps near the pump inlets that are too high. In general, such velocities should be less than 0.3 m/s (1 ft/s). Actual velocities usually differ greatly from calculated average velocities.
6. Interference between adjacent intakes. Space intakes no closer than 2.5 *D* c-c (where *D* is intake bell diameter). Also consider access clearance [1.1 m (42 in.) minimum] between adjacent machines.
7. Discontinuities such as corners without fillets and uneven distribution of currents caused by flow past pier noses that often result in the formation of air-entraining vortices. Although there is usually no surface indication, subsurface vortices may also occur, and they can be very damaging.
8. Stagnant areas in wastewater pumping station wet wells where velocities are too low to prevent the deposition of the putrescible solids in wastewater. Velocities of about 0.3 m/s (1 ft/s) are enough to keep organic solids moving, whereas velocities in excess of 0.5 m/s (2 ft/s) are required to keep grit moving. After organic material and grit are deposited, velocities in excess of 1.6 m/s (5 ft/s) are required to move them with reasonable celerity.

See Section 27-6 for a discussion of wet well failures in which the designer did not follow the above advice.

The water velocity into pump intakes recommended in ANSI/HI 9.8-1998 is given in Table 12-1. However, the authors recommend the velocity be limited to about 1.1–1.2 m/s (3.5–4.0 ft/s). This rec-

Table 12-1. ANSI/HI 9.8–1998 Recommended Velocities for Pump Intake Bells, Based on Bell OD

SI Units			U.S. Customary Units		
Flow, L/s	Range, m/s	Recommended[a] velocity, m/s	Flow, gal/min	Range, ft/s	Recommended[a] velocity, ft/s
<315	$0.6 \le v \le 2.7$	1.7	<5000	$2 \le v \le 9$	5.5
315–1259	$0.9 \le v \le 2.4z$	1.7	5000–20,000	$3 \le v \le 8$	5.5
≥1260	$1.2 \le v \le 2.1$	1.7	>20,000	$4 \le v \le 7$	5.5

[a]The editors consider 1.5 m/s (5 ft/s) to be the near-maximum velocity desirable for both C/S and V/S pumps.

ommendation should be applied to runout (the flow rate at the least possible dynamic head).

12-3. Types of Pump Intake Basins

Several types of pump intake basins are briefly described together with advantages and limitations for use. Dimensions are given in terms of D, the outside diameter of the suction bell of the pump. The ANSI-HI 9.8 standard should be constantly consulted as this section is read.

Unconfined

Pumps are suspended (usually by a platform) in a body of water with no confining walls or other flow-guiding structures. Unconfined pump intakes are inexpensive and useful for lagoons, lakes, rivers, and canals. Ice, debris, and fish that must be screened ahead of the pumps are of particular concern and require special consideration.

Design

Details of requirements are given in ANSI/HI 9.8.2.7-1998. Submergence requirements in unconfined (and, indeed, in all types of) pump intakes are given in ANSI/HI 9.8.2.7.4-1998 and by Equation 12-1:

$$S/D = 1 + 2.3\ \mathbf{F} \qquad (12\text{-}1)$$

where S is submergence, D is outside diameter of the suction bell, and **F** is the Froude number (dimensionless) given by:

$$\mathbf{F} = \frac{v}{\sqrt{gD}} \qquad (12\text{-}2)$$

where v is the average velocity at the mouth of the suction bell, g is the acceleration due to gravity, and D is the outside diameter of the suction bell.

If the intake is in a current, consider open-top barrel pumps as exemplified in Example 18-6.

Rectangular

In plan, the shapes can be either rectangular or trapezoidal with the pumps lined up against one wall. They were common in the past and are still useful for clear water, especially for installations where a large volume of stored water is desirable, such as clear wells or large electricity-generating plants. Dividing walls (see ANSI/HI Figure 9.8.1-1998) between the pumps improve the flow and are required for pumps with design capacities exceeding 315 L/s (5000 gal/min). With a suitable combination of fillets, flow splitter, and baffle as shown in Figure 12-1 (a composite of ANSI/HI 9.6-1998 Figures A.11 and Figure A.10), the flow in the intake throat is excellent. The dividing walls between pumps must be very long (at least 5.75 D), however, and forebay design is critical. Rectangular pump intake basins are costly because of their size. They are entirely unsuitable for wastewater because the large flat floor and the low velocity (< 0.5 m/s or 1.5 ft/s) cause deposition. With screens at the head of each channel they can be used for water that carries vegetation.

Design

Design is simple but the layout, dimensions, velocities, and pump intake submergence (given in ANSI/HI 9.8.1.4-1998) must be followed.

In open, rectangular wet wells (without partition walls between pumps), circulation patterns are likely to result in excessive swirling in the pump intakes. Swirling can be reduced by anti-rotation baffles between the pump intake and the adjacent wall and

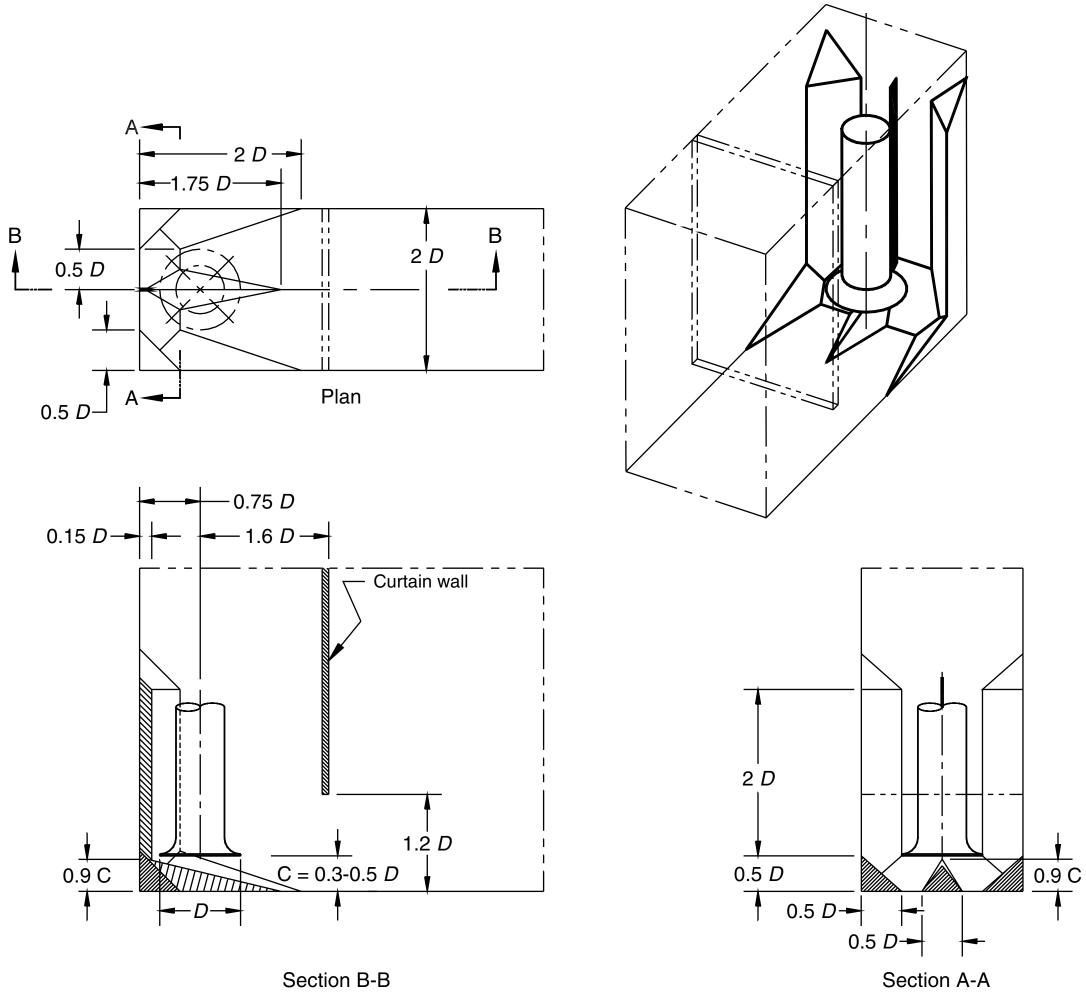


Figure 12-1. Details for best performance in a single bay of a large rectangular wet well. Each bay bounded by side walls at least 5.75 *D* long. After ANSI/HI 9.8, courtesy of the Hydraulic Institute [3].

nearly eliminated by dividing walls. One alternative is a wall that extends from 25 mm (1 in.) or so above the floor to about 0.3 m (1 ft) below LWL so that the pumps are confined within a trench. See Figure 27-1. Another alternative is the use of submerged barrel intakes as shown in Example 18-6.

Trench-Type

Trench-type wet wells (see Figures 12-2 and 12-3) are unique in that pump intakes are confined in a deep, narrow trench and in line with (but substantially lower than) the upstream inlet pipe as shown in the figures. Their advantages for all waters are: (1) a good hydraulic environment for the pump intakes; (2) minimum footprint size for wastewater; (3) a small floor area with consequent minimum accumulation of sludge; and (4) ease and rapidity of cleaning by dewatering the wet well (pump-down), thus allowing the water to flow down the ramp and along the floor at high velocity to the last pump. Sludge and scum can be removed without manual labor in five minutes or less after the start of pump-down. Trench-type wet wells are suitable for design flows exceeding about 130 L/s (3 Mgal/d). Some larger than $4.5\,m^3/s$ (100 Mgal/d) have been built. See also Sections 12-6 and 12-7.

The triangular fins under the suction bells in Figure 12-2 reduce floor vortices substantially but not quite enough to meet the stringent acceptance criteria in Section 3-12. Those criteria are, however, met with the installation of the flow splitter shown in Figure

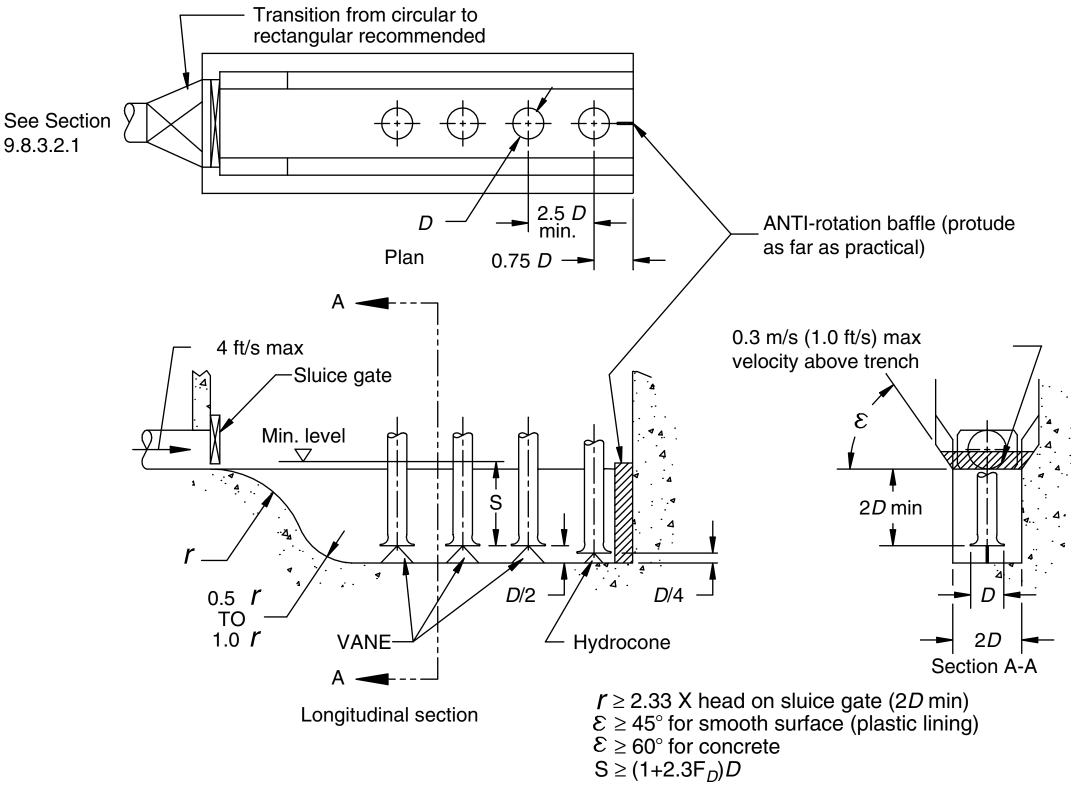


Figure 12-2. An early trench-type wet well for solids-bearing water and variable-speed pumps. Figure 9.8.13 from ANSI/HI 9.8–1998. Courtesy of the Hydraulic Institute.

12-3. The continuing evolution and progress in the design of pumping stations is illustrated in the above figures and in the discussion in Section 12-5.

Design

See Sections 12-5, 12-6 and Examples 12-1 for V/S pumps and 12-2 for C/S pumps. Constant-speed pumping requires temporary (or active) storage of water while pumps are resting. Typically, the water level rises and falls about 1.2 m (4 ft). To prevent a cascade and to increase the storage capacity of these compact wet wells, use an approach pipe at a 2% gradient for about 60 m (200 ft) from an upstream manhole. An approach pipe is designed in Example 12-2.

Circular

Circular wet wells are appealing because of their low cost. Small ones can be made of concrete pipe, but FRP and steel are also used. They can be used with dry pit pumps (especially self-priming units), but they are typically used with submersible constant-speed pumps. Although flat floors as in Figures 9.8.4 and 9.8.5 in ANSI/HI 9.8-1998 are acceptable for clean water, side wall vortices might occur. Vortices can be prevented by adding fillets or by confining the intakes in a can (Figure 11-29) at least 2*D* high. However, either hopper bottoms or an adequate means to mix and blend scum and sludge are needed for cleaning wastewater wet wells. (See Figures 17-21 and ANSI/HI Figures 9.8.15 and 9.8.16-1998). One pump manufacturer [8] offers a hopper bottom that contains the discharge fittings for submersible pumps as well as submersible pumps that bypass part of the discharge for a short period into the wet well to mix the contents and thus keep the wet well clean.

Design

Many small circular wet wells have serious deficiencies. One of the worst features is inflow from the sewer that cascades into the pool below, causing turbulence that

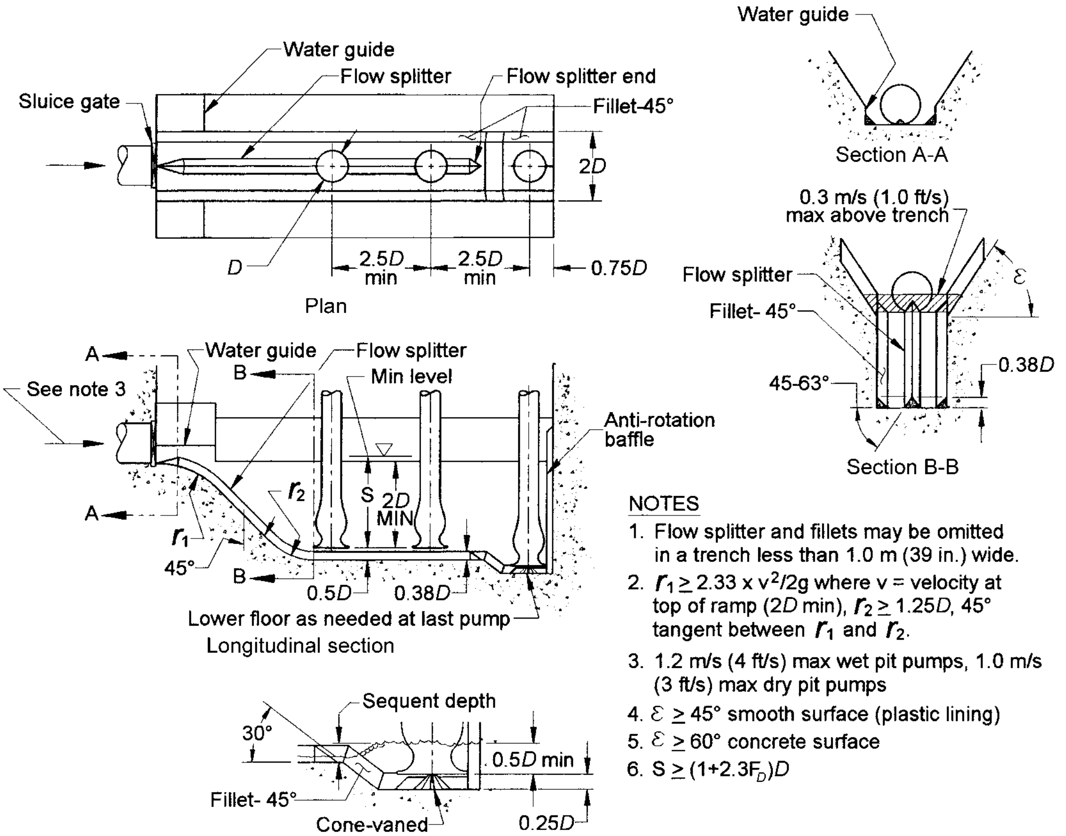


Figure 12-3. Modern trench-type wet well for solids-bearing water and variable speed pumps. Figure 9.8.13 from ANSI/HI 9.8-2004. Courtesy of the Hydraulic Institute.

releases sewer gas, air bubbles, and strong currents that drive the bubbles into the pump intakes. Air ingestion reduces pumping efficiency, head, and capacity, and reduces the life of bearings and seals. Some designers use a drop pipe or a drop manhole to avoid air ingestion by pumps, and that is much better, but sewer gas is still released. Other deficiencies include: (1) large, flat floor areas where sludge can accumulate and putrefy, and (2) lift stations that are expensive to clean. These features persist because of inertia and because the facilities are too small to justify model tests. It is quite possible to design economical lift stations that are self-cleaning and in which wastewater enters gently and without air bubbles. See Section 12-8 for details.

Cans or Barrels

These are for vertical column-type pumps mounted within a can or barrel as shown in Figures 11-29 and, in ANSI/HI 9.8-1998, Figures 9.8.10, 9.8.11, and 9.8.12. They are excellent and relatively inexpensive for pumping clear water, especially for booster pumping. The can may be either open at the bottom or closed. One or more vertical, anti-rotation baffles between the pump and the surrounding barrel can eliminate the fluid rotation that leads to swirling in the pump intake. Cans are customary for column pumps of all sizes. They are also useful for eliminating the need for dividing walls in (and reducing the size of) rectangular basins for large column pumps. Model studies are required for such configurations, particularly for pumps with discharge nozzles 500 mm (20 in.) and larger.

Design

For cans open at the bottom, some form of intake to prevent swirling and to produce nearly uniform flow is necessary. Model studies are required by the ANSI/HI standard if the pumped flow exceeds 3.2 L/s (5000 gal/m). A cross at the bottom of closed can prevents cross-current flow. Keep intake velocities below 1.2 m/s

(4 ft/s). Keep at least five pipe diameters between any obstruction such as a butterfly valve and the can. However, one pipe diameter is sufficient for a gate valve. To avoid conflict, the pump manufacturer should furnish the cans.

Submerged Barrel Intakes

A modified form of can or barrel intake consists of a submerged weir surrounding the pump column. The weir section extends downward to the floor to form the barrel (which contains 8 vanes to eliminate cross currents and swirling) as shown in Figure 18-29 of Example 18-6. With a cone at the bottom, vortices are eliminated, and velocities in the throat of the pump are well within ANSI/HI 9.8-1998 recommendations. Currents in the basin have no effect on the pump, and, therefore, it is an almost perfect intake for clear water (or even secondary or primary effluent) in existing basins that require retrofitting to obtain good performance, in channels where currents are strong, or in new basins where size can be reduced by eliminating the partition walls detailed in ANSI/HI 9.8-1998 Figure 9.8.1.

Design

At this writing, the submerged barrel intakes have not been included in ANSI/HI 9.8. Model studies are required for pumps with capacities greater than 315 L/s or 1100 m^3/h (5000 gal/min). The design is very simple, however, as described in Example 18-6.

Confined

The layout is shown schematically in Figure 12-4 and in detail in Figure 9.8.17 in ANSI/HI 9.8-1998. Its advantages are: (1) superlative hydraulic environment for the pumps during normal operation; (2) ejection of sludge any time a pump is running at full speed; (3) ejection of scum at pump-down; and (4) adaptability to large pumps. On the other hand, its disadvantages are: (1) the stored fluid volume is too small (without auxiliary storage) for C/S pumping; (2) the danger that a cell might fill with sludge and grit and clog the pump if the pump is out of service for considerable time; and (3) greater cost than circular wet wells.

Design

An approach pipe can solve the storage volume problem described above, just as it does for trench-type

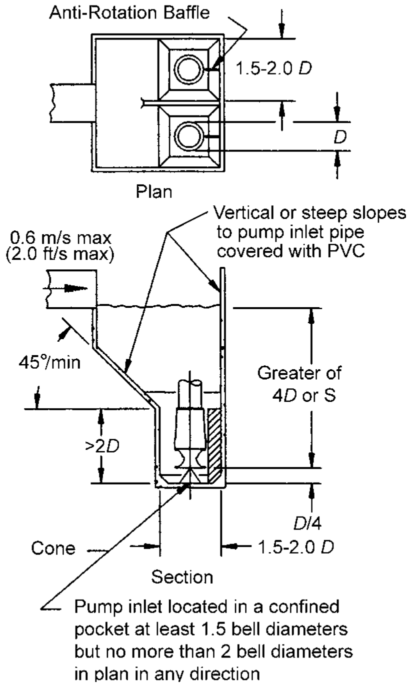


Figure 12-4. A confined wet well. Figure 9.8.17 in ANSI/HI 9.8-1998. Courtesy of Hydraulic Institute.

sumps. The danger of the cell filling with sludge or grit can be overcome by means of a backwash pipe in one corner to fluidize the contents, which can then be safely pumped out. The backwash pipe can usually be supplied from the discharge manifold.

Formed Suction Intakes (FSIs) for Large Pumping Stations

Draft tubes (FSIs) are useful for pumps requiring intake piping larger than 400 mm (16 in.) because eccentric plug valves for isolation in such sizes are so massive and expensive, whereas sluice gates, which are light and relatively cheap, are appropriate for draft tubes. Knife gate valves can also be used for isolation, but note that some engineers reject such usage, so be cautious in using them by investigating similar situations. FSIs can be used with either trench-type or rectangular wet wells.

U.S. Army Type 10 FSIs

Two styles of FSIs were developed by Fletcher [9] at the U.S. Army Engineers Waterways Experiment Station in Vicksburg, Mississippi. The U.S. Army FSI

Type 10 (having the lowest profile) is shown in Figure 9.8.3 in ANSI/HI 9.8-1998. Army Engineers' studies of a model with a throat diameter of 200 mm (8 in.) showed that flow may approach the intake at any angle up to 90 degrees and that forebay velocities of 1.2 m/s (4 ft/s), sometimes twice as much, caused no problems, according to Fletcher [10]. The velocities in the throat met all requirements for uniformity and lack of swirling or vortices if the throat velocity was limited to the value obtained with Equation 12-3.

$$v \leq 1.99\ (gD^5)^{0.5} \tag{12-3}$$

where v is velocity and D is the throat diameter.

Caution regarding high forebay velocities is advised regardless of the sweeping statement above. Not every design with FSI intakes works equally well, and the configuration of the forebay does affect swirling and vortices, according to Fletcher [11]. One disadvantage of the Type 10 FSI is the difficulty (and hence, high cost) of constructing it. A second disadvantage is the flow separation that occurs at the abrupt inlet and, under some operating conditions, may persist as full vapor core vortices to the pump. Model studies of a very large storm water pumping station indicated that the flow separation could be eliminated by increasing the entrance radius from 0.02 d to 0.2 d or 0.3 d where d is the diameter of the outlet. Adding horizontal grating baffles or, for water containing trash, an array of steeply sloping pipes above the intake eliminates surface vortices, but model studies are required.

Compact FSI

The compact FSI shown in Figure 12-5 originally had a flat floor with flow splitter, fillets, and vertical guide vanes for producing uniform influent currents. The compact FSI is designed for stations in which there are high cross-flow velocities, up to 1 m/s (3 ft/s). The FSIs for Tres Rios are submerged in individual cells so that a small gap between the pump suction bell and FSI need not be watertight. The performance of the original design did not fully meet all the acceptance criteria in ANSI/HI 9.8, so the design was modified as shown in Figure 12-5 with floors at two slopes to produce more uniform currents in the throat of the suction bell. Two horizontal vanes were added to break up vortices that originated in the forebay channel. The performance of the modified model was good. Even when fabricated of stainless steel, its cost is moderate—certainly less than the cost of the U.S. Army Corps of Engineers Type 10 FSI pictured in the second edition of *Pumping Station Design*.

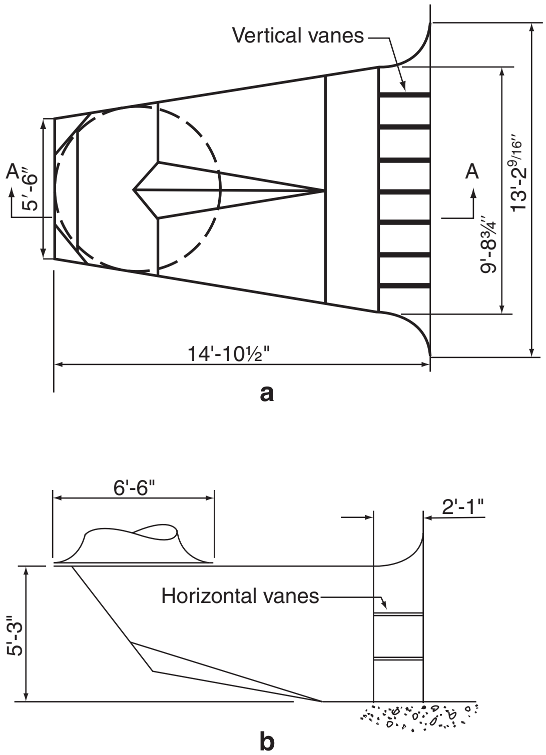


Figure 12-5. Proposed 110 Mgal/d FSI at Tres Rios secondary effluent pumping station. (a) Plan; (b) Section A-A. Courtesy of Northwest Hydraulic Consultants, Damon S. Williams Associates, U.S. Army Corps of Engineers, and City of Phoenix Water Services Department.

The guide vanes would be unsuitable for raw wastewater and would be replaced with fewer guide vanes with well-rounded noses to shed stringy material. Werth and Cheek [12] have described model tests of several similar designs.

Swan Island Pumping Station FSI

The Swan Island Pumping Station (SIPS) is a 4.4-m^3/s (100-Mgal/d) facility at the end of a long CSO tunnel about 30 m (100 ft) below grade. The large pumps are served by extremely simple but very effective FSIs made by splitting a 900-mm (36-in.) stainless steel pipe lengthwise into two pieces, separating them at the upstream end, and welding them to a long radius reducing elbow at the downstream end. Two triangular plates with base widths of 675 mm (27 in.) were welded between them. The designers were concerned that a single, vertical guide vane (used to inhibit cross-currents and swirling) would collect rags, so the guide

vane was split with a top part and a bottom part separated for a clear path in the middle. The noses were slanted downstream at 45 degrees so that any stringy material would wash away.

The FSIs were installed in a trench-type wet well with their floors level with the top of the fillets in the trench. The use of fillets is important to keep swiftly moving water (during cleaning) confined to a smooth, prismatic channel. Except at the floor, the mouths of the FSIs were rounded to a radius of 225 mm (9 in.). The bell mouth at the entrance can be fabricated of steel, but it is less expensive to terminate the steel 0.25 *D* short of the trench face and form the bell mouth of concrete. Model tests showed that this design passed the ANSI/HI 9.8 standards.

FSIs: Conclusions

Other styles of FSIs such as those in Figures 17-6 and 17-9 are also effective.

The pump can be isolated from the wet well with a sluice gate at the FSI mouth. A knife gate valve in the round pipe section can also be used, but first carefully investigate problems of malfunction, trash in the seat, and leakage.

Horizontal inlets, particularly large ones, sometimes cause large surface vortices that persist into the inlets even when the submergence exceeds ANSI/HI requirements (Equation 12-1). A grating submerged immediately below the LWL prevents such vortices in clean water applications. For wastewater use, the grating can be made of a single (sometimes a double) row of epoxy-coated steel pipe or stainless steel tubing. The tubing can be installed on the sluice gate. Solids slide off if the slope of the pipes is inclined at 45 degrees or more. (This discussion is continued with respect to Figure 12-13 in Section 12-5.) A model test can be used to determine whether the grating intercepts and breaks up the vortex.

To obtain reasonably uniform velocities ($v_{min}/v_{avg} > 90\%$) at the pump impeller, the nominal velocity at the exit from a reducing elbow (whether of short or long radius) must be at least twice the entrance velocity, so the reduction must be no less than 3×2 or multiples thereof.

Good flow patterns and lower pump elevations can be obtained by using a modified bend similar to the Turbo-Free® elbow in Figure 12-6.

The Turbo-Free® elbow, developed in 2003, exceeds the ANSI/HI 9.8-1998 requirement for uniform exit velocities, whereas either short- or long-radius (short is slightly better) reducing elbows do not unless the area reduction ($1 - A_{in}/A_{out}$) exceeds 0.5. On the basis of limited tests, the headloss appears to be

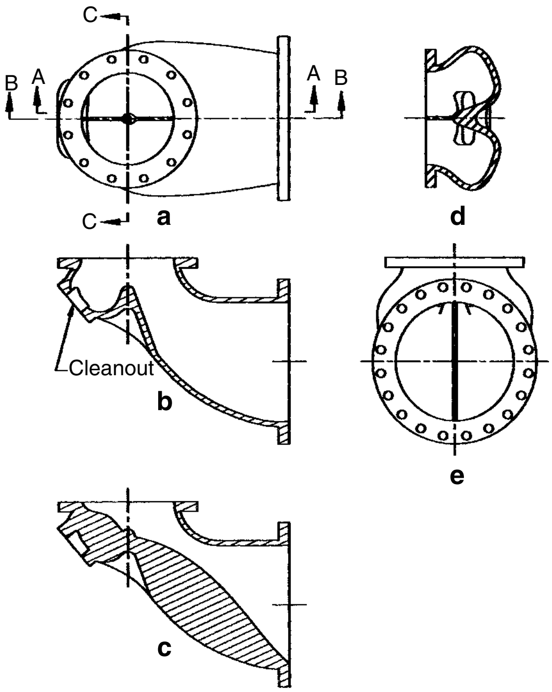


Figure 12-6. Turbo-Free® 90° reducing bend for producing uniform fluid velocity at exit. (a) Top view; (b) Section A-A; (c) Section B-B; (d) Section C-C; (e) end view. Courtesy of Fairbanks Morse Pump Corp., Pentair Pump Company.

about 20% more than that in a short-radius elbow. The headloss is, however, of minor importance as compared to the effect of a superior flow pattern in extending the time between repairs to the pump.

Corrosion

Corrosion is a serious problem, so if mild steel is used, the FSI must be coated inside with inert material such as epoxy. Fabricated stainless steel costs about 50 to 85% more than fabricated mild steel, but it needs no coating. Considering the high cost of installation (equal for both steels) and the exorbitant cost of possible future repairs to epoxy coatings, stainless steel is the more attractive alternative. Suitable stainless steels include types 316L and 347.

As shown by recent experience, however, microbially-induced corrosion can occur with both stainless and carbon steel FSIs if the pumped fluid is rich in organic material (as with wastewater applications) and stagnant conditions prevail. To avoid this concern, consider fusion-applied epoxy linings or provide a means for replacing the wastewater with fresh water

if it is to be out of service for prolonged periods. See Section 25-12 for further discussion.

Design and Installation

A good deal of care must be exercised in the details required for installation of draft tubes, especially if they are to be encased in concrete. External ribbing is recommended for sufficient rigidity during fabrication, transport, and installation. Removable internal bracing is required during the placement of concrete and means must be provided to grout behind and underneath the draft tube after the concrete has cured to ensure that no voids remain after concrete placement. Accurate location of the draft tube with respect to the planned location of the pump is of the greatest importance. Chairs and adjustment provisions should be designed into the draft tube assembly to permit precision placement of the draft tube after a base slab has been poured to within about 0.25 m (9 in.) of the planned invert for the draft tube. Because the equipment is large, precision alignment between drives and the pump is necessary. To aid in this task, use a separate connection piece at the draft tube exit as shown in Figure 12-7. This connection provides sufficient flexibility to allow precision leveling and minute adjustments in the pump position prior to welding the connection piece to the concrete-encased draft tube.

Manifold or Header

Intakes from manifolds or headers are useful where the manifold is under enough pressure to provide sufficient submergence at all times for the pump impeller, but only if the header fluid velocity is less than 2.4 m/s (8 ft/s). See Figure 12-46 and Figure 9.8.10 in ANSI/HI 9.8-1998. Because there is no expensive basin, such pump intakes are very cost-effective indeed. Any invasion by crustaceans, however, creates an especially difficult problem because of lack of easy access.

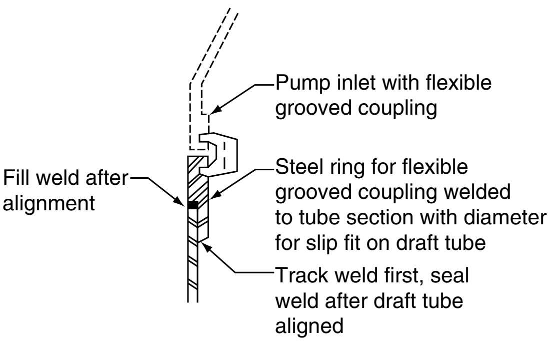


Figure 12-7. Connection device for precision positioning of the pump.

Design

Follow the recommendations in ANSI/HI 9.8-1998 Section 9.8.4.3.

Suction Tank

Suction tank intakes are used for pumping solids-free liquids to receivers such as water-bearing fire trucks, milk transport, gasoline, and the like (see Figures 9.8.8 and 9.8.9 in ANSI/HI 9.8-1998). There are a great many different conditions in the number and placement of outlets, simultaneous inflow and outflow, pressure (less or more than atmospheric), and the like.

Design

Follow the recommendations in ANSI/HI 9.8-1998 Section 9.8.2.5 wherein almost every conceivable condition is covered.

12-4. Model Study

Model studies are useful for finding and correcting faults in the hydraulic flow patterns in wet wells. Although expensive, the cost rarely exceeds 1 or 2% of the total cost of a pumping station—a small price to pay to verify the performance expected or to modify the design as necessary. The authors' experiences with such studies have demonstrated on several occasions that even with considerable experience with intake design, there is no substitute for the results of a carefully conducted, comprehensive physical model study, especially if the design in question is large or deviates in some material respect from the designs presented in ANSI/HI 9.8.

Need For Model Study

The ANSI/HI 9.8-1998 standard requires a physical model study if:

- The sump or piping geometry differs from the standard.
- The approach flow is non-uniform or unsymmetrical.
- The flow rate is more than 2520 L/s (40,000 gal/min) per pump or the total flow exceeds 6310 L/s (100,000 gal/min) with all pumps running.

- The flow rate in a pump of open bottom barrel or riser arrangement exceeds 315 L/s (5000 gal/min).
- Operation is critical and financial loss due to repair, remediation, or failure would exceed 10 times the cost of model study.

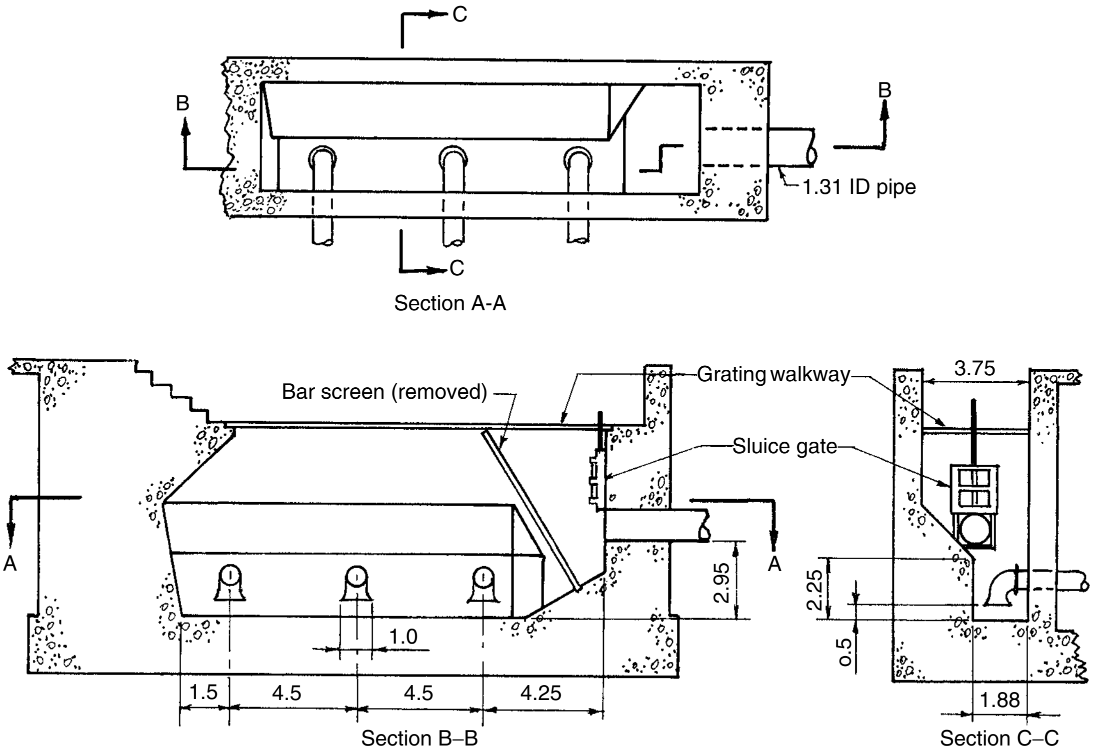
ANSI/HI 9.8 contains acceptance criteria for model studies. See also Section 3-12 for a discussion and illustrations of vortices.

12-5. Evolution of Trench-Type Wet Wells

Trench-type wet wells were invented in 1964 by Caldwell [13], who reasoned that (1) V/S pumps require only a benign hydraulic environment for the suction intakes; (2) wet wells (for V/S pumps) do not need appreciable storage; and (3) dewatering a narrow trench would wash sludge and scum to the last pump for discharge. This configuration (but without a ramp) has also been successfully used for clear water. Until the almost simultaneous publication of the second edition of this book and the ANSI/HI 9.8 standard in 1998, however, such wet wells were scarcely known and were essentially used by only one consulting firm. Since 1998, they have become popular and widely used. The following several paragraphs present a great deal of information on the design of trench-type wet wells because very little information on design details has been made available in most references. Design of trench-type wet wells is somewhat more complex than the corresponding effort required for other intake types. Design tools have been developed and are presented here to aid in the process. This intake design has been recognized in the industry as one that, in many ways, is superior to available alternatives.

Original Wet Wells

Twenty-seven trench-type wet wells were constructed in the mid-1960s for Seattle Metro (now King County Department of Environmental Services). All were geometrically similar to Kirkland Pump Station (Figure 12-8), which was chosen as the prototype because at a station flow rate of 240 L/s (5.5 Mgal/d), it is representative in size and because, being located in downtown Kirkland, Washington, it must be kept odorless. Pump intake velocities based on the OD of the suction bell were kept between 0.9 and 1.2 m/s (3 and 4 ft/s). These stations have proved to be far superior to other



All dimensions are multiples of *D* (OD of bell mouth) 406 mm (16 in.).

Figure 12-8. Kirkland Pump Station, a typical design for original trench-type wet wells.

designs of that time, the pump components have been long-lived, and the pumps have shown no signs of distress from the operating conditions.

During a tour of several of these wet wells in April 1992, Sanks [5] discovered that currents during pump-down were slower than anticipated because large, air-breathing vortices beside the last intakes prevented lowering the water level to the lips of the suction bells. At Kirkland, the last pump usually broke suction at a bell submergence between 0.25 *D* and 0.55 *D*. On the basis of a clean trench, the currents during cleaning averaged only about 0.27 m/s (0.9 ft/s)—too low to move grit. However, scum was completely removed via the vortex. A survey of Kirkland Pump Station (September 1992) made before cleaning indicated an average sludge depth of nearly 100 mm (4 in.). Cleaning removed most of the fresh, putrescible sludge and left a hard, consolidated, stable sludge layer averaging a little less than 50 mm (2 in.) thick. In spite of the remaining sludge, the cleaning procedure essentially eliminated odors.

During pump-down for cleaning, the influent splashed into the pool of water with nearly complete loss of energy. It was evident that if the water surface were lowered by lowering the last pump intake and the influent were to flow down a curving ramp, the currents along the trench would be much higher and the cleaning process more efficient.

Tests with Sand

To obtain an estimate of the effect of fluid velocity on sand beds, a 1:1 scale model of a portion of the trench floor was built in the Montana State University Hydraulics Laboratory. Tests with river sand to represent grit resulted in Figure 12-9. From that figure, it is apparent that velocities exceeding 1.5 m/s (5 ft/s) are required to move sand deposits with any practical celerity.

In a 1:3.3 scale model of the Kirkland Pump Station tested with screened and washed sand to represent sludge, only a portion of the sand was ejected at pump-down until equilibrium occurred at model flow rates corresponding to the flows customarily occurring in the prototype during cleaning. Even at the peak flow rate of Pump 3, it required 22 min to scour all but a small amount (about 5 mL) of sand from the trench.

Development of Improved Trench-Type Wet Wells

A curved ramp from the inlet pipe invert to the floor was installed in the model, the last pump was lowered to a floor clearance of *D*/4, and an anti-rotation baffle was installed between the last pump intake and the end wall. The resulting conservation of fluid energy produced a very high velocity at the foot of the

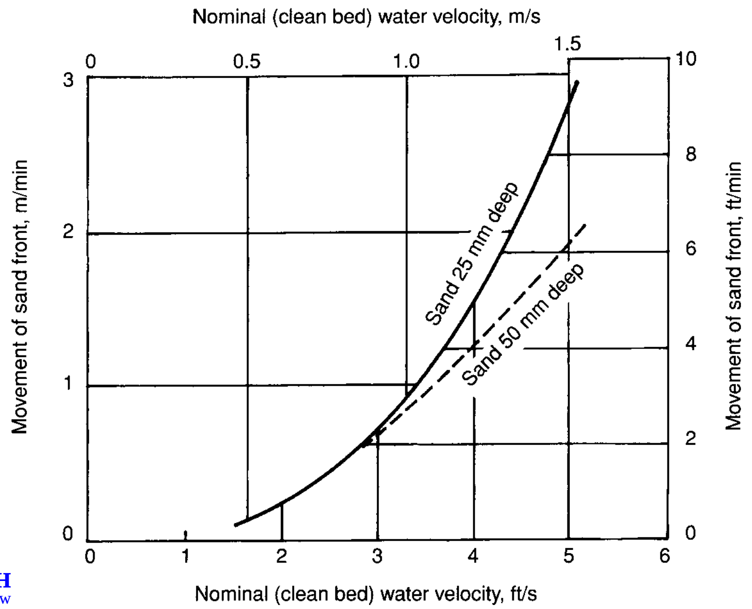
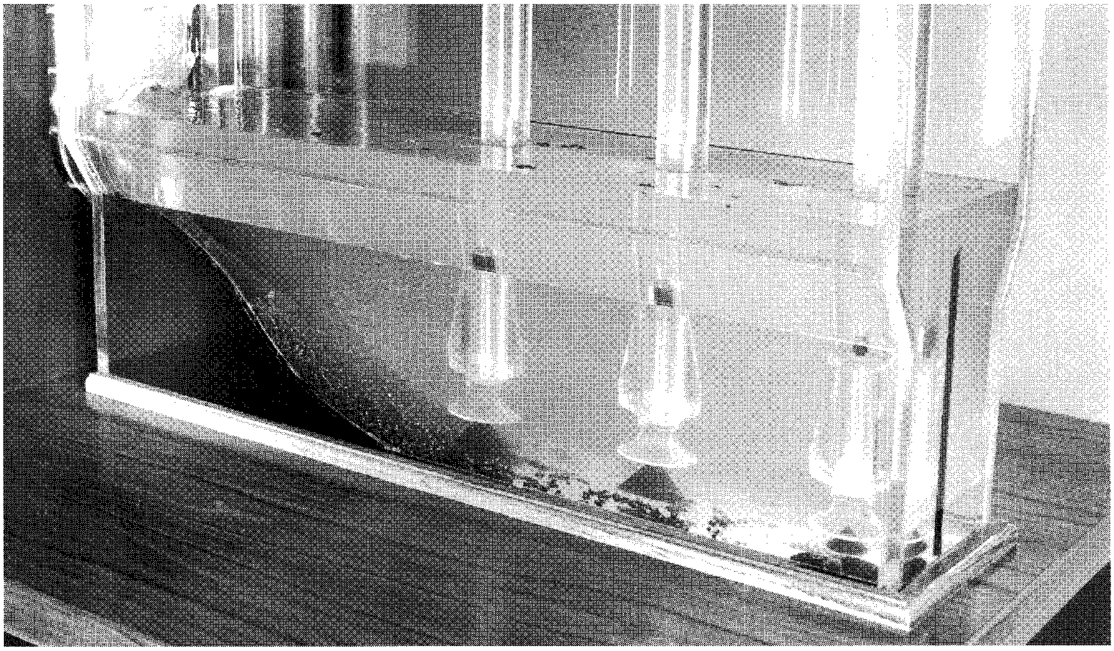
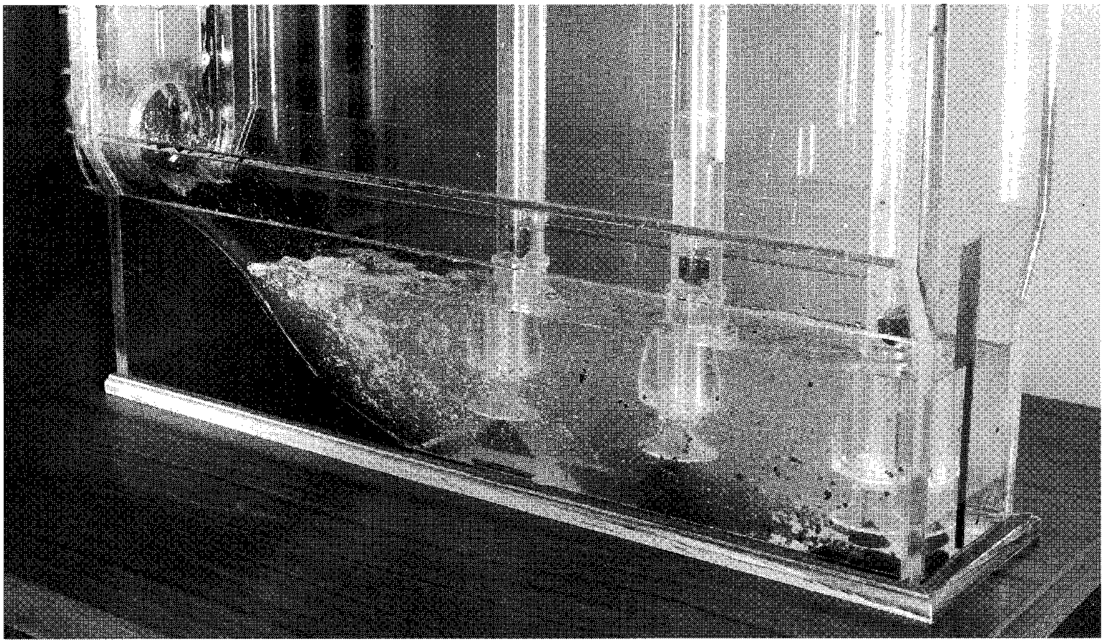


Figure 12-9. Average rate of sand movement as a function of fluid velocity. From Sanks, Jones, and Sweeney [6].

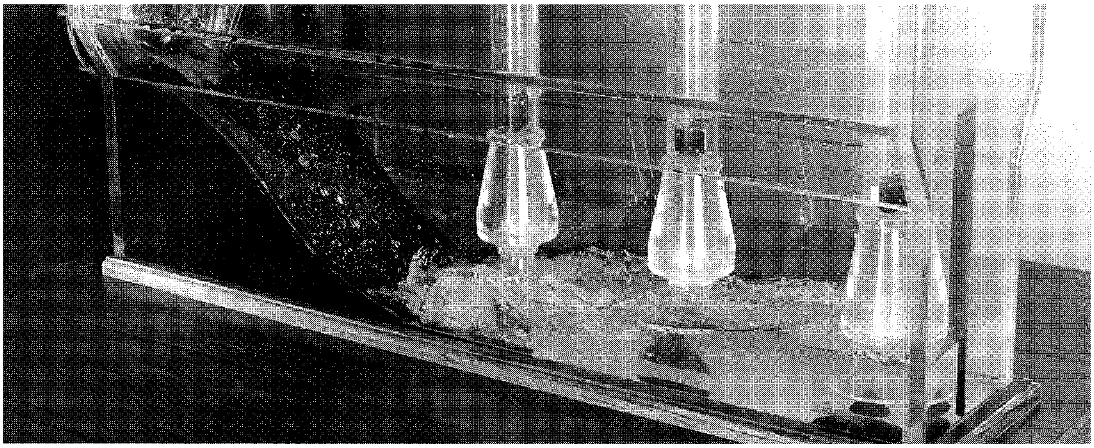


a

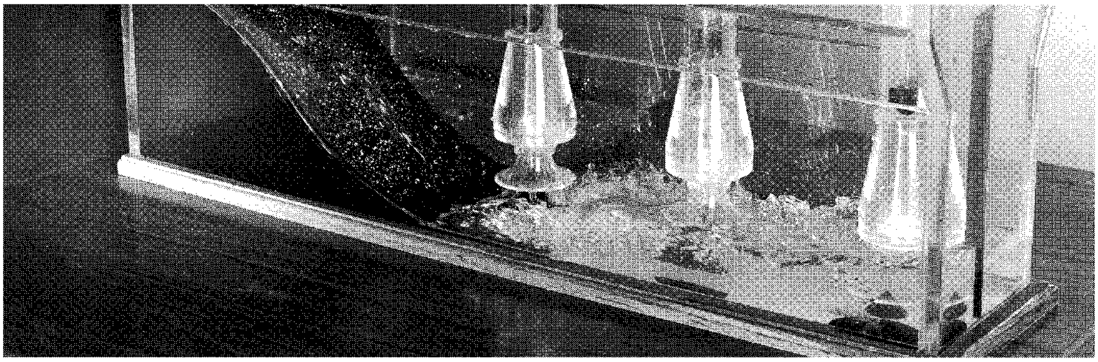


b

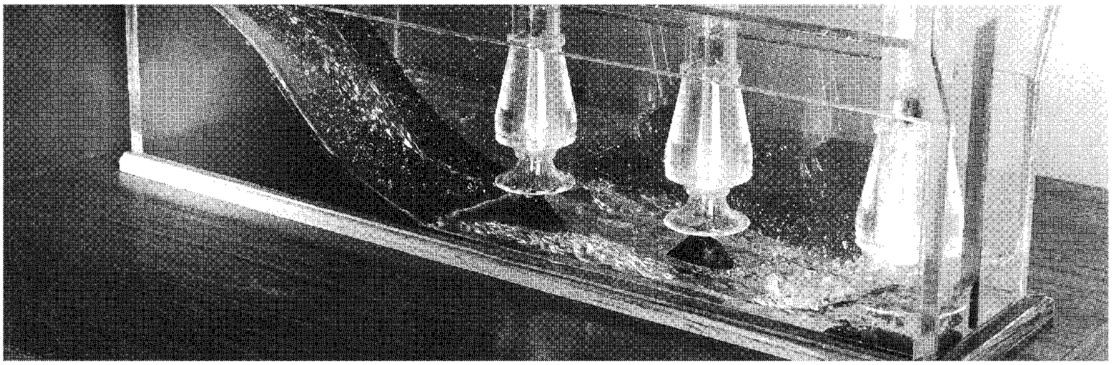
Figure 12-10. Demonstration model of a trench-type, self-cleaning wet well. (a) Before pump-down with water level at top of influent conduit; (b) pump-down begins. Hydraulic jump on ramp. Courtesy of Fairbanks Morse Pump Corporation.



a



b



c

Figure 12-11. Stages in the pump-down of the model in Figure 12-10. (a) Hydraulic jump at the toe of the ramp. The model "sludge" (represented by the black particles) has been ejected, but in a prototype, most of the grit deposits would remain; (b) hydraulic jump under Pump Intake No. 1; (c) jump in front of Pump Intake No. 3 and end of pump-down. Except for grease on the walls above the trench, a prototype would now be clean. Courtesy of Fairbanks Morse Pump Corporation.

ramp. A bonus was the creation of a hydraulic jump, which instantly suspended all of the solids within its influence and would, of course, homogenize the scum. The effectiveness of the ramp is clearly demonstrated in Figures 12-10 and 12-11. As cleaning then required less than half a minute, the improved model was between 45 and 100 times more effective than the original Kirkland model. Kirkland Pumping Station is thus the genesis of the modern trench-type wet well.

All of the development work to 2005 has been based upon a maximum pump inlet velocity of 1.3 m/sec (4 ft/sec). Based upon a limited number of studies that indicated adverse conditions at higher velocities, the authors do not recommend exceeding that value.

Two other models with geometry similar to Figure 12-2 were constructed (each to a linear scale of approximately 1:4), one at Montana State University (MSU) and one (under contract to MSU) at ENSR Consulting and Engineering Laboratory in Redmond, Washington. Tests between 1991 and 1995 were reported by Sanks, Jones, and Sweeney [14]. Beaty [15], assisted by Jones and Sanks, tested a full-scale wet well with sand (to represent grit and sludge) at the Fairbanks Morse Pump Corporation in Kansas City, Kansas. This pioneering research on trench-type pumping stations was funded by the U.S. Environmental Protection Agency, the Gorman-Rupp Co., Fairbanks Morse Pump Corp., ITT Flygt AB, Montana State University, Brown and Caldwell, and R. L. Sanks. The MSU model is still being tested at the time of this writing.

Because acceptance of a new type of wet well would no doubt depend on some form of official sanction, in 1993 Sanks requested that the Hydraulic Institute include it in its next edition of *Pump Intake Design Standards*. The Hydraulic Institute responded by creating the committee described in Section 12-2 that produced ANSI/HI 9.8-1998 in which provisions for cleaning became a requirement for solids-bearing waters. Trench-type wet wells and round wet wells with hopper bottoms are the only types specifically included for such waters, although any type can be used if adequate cleaning provisions are included in the design. Within less than a decade, trench-type wet wells have become widely used, and several large utilities now require them for intermediate to very large flow rates.

In 1998, the state of the art was represented by the design shown in Figure 12-2, but the acceptance criteria for model tests in ANSI/HI 9.8.5.6-1998 are so stringent that further improvements are virtually always needed when model tests are made. Uneven velocities in the throats of intakes have not been a problem. The phenomena that usually require better control are swirling and underwater vortices. For example, the flat, triangular vanes (oriented coaxially with the trench) under the upstream suction bells in Figure 12-2 do reduce floor vortices substantially but not enough to comply with the standard. Hence, other kinds of improvements are needed.

If model tests are not made, the acceptance criteria do not apply and designers are free to choose, for example, whether the expense of improvements is more cost-effective than that of using more robust pumps and cavitation-resistant impellers. The original wet wells (e.g., Kirkland) described in Section 12-3 have none of the improvements described below and yet have performed admirably for nearly four decades. Jones et al. [16] have also discussed improvements.

Improvements. Normal Operation

The improvements discussed above include influent baffles, vanes in suction bells, floor cones, and floor cones with vanes, vortex breakers, fillets, and flow splitters. See also the following subsections: "Improvements. Cleaning Operations" and "Fabricating Improvement Devices and Facilities." They are important; read them carefully. Then read them again.

Uneven Distribution of Throat Velocities in Pump Inlets

A velocity at any point in the throat must not vary from the average by more than 10%. Meeting this standard, however, does not seem to be a problem for trench-type wet wells.

The velocity from the influent conduit should not exceed 1.2 m/s (4 ft/s) in trench-type wet wells with column or submersible pumps, but for wet pit-dry pit pumps (where the suction bell is attached to an elbow), the velocity into the wet well should not exceed 0.9 m/s (3 ft/s). Furthermore, the influent pipe upstream should be coaxial with the pumps and straight and free from fittings that can disrupt the flow uniformity for a distance of eight pipe diameters. The current from the influent conduit tends to travel to the back wall as a jet with surprisingly little abatement. It impacts the back wall, dives, and returns along the floor to the ramp where it moves upward and joins the incoming jet. Now, a high velocity past a suction bell can lead to swirling and high differential velocities in the throat. Hence, reducing the velocity of the entrance jet (by using a larger conduit or by installing an influent baffle) reduces the velocity along the floor and mitigates the problems.

A horizontal baffle extending from side to side with the bottom about even with the influent invert and the

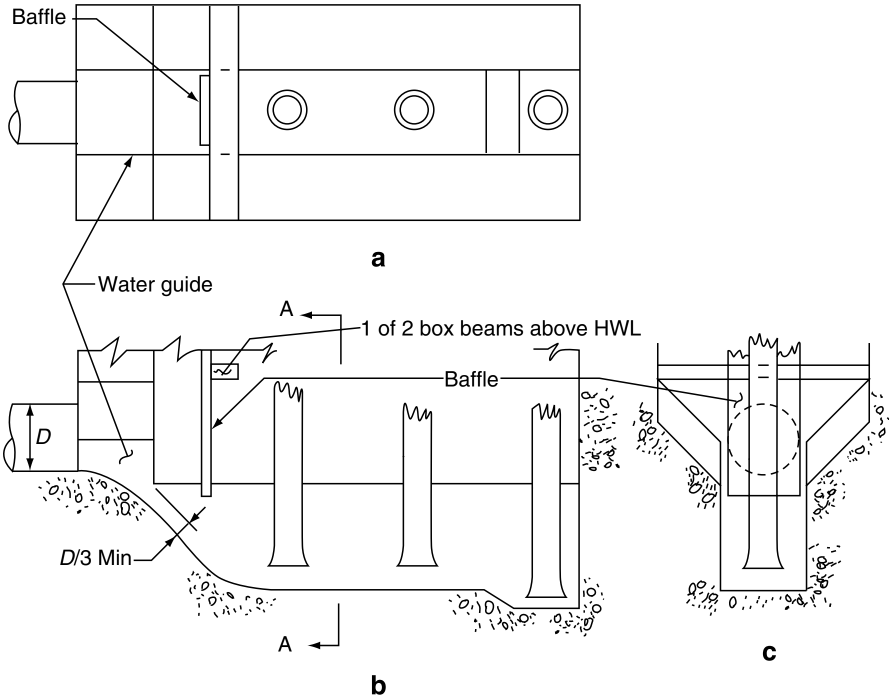


Figure 12-12. Influent baffle in a wastewater, trench-type wet well. Flow splitter and fillets not shown. (a) Plan; (b) longitudinal section; (c) Section A-A.

top below HWL has been modeled with success, but it is very sensitive to location. The vertical baffle in Figure 12-12 is more effective, is self-cleaning, does not trap scum at any stage of flow, and is less sensitive to location. However, the best geometry for, say, a middle pump alone may not be optimum for that pump if a second pump is running. The best geometry and location are obscure due to the many combinations of widths, depths, and downstream distances possible. On the other hand, the baffle significantly improves the flow regardless of geometry. The geometry depicted in Figure 12-12 is reasonably good. The baffle, 1.67 *D* wide with its bottom *D*/2 below the inlet invert, reduces floor currents dramatically.

Surface Vortices

Surface vortices are rarely a problem in trench-type wet wells if the submergence requirement is met. The surface currents tend to wash surface swirls downstream and thus prevent the organization of vortices.

Surface vortices sometimes tend to form above horizontal intakes such as FSIs, particularly when the surface water is stationary. Increasing the submergence may not be effective, but a horizontal grating at or just under the surface is very effective for clear water. In wastewater applications, a horizontal grating would plug with debris, but a row (sometimes a double row) of, for example, epoxy-coated steel (or stainless steel) pipes fastened to the wall just below LWL and sloping downward at 45 degrees is self-cleaning and just as effective as a horizontal grating. Both grating and pipes must extend far enough from the wall to intersect incipient vortices. See Figure 12-13. Openings in gratings or between pipes should not exceed 75 mm (3 in.). If the FSI is isolated by a sluice gate, the row of pipes above the opening can be fastened to the sluice gate itself, whereas those at either side are fastened to the wall.

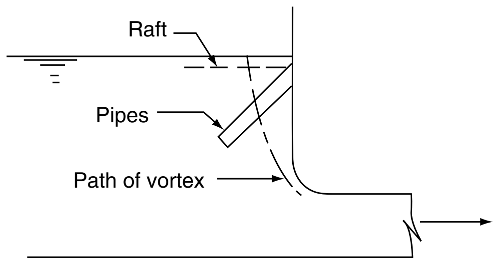


Figure 12-13. Vortex breakers. Raft for clean water; pipes for wastewater.

Underwater Vortices

A vortex occurs when a flow separates from an underwater surface to move toward an intake. Underwater vortices increase in intensity with an increase in flow rate and with a decrease in distance from the intake. Sometimes an underwater vortex forms within the liquid between two operating intakes. These are difficult to detect and to guard against but, fortunately, they rarely seem to be strong or well-defined.

Sidewall vortices. Sidewall vortices form beside intakes in trenches at about 0.28 D below the suction bell. The surest control device is a fillet. The apex must be above the nucleation point of the vortex—no less than $\frac{3}{8}$ D above the floor for suction bells $D/2$ above the floor. The side slopes can be between 40 and 50 degrees to the floor. The sloping surface prevents a vortex from organizing. See Figure 12-14.

Lowering the suction bell to a floor clearance of $D/4$ also eliminates sidewall vortices because the nucleation point is lower than the floor. To avoid the loss of intake energy (and a reduction of pump capacity), the velocity under the rim of the bell cannot be allowed to decelerate in approaching the mouth of the bell or in the bell itself. A cone as in Figure 12-15 creates a pathway of virtually equal velocities from under the rim of the bell where the velocity is $Q/[(\pi D)(D/4)]$ to the mouth of the bell where velocity is $Q/(\pi D^2/4)$. As the waterway passage area does not change appreciably, there is no loss of energy or pump efficiency.

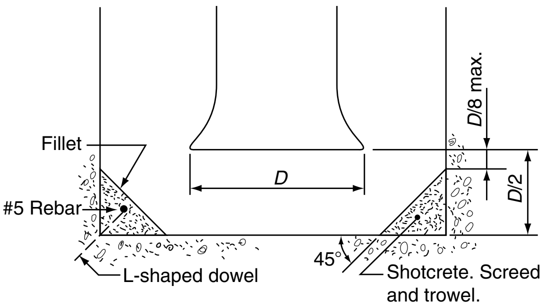


Figure 12-14. Fillets.

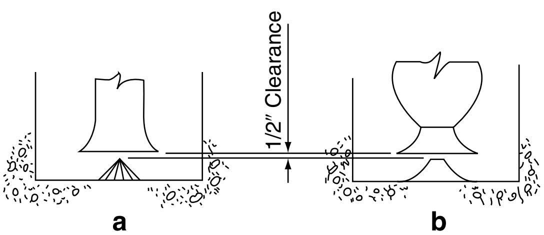


Figure 12-15. Floor cones. (a) Right cone; (b) flat cone.

Floor vortices. Floor vortices develop on flat floors under suction bells. They are strong at a bell floor clearance of $D/2$ and slightly stronger at a floor clearance of $D/4$. Either flow splitters or floor cones can reduce or eliminate floor vortices.

Cones. In clean water wet wells, cones (also "hydrocones") eliminate vortices, and they are useful for all the pump intakes. Set the tops of cones about 12 mm ($\frac{1}{2}$ in.) below the bell to avoid collision and damage when installing or removing pumps. The cone in Figure 12-15a is proper for bells shaped like cast-iron flares. Flatter, truncated cones (Figure 12-15b) are more suitable for flat bell mouths and for pumps with an end bearing outboard of the impeller.

Flow splitters. In a wastewater trench-type wet well intended for periodic pump-down for cleaning, a cone is useful under only the last pump intake. Under upstream intakes, cones would interrupt the thin sheet of high-velocity water during cleaning, so a flow splitter is the only device applicable to control those floor vortices. The flow splitter should be close to $\frac{3}{8}$ D high for a suction bell with a floor clearance of $D/2$. Flow splitters should clear suction bell rims enough to avoid trapping debris. If a clearance of 75 mm (3 in.) is required, the minimum D is 600 mm (24 in.). If the clearance is allowed to be 50 mm (2 in.), the minimum D is 400 mm (16 in.). Floor vortices cannot be avoided for smaller suction bells, so impellers should be either coated with cavitation-resistant material or made of cavitation-resistant metal (see Table 10-3). Terminate the flow splitter between the last two pumps.

During model tests of a wet well with column pumps at Alden Research Laboratory and at Montana State University, a narrow flow splitter (such as that in Figure 12-16) $\frac{3}{8}$ D high (for suction bells set $D/2$ above the floor) with a base width of $\frac{3}{8}$ D reduced floor vortices enough to meet the ANSI/HI 9.8-1998 standard. When high inflow velocities to the wet well occur (e.g., at the exit of an approach pipe or where a sluice gate is used as an aid to cleaning), the flow splitter should be narrow and the nose should taper from about mid-height of the ramp to zero at the top of the ramp to maintain a coherent flow. See the subsection "Improvements. Cleaning Operations," below. A base width of $\frac{3}{4}$ D, however, is more effective in reducing floor vortices, but it should only be used for V/S pumping where the inflow velocity to the wet well is scarcely above critical. In clean water wet wells, use either flow splitters or cones.

Dry Pit Pump Intakes

In general, a horizontal suction pipe extends from the wet pit to the dry pit with a valve (for isolating the pump) followed either by an eccentric reducer and a short-radius elbow or, better, by a reducing elbow to the pump. To obtain uniform (within 10%) exit velocities, keep the exit area equal to or less than half the entrance area of the elbow. A long-radius reducing elbow produces better flow distribution than does a short-radius one. The Turbo-Free™ elbow (Figure 12-6) is designed to produce uniform exit velocities and it allows a lower pump setting. Follow the advice given elsewhere in this chapter concerning layout and support of dry pit piping.

The entrance to the horizontal pipe may be a down-turned bend-and-flare as in Figures 12-16a and b, or it may end at the wet well wall with a horizontal flare as in Figures 12.16c and d. To keep the horizontal flare from interfering with the supercritical flow during cleaning, locate the lower edge of the flare at the top of the fillet. Fillets are necessary to keep the suction inlet from creating waves and interfering with the flow of water at supercritical velocities and, by increasing the hydraulic radius, they reduce friction loss. Note:

- There is no flow splitter in Figure 12-16c and d.
- A down-turned bend-and-flare must be used at the last pump to obtain cleaning velocities along the trench.
- The horizontal flare produces better flow distribution but offers less protection against the entrance of large debris that could damage the pump and the potential for vortex formation than does the down-turned elbow.
- Upstream down-turned bend-and-flares interfere asymmetrically with flow during normal pumping. Either limit the wet well inlet velocity to 0.9 m/s (3 ft/s) or install an inlet baffle as in Figure 12-12.

Any valve in the suction piping that interferes with the flow (e.g., a gate valve *almost* wide open) causes flow separation that may extend into the pump. The best valve is a ball or cone valve, but they are massive and expensive. The next best is a full-port eccentric

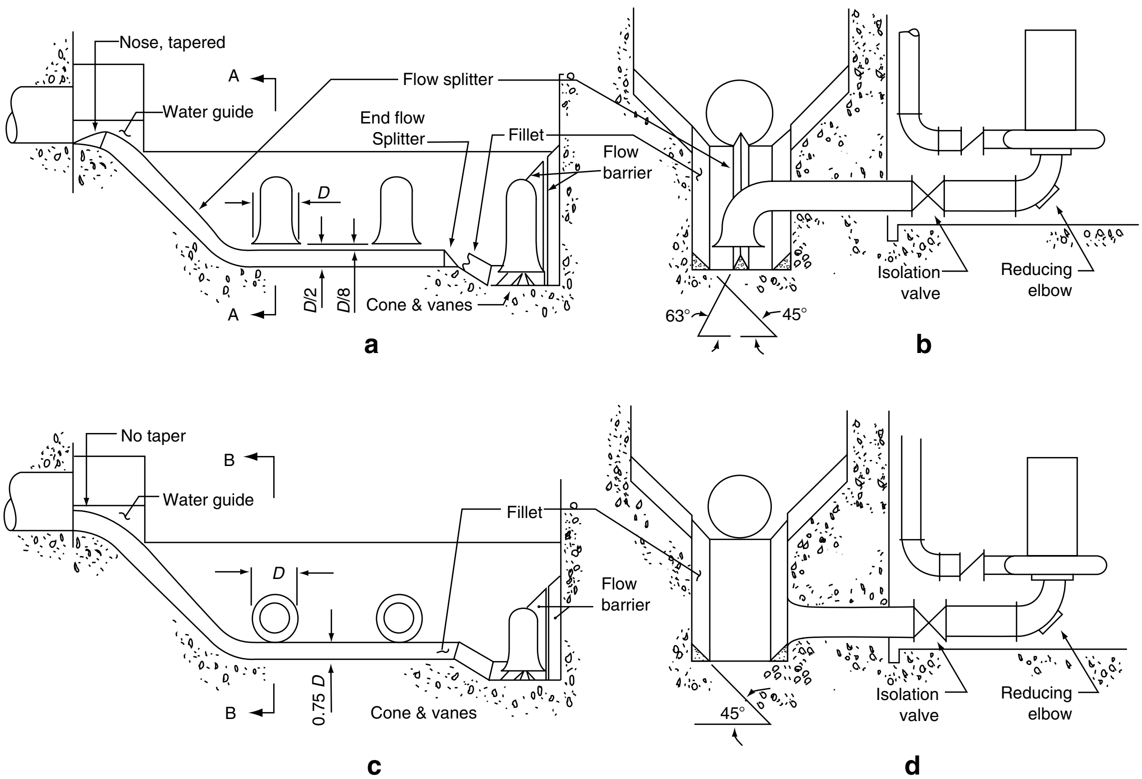


Figure 12-16. Flow splitter. (a) Longitudinal section, filled not shown; (b) Section A-A; (c) Longitudinal section with horizontal intakes and fillets but without flow splitter; (d) Section B-B.

plug valve. Some designers even use solid wedge resilient seat gate valves or knife gate valves for cost reduction, but such applications are risky. Investigate thoroughly before using gate valves.

Van Atta and Demlow [17] have discussed pump suction piping at length.

Swirling

Asymmetrical floor currents cause swirling, particularly in trench-type wet wells with dry pit pumps. Keeping the influent pipe velocity less than 0.75 m/s (2.5 ft/s) helps to reduce swirling, but such a low velocity requires either a large influent conduit or an influent baffle.

Suction bell vanes. The greatest swirling occurs in the middle pumps, and in a wet well with no improvements the swirling may be about double the allowable. Four vanes in the suction bell as shown in Figure 12-17 can reduce swirling to meet the standard with no assistance from other devices. Small vortices trail from the free edge of the vanes. The vortices could result in cavitation in wet pit-type pumps, but if the impeller is at least 1.0 d (where d is throat diameter of the suction pipe) from the vanes, the effect of the vortices is not significant. Cavitation from that source is not a consideration for dry pit pumps. As cast-iron impellers have comparatively low resistance to cavitation, consider coatings (such as epoxy, which can be renewed) or other materials (such as stainless steel).

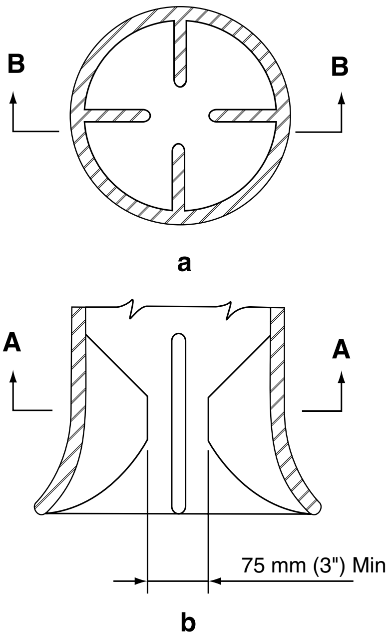


Figure 12-17. Suction bell vanes for wastewater service. (a) Section A-A; (b) Section B-B.

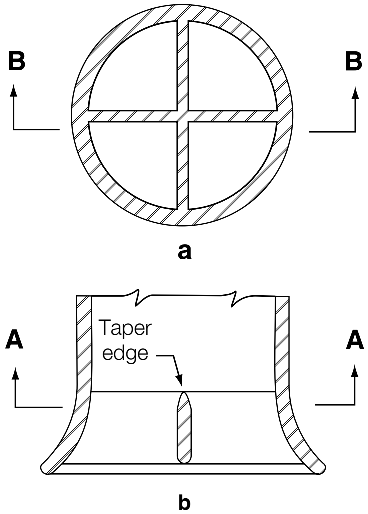


Figure 12-18. Suction bell vanes for clear water service. (a) Section A-A; (b) Section B-B.

In clear water service, the suction bell vanes shown in Figure 12-18 are even more effective, and they do not create significant vortices.

Floor cones with vanes. In clear water service, cones with four attached vanes under suction bells meet the swirling standard with only a small margin of safety. Six vanes, as shown in Figure 12-19, reduce swirling to about half the allowable limit. Six vanes are therefore strongly recommended. Of course, the cone also eliminates floor vortices.

Influent baffle. With no other improvements, the baffle as shown in Figure 12-12 cuts the swirling nearly in half, but swirling still might not always meet the allowable standard. If there are suction bell vanes, the addition of an influent baffle increases the margin of safety by about one-third and reduces the vortices trailing from the free edges of the vanes.

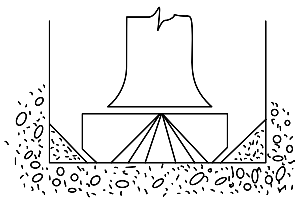


Figure 12-19. Clean water, anti-swirl cone to eliminate floor vortex. Of six vanes spaced at 60°, two are shown notched to clear fillets.

A baffle in conjunction with a cone having four vanes is roughly equivalent to a cone having six vanes.

Improvements. Cleaning Operations

Self-cleaning in any type of wet well is accomplished by lowering the water level (pump-down) as much as possible to concentrate scum and sludge so close to a pump intake that a vortex forms beside the pump and engulfs the solids. In self-cleaning, trench-type wet wells, a small flow of water (from 30 to 60 or 70% of the last pump's capacity, and 30% is better) gathers speed flowing down a ramp and forms a vigorous hydraulic jump that moves rapidly along the floor to the end of the narrow trench. The hydraulic jump mixes scum and sludge into a homogeneous mass readily ejected by the last pump. Obtain the proper flow rate by (1) waiting for the right time of day for cleaning, or by (2) operating the sluice gate to pass the proper rate. The first way is much simpler and easier, and it can be extended to flows either too small or too large. If the flow rate is insufficient, shut off the pumps to store sufficient water upstream, then turn on the last pump plus another (if needed) to dewater the wet well quickly and achieve the proper flow rate as the upstream storage diminishes. If the flow rate is more than wanted, operate the two end pumps at full speed and let turbulence do the cleaning. Using the sluice gate produces a high-velocity influent and requires a relatively long radius for the upper ramp curve and a long, tapered nose for the flow splitter.

Grease clings to the walls between LWL and HWL. It can be washed away at suitable intervals with a high-pressure hose. See Section 24-3, Subsection "Cleaning."

Entrance Conditions

If the sloping sides above the trench are extended all the way to the entrance wall, the influent water at pump-down climbs part way up the slope then drops into the sheet of water flowing down the ramp. The interference develops large cross-waves that create a "rooster tail" at the foot of the ramp. In a full-scale test, the rooster tail impacted the first pump, prevented formation of a downstream hydraulic jump, and largely destroyed the self-cleaning properties. When the influent was confined within a width of $2D$ by the water guide as shown in Figure 12-16, the rooster tail disappeared, a coherent hydraulic jump traveled downstream, and cleaning was completed in a few seconds. The walls of the water guide need to reach only high enough to contain the pump-down flow. Hence, there is no need to extend the height of the water guide above the centerline of the influent conduit.

Ramp. The required radius of curvature for the upper part of the ramp depends on the liquid velocity at the top of the ramp. Water at subcritical flow passes through critical velocity near the end of a pipe when discharging freely. Therefore, the fluid velocity at the top of the ramp is always slightly supercritical in wet wells designed for V/S pumps.

In wet wells designed for C/S pumps wherein an approach pipe is used, the velocity at the top of the ramp equals the terminal velocity down the approach pipe. That velocity is somewhat less than the velocity for the same size pipe given in Table B-10 or B-11 for maximum flow. For more accuracy, use the velocity obtained from Manning's equation (Equation 3-16) for the flow rate for pump-down, or, better still, use the Cahoon−Sanks program, ***Approach*** (obtainable free from the Internet [18, 19]), to find velocities during cleaning. Fit the nappe of water in free fall (a parabola) with a circular curve large enough to avoid a partial vacuum. The required radius is:

$$R = \text{FS}2.33\, v^2/2g \qquad (12\text{-}4)$$

where R is the radius, FS is a chosen factor of safety, v is the fluid velocity, and g is the acceleration due to gravity. It is wise to make R substantial regardless of Equation 12-4—not much less than 2 D for V/S pumping and 2.5 D for C/S pumping. The curve at the toe of the ramp should be large enough to prevent the formation of waves—not less than 4/3 D. Equation 12-4 is valid only if the tangent between the two curves slopes at 45 degrees.

Flow splitters and fillets. At pump-down, water at the toe of the ramp moves at velocities exceeding 4 m/s (13 ft/s). If the upstream ends of the flow splitter and fillets are terminated at the foot of the ramp, the high-velocity water strikes the noses of the flow splitter and fillets and bursts into a high, arching spray. The water loses energy and does not thereafter form a vigorous and coherent hydraulic jump, so considerable ability to scour solids rapidly is lost. Consequently, it is important to extend both the flow splitter and the fillets to the top of the ramp where the velocities are lower and where disturbances have the full length of the ramp for recovery.

The noses of the fillets can be blunt because water from a round pipe does not touch them.

Flow splitter noses. Near the top of the ramp, the full-sized flow splitter must taper to zero in a manner to minimize hydraulic disturbances. In V/S pumping (without use of a sluice gate) wherein the water issues from the inlet pipe at velocities not much above critical

velocity, the tapered section (nose) can be short—perhaps about half the length of the upper curve of the ramp. A somewhat longer taper is acceptable.

A long, easy nose taper is required for C/S pumping where inflow velocities down an approach pipe are high. Extend the nose from the top of the ramp to about the mid-depth of the ramp. Longer tapers (to the bottom of the ramp) also work well. It is important to clean at low flows (from 30% to no more than about 50% of the last pump BEC) to avoid the formation of cross-currents that occur with excessive depths over the flow splitter.

If a sluice gate is used to store water for cleaning, the velocity under the gate is high. Again, a short nose would send water flying and probably never permit an adequate hydraulic jump to form. The upper ramp radius for C/S pumping (or for sluice gate-assisted cleaning) should be no less than 2.5 D (regardless of the value given by Equation 12-4), and the nose should extend from the apex to the mid-depth of the ramp. Longer noses and slightly shorter noses also work quite well. For this condition, keep the base of the flow splitter equal to its height. Wider flow splitters are too disruptive while narrower flow splitters do not suppress floor vortices sufficiently. It is important to choose times to clean the wet well when flow rates are low enough to produce uniform water depths between flow splitter and fillets. The flow should be only about one-third Q_{LP} (the last pump's capacity) and certainly no more than one-half Q_{LP}. Larger flows ride too far over the top of the flow splitter, spill off its sides near the bottom of the ramp, and climb up the fillets and walls, leaving no water near the flow splitter for a short (halfway to Pump 2) distance downstream.

Trench

Friction losses are high at high velocity. If an analysis (***Trench2.0*** [19]) shows velocity at the end of the trench is too low, there are several means to conserve energy and keep the Froude number above 3.0.

- Make the trench smooth by troweling concrete surfaces very smooth to reduce Manning's n to 0.010. Make it even smoother by coating it with epoxy or lining it with PVC to reduce Manning's n to 0.009.
- Increase the hydraulic radius by making fillets larger.
- Improve the hydraulic radius by using a wide flow splitter (base equals twice the height).
- Increase the height of the ramp (expensive and not very effective).
- As a last resort, slope the floor sufficiently beginning at the point where the Froude number becomes less than 3.0.

Circulation behind Last Pump

During pump-down for cleaning, water tends to circulate strongly behind the last pump. Consequently, water on one side of the pump intake actually flows upstream with a considerable force that upsets the hydraulic jump and prevents adequate cleaning. Such circulation must be prevented.

Anti-rotation baffle. An anti-rotation baffle is compulsory, and it must fill virtually all the space between the pump intake and the back wall so that such circulation cannot occur at low water levels. For some intakes, it may be necessary to fasten part of the anti-rotation baffle to the pump intake itself, whereas part of it may be fastened to the back wall. There must be adequate construction clearance, of course, and for column pumps, enough clearance so that collisions do not occur as the column moves during pump operation.

The tendency to circulate around the last pump is sometimes severe enough to make water flow sideways under the bell. A fore and aft vane attached to the cone as shown in Figure 12-20 prevents such flow.

Elevation of Last Pump

In 2002, Cahoon, urged by Sanks, developed a program for computing fluid velocities and sequent depths in trenches; it is freely available on the Internet [19]. The trenches may be rectangular or they may have fillets, flow splitters, or both. Model studies (by Sanks and Williams) of minimum submersion of the last pump suction bell disclosed that suction was lost at submergences varying from 0.2 D to 0.3 D during pump-down in the MSU model, whereas the field study at Kirkland Pump Station showed loss of suction sometimes as low as 0.25 D and once (when low water levels persisted for several minutes) at 0.55 D. In factory pump tests, suction may not break until zero submergence is reached, but that occurs only where there is no turbulence, no swift currents, and no adjacent walls and floors. It is risky to depend on submergences less than $D/2$ while the last pump is cleaning the wet well.

To determine the required elevation of the last suction bell: (1) calculate the sequent depth near the end of the wet well from the Cahoon program, ***Trench 2.0*** [19]; (2) set the bottom of the suction bell no less

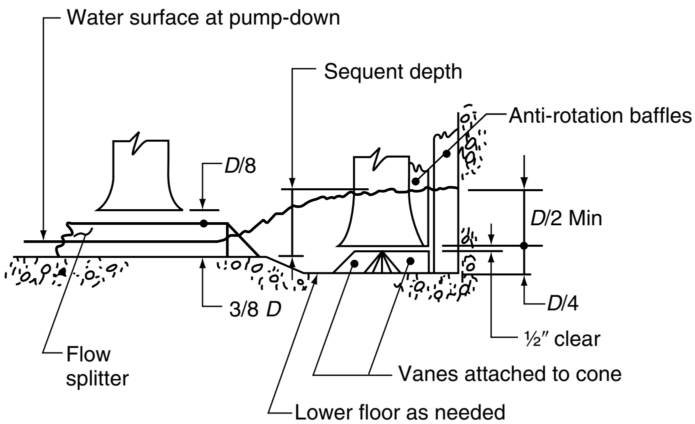


Figure 12-20. Construction at end of wastewater, trench-type wet well.

than *D*/2 below the sequent depth; and (3) set the floor *D*/4 below the suction bell. Slope the floor at about 30 degrees (as shown in Figure 12-20) between the last two pumps, if it is necessary to reach a new floor level.

In wastewater stations, a cone is better than a flow splitter under the last pump intake. Vanes attached to the cone are helpful. The end of the front (fore) vane must be smooth, rounded, sloped at 45 degrees (to prevent the accumulation of stringy material), and there should be 75 mm (3 in.) clearance between vane and the rim of the suction bell for the passage of debris. The rear (aft) vane can nearly touch the anti-rotation baffle. Under some conditions, front and rear vanes are necessary to aid the anti-rotation baffle by preventing rotation (sideways currents) from occurring underneath the bell.

Fabricating Improvement Devices and Facilities

The most appropriate methods for making the improvements discussed in the above subsections are given below. Note that stainless steel (ss) for welding fabrication must be type 347 or 316L. Note further that workers cannot install fillets or flow splitters in trenches less than about 0.9 m (3.0 ft) wide because of lack of working space, and that installation prices are a premium in trenches less than 1.2 m (4.0 ft) wide.

Ramps

Construct curved ramps, sloped at 45 degrees between curves, by casting the concrete in stair steps. Install dowels to project from the inside corners, attach reinforcement, and finish the ramp with shotcrete screeded (with the aid of templates bolted to the walls) then troweled to the proper surface.

Support of Wall and Sloping Floor (Shoulder) above Trench

The support must assure no cracking of shoulder and outer wall. Typically a foundation slab is cast the full width of the upper structure. In the U.S. where labor is expensive and materials are relatively cheap, the most economical means to support the shoulder above the foundation slab is solid concrete, thus making the trench wall very thick. For walls up to 1 m (3 ft) thick, use ordinary high-strength concrete per ACI 350 requirements. Thicker walls require mass concrete with low heat generation. That requires careful attention to the concrete mix design. See ACI 350. The mass of concrete may be needed for another purpose—to resist buoyancy.

Flow Splitters

Make flow splitters of type 316L or 347 stainless steel plate from 6 to 13 mm ($\frac{1}{4}$ to $\frac{1}{2}$ in.) thick stiffened as necessary. Floors and especially ramps are never perfectly formed, so considering the cost of the steel and its fabrication, it is wise to make thin, say 6-mm ($\frac{1}{4}$ -in.) plywood models of sections of the flow splitter, lay them on the ramp and, if necessary, on the trench floor, and trace offset lines on the plywood for use in cutting the steel plate. Except for warping due to welding (to be anticipated by the shop or the field welders), the steel should now fit the concrete perfectly. In the specifications for flow splitters to be welded to embedments and for flow splitters on ramps, require the

contractor to use this method to preclude delays, shoddy installation, and high costs.

Cut the plywood model to fit the curved portion of the ramp according to the equation

$$r = R/\sin\ \alpha \qquad (12\text{-}5)$$

where r is the radius of cut in the plywood, R is the radius of the ramp curvature, and α is the angle between the floor and the plywood. For the radius at the apex of the flow splitter, substitute $R \pm H$ (where H is the height of the flow splitter) for R in Equation 12-5. The plywood strips must be flexed somewhat to make them meet at the apex.

Near the top of the ramp and beginning, say, 22.5 degrees from the vertical, the flow splitter (for V/S pumps only) should taper to a point at the top of the ramp. Cut the inside radius of the nose or tapered part according to the equation

$$r = R/(\sin\ \alpha\ \cos\ \beta) \qquad (12\text{-}6)$$

where β is the angle between the centerline of the flow splitter and the intersection of splitter and ramp floor. The outside radius can vary from $(R + H)/\sin\ \alpha$ to $R/\sin\ \alpha$ or even less, depending on the desirable subtleties in the nose shape. The joint between nose and flow splitter requires a small triangular filler plate whose shape is best determined with the plywood template.

For wet wells with constant-speed pumps, the nose must be long and gently tapered to prevent the swiftly moving water from breaking into spray. Extend the nose from the ramp apex to about the midpoint of the ramp.

Install triangular pieces inside the apex to keep the plates at the proper angle, and add whatever stiffeners are necessary to keep the plates straight and true. After installation, it might be desirable to fill the flow splitter with grout to prevent microbial-induced corrosion and the entrance of putrescible material.

Costs vary greatly with time, place, and contractor, but in 2002, the cost of the flow splitter in Figure 12-21a

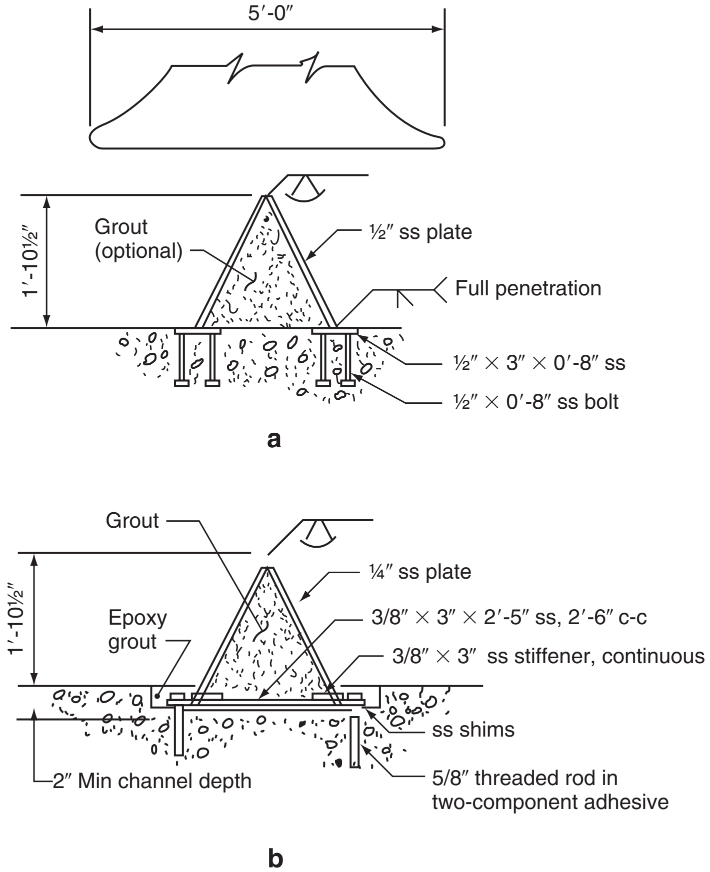


Figure 12-21. Flow splitters and anchorages. (a) Embedments for a prototype; (b) alternative design.

was $454 per linear foot including embedments, installation, and a grout fill. The cost of the flow splitter in Figure 12-21b was estimated by the same contractor to be $310 per linear foot or $350 if made of 10-mm ($^3/_8$ in.) plate. Grout fill adds $20 per linear foot, but it adds stiffness and prevents microbial-induced corrosion (MIC) that can produce pinholes that can deepen up to 1 mm/yr even in stainless steel (see Section 25-12). MIC would not occur on the outside because the water there is not stagnant unless sludge is allowed to build up unchecked. These costs might increase by 50% on the ramp because of more difficult working conditions. Size is almost inconsequential. Most of the cost is labor, and that changes little with size.

Fillets

Make fillets of shotcrete anchored into the corner with dowels, reinforced, and screeded to shape and troweled smooth.

Flow Splitters and Fillets for Laboratory Models

Painted wood held in place with double-sided adhesive tape is satisfactory for straight flow splitters and fillets, but trying to use short segments of wood along curved ramps is not. Cast a two-component, polyether-based urethane [20] in a straight wood mold coated with floor wax. Trim the exposed surface of the urethane with a sharp knife. The product is strong but rubbery, and bends easily over ramp curves, especially with the aid of warm air from an electric fan heater. Let the urethane cool to room temperature in a mold with a somewhat shorter radius than the one for the ramp to allow for some relaxation when freed. Hold it in the model with screws, rubber cement, or double-sided adhesive tape applied to the trimmed side. Form a nose by tapering the mold at one end.

Entrance Baffle

There is little force on the entrance baffle, and if strength were the only consideration, a flat steel plate held by a pair of horizontal beams would do. However, a flat plate has almost no torsional rigidity. To keep the baffle from fluttering, fabricate it as a relatively thin box section with, preferably, rounded edges. Support the baffle with fabricated box beams anchored into the sidewalls. As the atmosphere is corrosive (especially in wastewater wet wells), use stainless steel plate and tubes. See Figure 12-12.

Cones and Vanes

Cones can be cast of aluminum, cast iron, or stainless steel, but if cones have attached vanes, the molding forms are likely to be so expensive that fabricated stainless steel for the few elements required is apt to be the best alternative.

Welding cast iron invites the danger of cracking, so welds are not to be trusted for any parts requiring strength. Consequently, there is no satisfactory way to attach vanes to cast-iron suction bells. Suction bells with vanes (or, indeed, almost any complex form) can be fabricated of stainless steel by any good shop equipped with bending facilities and programmable plasma cutters.

Anchorage

Anchor cones and flow splitters with either (1) welds to embedments placed in fresh concrete (Figure 12-21a), or (2) bolts recessed and covered with epoxy grout. There must be no protrusions, not even bolts or nuts, to interfere with the smooth flow of high-velocity water. The embedments of Figure 12-21a make a strong, excellent anchorage, although the forces on flow splitters are usually low, and high strength is not normally needed.

Alternatives

Although devices such as fillets and flow splitters reduce vortices and cavitation, they might be less cost-effective for small pumping stations than some alternatives; also, they cannot be installed in trenches too narrow for access. None of the 27 Seattle Metro pumping stations built in the 1960s had the improvements discussed above, and those stations have performed well. The use of more resistant metals (see Table 10-3) such as stainless steel or aluminum bronze (see ANSI/HI 9.3.2-2000, Figure 9.20) for impellers and the addition of nickel to cast iron casings may be more cost effective, particularly for small pumps. Designers must decide which alternative is better for the particular application. Sometimes the client's O&M or engineering staff may have experience with materials for pump impellers, casings, and other pump components.

Submersible Pumps

If the pump volute is small enough, a submersible pump is best located in the trench with the motor

submerged below the LWL by setting the discharge elbow in a deep recess. Attach a short nozzle with straightening vanes to the pump and set the suction bell at *D*/2 above the floor as in Example 12-2. The nozzle for the last pump is extended to *D*/4 above the floor. After installing the discharge fitting, fill the recess with grout troweled into place with only enough room for the pump to be pulled out. If the volute is too large, the pump can be equipped with a long suction nozzle and mounted above the trench. Disadvantages of nozzles include cost, the unsuitability of some volutes for attaching the nozzle, the loss of static suction head, the extra head space required to pull the pump out of the wet well, and the need for some sort of cradle or dunnage to support the pump on the floor or on a truck bed. (Without a nozzle, a submersible pump is stable on a floor.) Nozzles make it possible to insert vanes to reduce swirling.

Other Pumps

Instead of submersible pumps, consider self-priming pumps if the suction lift is less than about 7.3 m (24 ft).Pumps from 63 L/s (1000 gal/min) to 1800 L/s (28,000 gal/min) at heads as great as 137 m (450 ft) are available.

For flow rates larger than about 125 L/s (2000 gal/min), consider the use of solids-handling column pumps instead of submersible pumps. Column pumps are well adapted to trench-type wet wells.

Choice of Drivers

V/S drives are preferred for wastewater pumps for several reasons: (1) V/S pumping prevents spikes in the flow rate—spikes that would otherwise disrupt sedimentation basins and other wastewater treatment processes such as chlorination; (2) the upstream sewer can be used as part of the wet well by programming water levels to maintain the normal velocities in the sewer; (3) the wet well does not need a large storage volume and can be made smaller and less costly; and (4) during periods of low flow, the wastewater stays fresher and easier to treat. The principal disadvantages are (1) the cost of V/S drives, (2) problems with rags at reduced speeds, and (3) a reduction in reliability (due to one more electrical component).

Combining C/S and V/S pumps

To reduce construction costs, some engineers substitute C/S pumps for some V/S units. Combined usage cannot be tolerated unless all pumps operate within their AORs at all station flow rates. Most pumps cannot operate at less than 35% of their maximum capacity. For the fewest number of pumps in a combination (three), two must be V/S pumps, and (as one is the standby) each must have about 50% more capacity than the C/S pump. For pumps to be of equal capacity, there must be at least four, of which three must be V/S units, because one V/S pump must be the standby. The basis for these recommendations is given in Section 15-5. There are no great savings in avoiding V/S drives for some of the pumps, but there is loss in operational flexibility to which some utility managers might object.

12-6. Summary of Trench-Type Wet Well Characteristics

Sumps for Solids-Bearing Waters

The characteristics of a good trench-type wet well for solids-bearing waters are:

1. Water enters the sump horizontally with no free fall from an influent conduit at least eight pipe diameters long in the vertical plane of the pumps. For either V/S or C/S pumping, LWL is far enough above the influent conduit invert to limit the entrance velocity to 1.2 m/s (4 ft/s) for column or submersible pumps and 1 m/s (3 ft/s) for dry pit pumps. For C/S pumping, the influent conduit is an approach pipe with its invert at the apex of the curved ramp.
2. During normal operation, the HWL for V/S pumping is at the soffit of inlet pipe (see Figure 12-22). For C/S pumping, the HWL can be anywhere above the inlet pipe.
3. For protecting the wet well, a motorized sluice gate allows isolating the sump from the inlet pipe. Using the sluice gate in cleaning operations is discouraged, however, because (1) it changes cleaning from a simple to a complex operation, and (2) it degrades the smooth nappe of water flow down the ramp.
4. The trench is $2D$ wide $\pm$ 10% (although only $2D$ is sanctioned by ANSI/HI 9.8) by about 2.5 D deep. But actual depth is also governed by: (1) required submergence of pump intakes (see Equation 12-1), (2) pump intake floor clearance (usually *D*/2), and (3) the required projected net cross-sectional area above the trench (see Item 6 below). The top of the trench should, however, be no less than 2 D above any intake unless model studies indicate less is allowable.

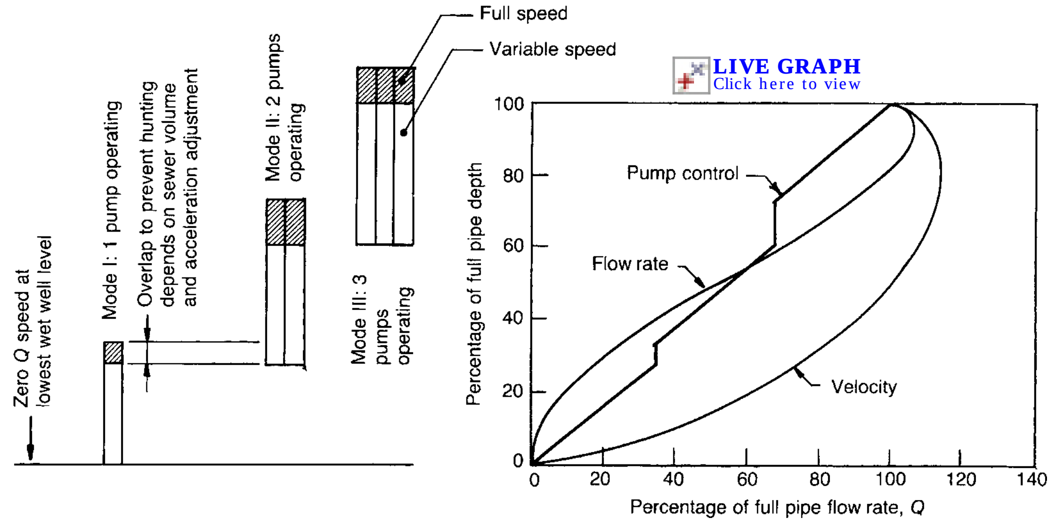


Figure 12-22. Design concepts for variable-speed pumping with the sewer used as part of the pump sump. After Brown and Caldwell.

5. The invert of the influent conduit is no less than 2 D above the highest bell mouth to keep incoming currents well above the pump intakes unless model studies indicate less is allowable.
6. The projected net cross-sectional area (gross area minus area of column or submersible pumps) of the water above the trench is sufficient at any depth (and any consequent flow rate in the inlet conduit) to maintain an average velocity (i.e., a nominal plug flow velocity) that never exceeds 0.3 m/s (1 ft/s).
7. The ogee spillway (used only during cleaning) extends from the invert of the inlet to the floor of the trench. The radius of the upper spillway curve is given by Equation 12-4, but it should be no less than 2 D. The radius of the reverse curve at the bottom of the ramp should be about 1.3 D. A short tangent between the two curves is desirable.
8. A water guide (the form is shown best in Figure 12-12) keeps water in a rectangular section from the inlet to the trench. Without the guide, water climbs up the sloping sides, falls back into the trench, and interferes with the flow of water down the ramp and along the trench during cleaning.
9. Above the trench, the sides are sloped at least 45 degrees if lined with, for example, PVC (or another plastic equally smooth) or 60 degrees if the surface is smooth concrete (steeper is better) to assist solids to slide into the trench. PVC makes it easier to hose grease and debris off the walls. For wastewater, the entire sump above LWL should be lined with a plastic such as Linabond® [21] or with high-grade epoxy to prevent corrosion.
10. Intakes may be spaced as close as 2.5 D center-to-center. Closer spacing may be used subject to model study. For intakes so close, the pump suction pipes may be splayed apart if necessary to achieve a clear space on three sides of pumps and valves no less than 1.1 m (42 in.).
11. Upstream intakes are no less than $D/2$ above the floor to avoid interference with the supercritical flow of water during cleaning. Interference patterns generated at the sides of the trench during cleaning may cause shock waves (or "rooster tails") that can in extreme situations reach a height of $D/2$. Their impingement on a suction bell would interfere with the flow of water underneath.
12. The allowable flow rate into the pump suction is based on the OD of the suction bell and the allowable velocities shown in Table 12-1. The authors, however, prefer a maximum intake velocity of 1.2 m/s (4 ft/s).
13. The last pump intake clears the end wall by, preferably, no more than $D/4$ to inhibit surface vortices. The intake must be no more than $D/4$ above the trench floor. The intake can be lowered by (a) lowering the floor to make a shallow depression near the intake, especially advantageous for small intakes that must pass large solids, or (b) by sloping the floor downward from the adjacent intake. See Figure 12-20.

14. An anti-rotation baffle as shown in Figure 12-20 must be installed at the last pump inlet. Otherwise, water currents tend to encircle the last pump, go upstream on one side, and stall the hydraulic jump far upstream. The cone is also necessary and the attached vane is very desirable.
15. Floor vanes under suction bells as shown in Figure 12-2 are unable to meet ANSI/HI requirements for suppression of floor vortices, so they are omitted in favor of a flow splitter as per Figures 12-3 and 12-16.
16. Swirling is often excessive. Vanes in the suction nozzle as in Figures 12-17 and 12-18 are, however, quite effective.
17. Install a backflow-protected water supply and hose valves with quick-connect fittings for a hose for washing grease off the walls with a 9- or 13-mm ($^3\!/_8$- or $^1\!/_2$-in.) nozzle. At a nozzle (pitot) pressure of 620 kPa (90 lb/in.^{2}), 2.5 L/s or 4.4 L/s (40 or 70 gal/min) are used at a velocity of 35 m/s (120 ft/s).
18. There is easy, convenient access for either a commercial pressure washer or the water supply system of Item 17 above for washing wet well walls. Best access is a sidewalk. Second-best is a catwalk. Either requires the wet well (a confined space) to meet NFPA 820 and OSHA confined space requirements.
19. Above LWL, the concrete is coated with a high grade of epoxy if H_2S is expected to be less than 200 ppm, or lined with PVC if the concentration of H_2S is expected to be greater. If extreme smoothness is required to maintain high velocity during pump-down, the fillets and floor may also be coated.

Controls for V/S Pumping

The controls are set to maintain the water level in the sump at the same elevation as the normal water level in the upstream sewer. Consider Figure 12-22 drawn for three duty pumps operating at V/S.

At 100% of full flow (Q_{max}), three pumps operate at V/S within Mode III. At 66% of Q_{max}, two pumps operate, and at 33% of Q_{max}, one pump operates. The modes (or ranges) are nearly equal in depth, so the pumping station control system can be programmed for a linear response. Hence, the normal depth of flow in the sewer is ensured at all times, which maintains carriage (scouring) velocities. Note that pump capacity does not vary linearly with speed. However, the difference in the relationship between sewer capacity and pump capacity is usually small, and refinements are not considered justified in terms of benefit versus additional cost for more sophisticated control. If instability (or surging) is observed, however, the likely cause is rapid acceleration of the pumping equipment, so surging can be eliminated by decreasing the rate of acceleration. Backup in the sewer is not a problem with properly set elevation controls. These adjustments should be part of the start-up procedure.

Contrary to the above, some engineers are concerned that the pump speed controllers may cause backup or surging in the sewer pipe and prefer to design "more conservatively" by setting the maximum height of Range 3 at the mid-depth of the sewer or even at the invert. The problem with this approach is that, at flows greater than half the capacity of the sewer, the incoming flow enters the wet well at a velocity greater than that allowed by ANSI/HI 9.8-1998. Furthermore, if the control level is set at the invert, the incoming stream reaches critical depth and drives air (entrained at the wet well water surface) deep into a pool where it can be ingested by the pumps, thereby reducing their head, capacity, and efficiency and causing a potential for damage. The free fall also induces turbulence and odor release. The authors object to this approach. If the HWL in the sump is lowered by, say, 0.3 m (1 ft) more than necessary in a 0.22-m^3/s (5-Mgal/d) pumping station where the average cost of electricity is 6¢/kW·h, the 20-year present-worth cost of such a decision is nearly $5000.

12-7. Trench-Type Wet Well Design

This section contains two designs, one for variable-speed (V/S) pumps and one for constant-speed (C/S) pumps, with all the relevant calculations.

Wet Wells for V/S Pumps

The design of a trench-type wet well with wet pit pumps operating at V/S is illustrated in Example 12-1. The example abounds in advice for improving wet wells. See Section 26-2 for a somewhat different wet well and for the design of the entire station. For clarity, Example 12-1 is worked in only U.S. customary units.

Example 12-1
Design of a Trench-Type Wet Well for V/S Wastewater Pumping

A. Design conditions.

Station $Q_{max} = 5.0$ Mgal/d (3470 gal/min or $7.8\,ft^3/s$)
Pump $Q_{max} = 3.0$ Mgal/d (2100 gal/min or $4.7\,ft^3/s$) at runout (also called Q_{runout})
Influent pipe = 24-in. diameter RCP; slope = 0.002; $n = 0.013$.

B. Select pump and size of suction bell.

Use nonclog column pumps to avoid a dry pit. See Table 25-4. Fairbanks Morse VTSH® is one example. There are at least two other manufacturers.

Suction bell $D = 19$ in. = 1.58 ft. Intake velocity = $Q/A = 4.65/1.97 = 2.36$ ft/s. This is a little low, but the bell is very flat so velocity increases quickly. In a turbulent hydraulic jump, it should ingest grit readily.

If you have a choice of suction bell diameter, size it for an entrance velocity of 2.5 to 3.5 or even 4.0 ft/s at maximum flow rate.

C. Determine the dimensions of the cross section of the wet well.

Trench width = 2 D = 38 in. = 3.17 ft.
Trench depth = 2.5 D = 47.5 in. = 3.96 ft.
Side slopes above trench = 45 degrees, lined with plastic.
Elevation top of trench with respect to invert of inlet. Note: (a) at any flow rate, velocity of inflow above trench must be less than 1.0 ft/s; (b) water surface elevation in wet well is same as that in inlet pipe.
Check three water level stages: (1) for Q_{min} = 35% of Q_{runout}, (2) for Q_{runout}, and (3) for Station Q_{max}.

Use Escritt = 1 (for corrected Manning's equation.) in ***UnifCrit2.2***

Stage 1: $Q_{min} = 0.35 \times 4.65\,ft^3/s = 1.63\,ft^3/s$. From ***UnifCrit2.2***, $y_n = 0.61$ ft, $v = 2.01$ ft/s.
Stage 2: $Q_{runout} = 4.65\,ft^3/s$. From ***UnifCrit2.2***, $y_n = 1.06$ ft, $v = 2.75$ ft/s.
Stage 3: Station $Q_{max} = 7.75\,ft^3/s$. From ***UnifCrit2.2***, $y_n = 1.45$ ft, $v = 3.18$ ft/s.

Note that Q_{max} will probably never occur.

Locate top of trench relative to influent pipe invert. Heed advice above.

Stage 1: $Area_{req'd}$ above trench = $1.63\,ft^2 = 3.2\,y + y^2 \cdot y_{req'd} = 0.45$ ft.
$y_n = 0.61$ ft OK.

Stage 2: $Area_{req'd}$ above trench = $4.65\,ft^2 = 3.2y + y^2 \cdot y_{req'd} = 1.09$ ft.
$y_n = 1.06$ ft OK.
Stage 3: $Area_{req'd}$ above trench = $7.75\,ft^2 = 3.2y + y^2 \cdot y_{req'd} = 1.61$ ft. Inadequate, so far.
$y_n = 1.45$ ft

Increase depth in sewer to 1.61 ft. From ***UnifCrit2.2***, $v = 2.86$ ft OK.

Check submergence of pump suction bells.

Froude number: $\mathbf{F} = v\,(gD)^{-0.5} = 2.36\,(32.2 \times {}^{19}/_{12})^{-0.05} = 0.33$

Equation 12–1: $S = (1 + 2.3\mathbf{F})D = (1 + 2.3 \times 0.33)\ {}^{19}/_{12} = 2.79$ ft

Actual submergence at runout (from Fig. 43) = $3.96 + 1.06 - {}^{9.5}/_{12} = 4.23$. OK.

D. Design ramp for cleaning.

The velocity is always slightly supercritical at the end of a freely discharging pipe. From ***UnifCrit2.2***, $v = 5.0$ ft/s.

From Equation 12-4, the radius of curvature at the top of ramp is
$R = \text{FS}2.33v^2/2g$
Where FS is a factor of safety. For FS = 2,
$R = 2 \times 2.33(5.0)^2/64.4) = 1.8$ ft. Nevertheless, use $R \approx 2D$. Use 3.0 ft.
At bottom of ramp, use $R = 2.0$ ft.
Between curves, ramp slopes 45 degrees.

Length of ramp

Horizontal projection of upper curve = 3.0 sin 45 degrees = 2.12 ft.
Vertical projection of upper curve = 3.0 − 3.0 cos 45 degrees = 0.88 ft.
Horizontal projection of lower curve = 2.0 sin 45 degrees = 1.41 ft.
Vertical projection of lower curve = 2.0 − 2.0 cos 45 degrees = 0.59 ft.
Vertical (= horizontal) projection of tangent between curves = 3.96 − 0.88 − 0.59 = 2.49 ft.
Total horizontal projection of ramp = 2.12 + 1.41 + 2.49 = 6.02 ft.

E. Length of wet well flat floor.

Length = toe of ramp to first pump (0.5 *D* min) + pump spacing × (n−1) pumps + last pump to end wall (0.75 *D*)
Assume pump spacing = 3.5 *D* to provide room for maintenance.
For three pumps (two duty + one standby), flat floor length = $0.5\,D + 2 \times 3.5\,D + 0.75\,D = 8.25\,D = 13.06$ ft

F. Total length of wet well.

Ramp + floor = 6.02 + 13.06 = 19.08 ft

G. Fillets and flow splitter.

Whether to use fillets and flow splitter in a trench so narrow is a matter of judgment and economics. If the velocity at the end of the wet well is insufficient for adequate cleaning, however, a flow splitter and especially fillets may be necessary. Run the ***Trench2.0*** program (1) both with and without flow splitters and fillets, (2) with different rates of inflow, and (3) with smooth coatings or linings if needed to produce low Manning's *n* values before making a decision. Note that fillets extend over the full length of the trench and flow splitters only extend from the apex of the ramp to halfway between the last two pumps. Also note the Froude number at the last pump should be between 3 and 6 for best results. Cleaning works best between flow rates of ⅓ to ⅔ of the capacity of the last pump.

On the basis of Figures 12-24 and 12-25 (for one set of many possible sets of parameters), consider the use of a flow splitter and fillets. The Froude number of either is satisfactory, but the higher Froude number with fillets and flow splitter (Figure 12-25) means that less flow could be used for cleaning. That gives operators more flexibility, and the suppression of vortices extends the life of impellers.

The height of fillets must never be less than ¾ of the floor clearance (*D*/2) of upstream pumps. The net height is, therefore, 0.375 *D*—in this example, 7.12 in.

The height of the flow splitter usually equals that of the fillets, but here the clearance between the lip of the suction bell and the flow splitter is 2.38 in. Many engineers would prefer a minimum of 3 in. even though the bottleneck occurs at only two points. Hence, make the flow splitter 6.5 in. high. See Figure 12-23. Flow splitters with heights as low as 0.25 *D* are almost as effective in preventing floor vortices as those 0.375 *D* high.

H. Elevation of last pump intake.

For the sequent depth at the last pump intake of 0.68 ft (Figure 12-25), the elevation of the last suction bell should be $0.68 - D/2$ (0.79) and the floor should be *D*/4 (0.40) below the bell. Therefore, drop the floor 0.68 − 0.79 − 0.40 = −0.51 ft below the upstream floor.

Next Page

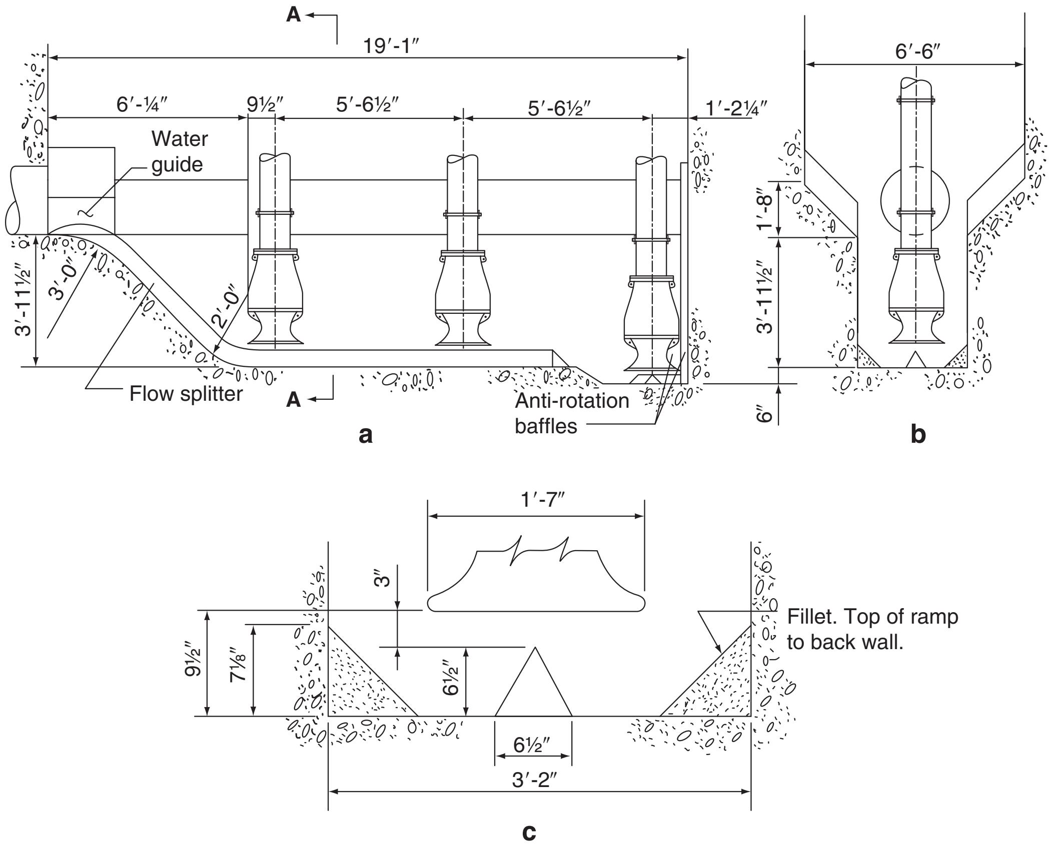


Figure 12-23. Wet well. Example 12-1. (a) Longitudinal section; (b) Section A-A; (c) detail of Section A-A.

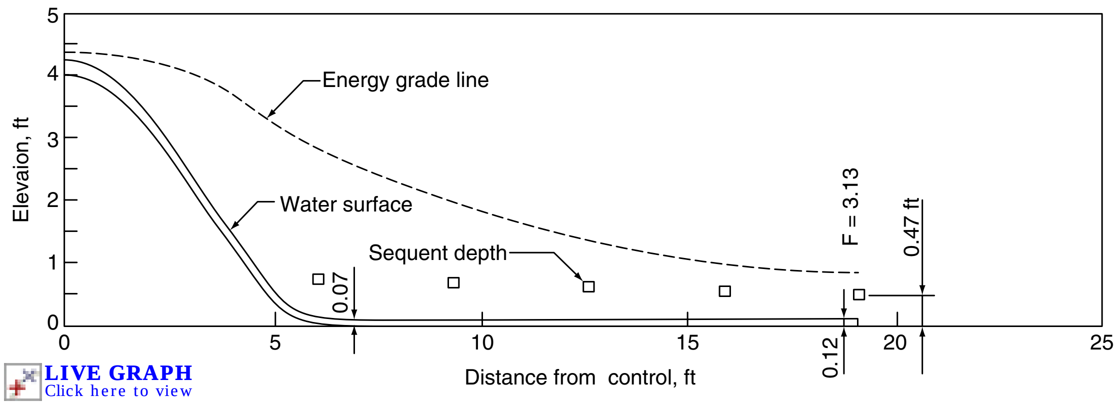


Figure 12-24. Hydraulic and energy grade lines for trench without fillets and flow splitters. Q = 2.32 ft^3/s, $n = 0.011$.

I. Time period for cleaning.

The time required to empty and scour the trench depends primarily on the rate of inflow. Inflow should be small so the pumps do not operate long at the severe conditions of intake submergence, but it must be large enough to be effective. At low submergence, the capacity

Chapter 13

Electric Motors

PAUL C. LEACH
ALAN VAUSE

CONTRIBUTORS
Robert S. Benfell
Harrison C. Bicknell
Donald R. Bjork
Victor G. Gerez
Mayo Gottliebson
Stanley S. Hong
Philip A. Huff
Richard L. Nailen
Stephen H. Palac
Russell L. Parr
M. LeRoy Patterson
Derek Petersen
Richard N. Skeehan
Patricia A. Trager
Frank A. Woodbury

The focus of this chapter is the practical selection (from the myriad of special features and characteristics available) of the proper driver in a manner that will result in an economical, dependable drive unit. The discussion of motors includes an elementary explanation of the several types commonly used as pump drivers. Refer to standard texts on electrical fundamentals and alternating current machines for an in-depth theory of operation.

Before designing the electrical system for driving the pumps, it is important to be aware of choices, restrictions, and requirements, such as: (1) the various motor characteristics available both as standard and as specialty or custom products; (2) the costs of the various types and configurations of suitable motors; and (3) the restrictions resulting from station design criteria or earlier decisions. For example, in rebuilding a station, there are usually severe restrictions on space—sometimes for the motor and pump combination and usually for motor controls.

References to a code or standard are given in abbreviated form (such as NEC for National Electrical Code).

13-1. General

Some of the definitions (given here for convenience) are shortened, simplified versions of the definitions in ANSI/IEEE Standard 100. For more definitions, see Chapter 2 (which also contains abbreviations), Chapter 8, and the latest edition of the NEC.

Definitions

Adjustable-frequency drive (AFD): A device that can be regulated to convert the frequency (hertz) of incoming electrical current to lower or higher frequencies to change electric motor speeds.

Adjustable-speed drive: A drive designed so that speed can be varied throughout a considerable range (1) independently of the load, and (2) as regulated by some external signal (see variable speed).

Armature: The member of an electric machine in which an alternating voltage is generated by virtue of relative motion with respect to a magnetic field. It is the stator in a squirrel-cage motor and the rotor in a dc motor.

Brush: A conductor (usually made of carbon) that maintains an electrical connection between the stationary and moving parts of a machine.

Collector ring (slip ring): A metal ring mounted on an electric machine that conducts current into or out of the rotating member through stationary brushes.

Commutator: A cylindrical ring or disc assembly of conductors, individually insulated in a supporting structure with an exposed surface for contact with current-collecting brushes. In effect, it is a switching mechanism between the dc of the brushes and the ac in the armature of a dc machine.

Drip-proof enclosure: An open machine with ventilating openings constructed so that drops of liquid or solid particles falling on the machine at any angle not greater than 15 degrees from the vertical cannot enter the machine.

Explosion-proof enclosure: An enclosure designed to withstand an explosion of a specified gas that may occur within it and to prevent the ignition of the same gas surrounding the enclosure. It must operate at a temperature that prevents the surrounding flammable atmosphere from being ignited.

Framework (frame): The stationary supporting structure that contains the stator windings and the rotor.

Frame mount: A supporting structure that connects the motor to the pump with an intervening shaft coupling.

Induction motor: An alternating current motor in which a primary winding (usually on the stator) is connected to the power source and a polyphase secondary winding or squirrel-cage secondary winding (usually on the rotor) carries induced current.

Inrush: The surge of electricity into a motor at rest when it is energized. The surge, which diminishes to the running load current at full speed in approximately 5 to 30 s, is typically about 6 times the full-load current, but it may be as much as 10 in some motors or it may be reduced in "soft starts" to about 2.5 times the full-load current.

Locked rotor current: Steady-state current taken from the line with the rotor at standstill (at rated voltage and frequency)—the current used when starting the motor and load.

Locked rotor torque: The minimum torque that a motor develops at rest for all angular positions of the rotor (with rated voltage applied at rated frequency).

Power factor: True power divided by apparent power. Lag of current behind voltage in inductive ac circuits reduces true power from VA to $V \times A \times \cos\theta$, where V is volts, A is amperes, and θ is the lag angle (see Equation 8-4).

Rotor (Squirrel Cage): The rotating member of an induction motor made up of stacked laminations. A shaft running through the center and a squirrel cage which holds the laminations together and act as a conductor for the induced magnetic field. The squirrel cage is made by casting molten aluminum into the slots cut into each lamination.

Round (cylindrical) rotor: A cylindrical rotor in which the coil sides of the windings are contained in axial slots.

Salient pole: A field pole that projects from the yoke or hub toward the primary winding core. Salient pole and round (cylindrical) rotors ordinarily are used to describe the rotor design of synchronous motors [i.e., a salient-pole synchronous motor or a round-(cylindrical)-rotor synchronous motor].

Service factor (SF): A multiplier (applied to the rated power) that indicates the built-in capability of carrying a continuous load greater than the nameplate rating (but only at the rated voltage and frequency). Standard service factors are 1.0 and 1.15.

Slip: The difference between the rotational speed of the magnetic field in an induction machine and the rotor. An induction motor develops its ability to drive a load by virtue of its slip, whereas a synchronous motor operates at zero slip.

Soft start: A type of motor starting in which the inrush is greatly reduced. Soft start is generally applied to solid-state starting and acceleration control systems.

Splash-proof enclosure: An open motor with the ventilating openings constructed so that drops or particles falling or coming toward it at any angle not greater than 100 degrees downward from the vertical cannot enter. The splash-proof enclosure is no longer generally available. Weather-protected Type I is an alternative enclosure type.

Stator: The portion of a machine that includes and supports the stationary portions of the magnetic circuit and the associated windings and leads. It may include a frame or shell, winding supports, ventilation circuits, coolers, and temperature detectors. A base is not usually considered to be part of the stator.

Submersible enclosure: An enclosure constructed so as to operate successfully when submerged in water under specified conditions of pressure and time.

Synchronous motor: A motor that has field poles excited by direct current so that the speed of the rotor equals that of the rotating magnetic flux.

Synchronous speed: The rotational speed of the magnetic field in an electric motor and, in synchronous motors, the rotational speed of the shaft and rotor.

Torque, accelerating: The difference between the torque produced within the motor and that required by the driven load at any particular shaft speed.

Torque, full-load: The torque required to produce the rated motor horsepower at the rated speed.

Torque, locked-rotor: The torque produced with the motor at a standstill but energized at the rated voltage. Locked-rotor torque is also called "starting torque." At the instant power is applied to an induction motor at rest, the motor is analogous to a transformer with its secondary short-circuited through a very low impedance.

Totally enclosed (TE) motor: An integral enclosure constructed so that, although it is not necessarily airtight, the enclosed air has no deliberate connection with the external air except as required for draining and breathing.

Totally enclosed, fan-cooled (TEFC) motor: A totally enclosed motor equipped for exterior cooling by means of fans integral with the motor but external to the enclosing parts.

Totally enclosed, nonventilated (TENV) motor: A totally enclosed motor that is not equipped for cooling by means external to the enclosing parts.

Totally enclosed, pipe-ventilated motor: A totally enclosed motor except for openings arranged so that inlet and outlet ducts or pipes may be connected to them for the admission and discharge of the ventilating air. This air may be circulated by means that are: (1) integral with the machine, or (2) external to and not part of the machine. The latter machines are called separately ventilated or forced-ventilated machines.

Variable-speed drive: A drive designed so that the speed varies through a considerable range as a function of load (see adjustable-speed drive).

Weather-protected (WP) machine: A guarded machine with the ventilating passages designed to minimize the entrance of rain, snow, and airborne particles to the electrical parts. There are two types, I and II. Type II is designed for more severe applications.

Wound-rotor motor: An induction motor with the rotor windings connected to collector rings. The brush terminals may be either short-circuited or closed through adjustable circuits.

Standards and Codes

The most authoritative information on standards relating to electric motors is NEMA MG-1. The Institute of Electrical and Electronics Engineers provides many sources of reference material in the *IEEE Standards* and in various technical articles in *IEEE Transactions on Industry Applications*. These references are recommended for in-depth information on motors, new developments in research, and motor manufacturing. Many manufacturers' catalogs, Internet websites, and special brochures provide useful information on motor characteristics and pricing.

If motor characteristics are selected from a particular standard, there is no assurance that any manufacturer will build the motor as specified because each manufacturer has its own "standard" models and modifications (called "adders"). During early stages of design, therefore, check with several manufacturers whose representatives can supply current information.

13-2. Applications of Motors

Electric motors are the most frequently used drivers in pumping stations, primarily because of their versatility, compactness, and low maintenance. The most common machine is the polyphase (three-phase) squirrel-cage induction motor; these motors range in size from less than one to several thousand horsepower. Induction motors up to about 600 kW (800 hp) are usually used for adjustable-speed drives, but larger drives tend to be more economical with a wound-rotor or a synchronous motor. Large squirrel-cage motors, however, have high efficiencies and power factors that make the operating costs approach those of synchronous motors without the high capital cost. Furthermore, the controls are more complex for synchronous and wound rotor motors. Single-phase motors are used to drive small loads and are not considered as drivers in pumping

stations. Therefore, the focus of this chapter is on the application of three-phase alternating current motors.

Almost all pumping stations also require electric motors to drive auxiliary equipment. These auxiliaries can range from fractional kilowatt or horsepower ratings up to 30 or 40 kW (40 or 50 hp) in the larger stations. The advantages of electric motors are summarized in Table 13-1.

Motor Nameplate

To describe the basic construction, performance, and mounting parameters of a motor, the National Electrical Manufacturers Association (NEMA) defines some basic design and dimensional parameters in NEMA Standard MG-1. These parameters are then coded onto the motor nameplate. Section MG 1-10.40, "Nameplate Marking for Medium Single-Phase and Polyphase Induction Motors," of the NEMA standard requires that "The following minimum amount of information shall be given on all nameplates of single-phase and polyphase induction motors."

1. Manufacturer's *type* and *frame* designation. "Type" is not defined by NEMA and is often used by motor manufacturers to define motor as single- or polyphase, single- or multi-speed. Motors of a given horsepower rating are built in a certain size of frame or housing. For standardization, a frame size has been assigned for each integral horsepower motor so that shaft heights and dimensions will be the same to allow motors to be interchanged. Refer to NEMA MG-1 for frame designations.
2. Horsepower. The rated shaft output of the motor.
3. Time rating. The time rating or "Duty" defines the length of time during which the motor can carry its nameplate rating without exceeding design limits. Pump motors are rated for continuous duty or "Cont."
4. Maximum ambient temperature for which the motor is designed (i.e., usually 40 or 50°C).
5. Insulation system designation. Class A, B, F, or H.
6. RPM at rated full load.
7. Frequency. 60 hertz in North America, 50 hertz in Europe.
8. Number of phases. Usually three-phase for motors ½ hp and larger, one-phase for less than ½ hp.
9. Rated load current.
10. Voltage.
11. Code. A letter code indicating the starting current characteristic of the motor and defined as the locked rotor kVA on a per hp basis. Codes are defined in MG-1 by a series of letters from A to V. As a rule, the further down the alphabet from A, the higher is the value of inrush kVA per hp.

Table 13-1. Electric Motors as Pump Drivers

Advantages	Disadvantages
Compactness and minimum space requirements	Subject to damage by flooding (most motors)
Usually less expensive for installed cost than engines (engines may be more economical if electricity is costly and fuel is cheap or if inertia is needed for water hammer control)	High starting currents for most types
Reliable	Subject to utility outages
Usually easily removed for major maintenance or replacement	Standby generation may be required for the full capacity of the system
Available in wide range of sizes	Control equipment and switchgear sometimes require considerable space
Insulation systems available to suit a variety of ambient temperature and environmental conditions	Protection systems may be complex and sometimes expensive
A number of mounting configurations available	Most applications use purchased electrical power where engine fuels may be less costly
A number of enclosures available to suit physical conditions	Subject to possible overheating due to: (1) nonsinusoidal current as in adjustable-frequency drives (AFDs) using rectifier/inverter power sources (the harmonic component of current creates additional I^2R and core losses in the motor); (2) unbalanced phase voltages [1,2] (see also NEMA MG-1); (3) over-voltage that causes over-excitation of magnetic materials and a consequent increase of core losses; (4) short cycling (too-frequent starting of motor by faulty ON-OFF controls or by careless design of pumping station or its control system); (5) excessive pump load conditions

Codes F or G are the most common codes for centrifugal pump motor applications.

12. Design. Letter code designating the NEMA MG-1 category for the torque-current characteristic of the motor. Designs B and A are the most common codes for centrifugal pump motor applications.
13. Efficiency. Nominal efficiency (where required) as defined in NEMA MG-1 Tables 12-11 or 12-12.
14. Service Factor (SF).
15. Enclosure code. TEFC, TENV, ODP, WPII, etc. as defined by NEMA MG-1.
16. Over Temp Prot. For motors rated above 1 hp equipped with over-temperature devices or systems.

Ambient Conditions

Electric motors are usually comparable in size to the pump itself—particularly in the small- to medium-sized stations. Therefore, the use of electric motors as drivers is not a major problem in pumping station space allotment. Enclosures for electric motors can be selected to suit a wide range of environments: (1) totally submerged, (2) humid, (3) explosive atmosphere, (4) clean and dry areas indoors, or (5) out-of-doors directly in intense heat, rain, and freezing temperatures. Although different environments require different enclosures, the basic operating characteristics are the same. An often-overlooked ambient condition is elevation above sea level. Air-cooled motors must be derated for service at elevations above 1000 m (3000 ft) because heat transfer surfaces are less efficient in thinner air.

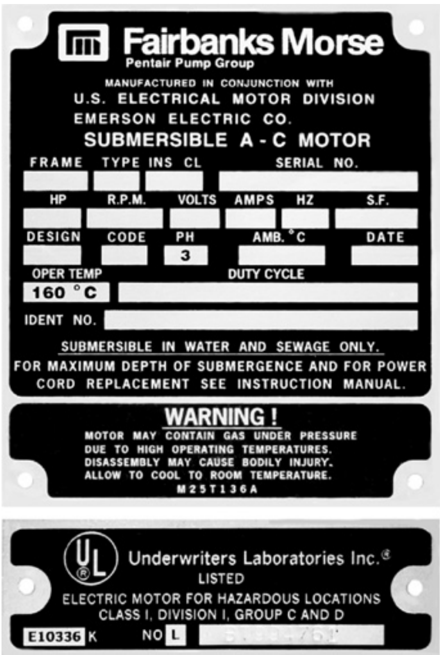


Figure 13-1. A motor nameplate. Photo by George Tchobanoglous.

Mountings

Electric motors may be mounted in many ways. The motor can be close-coupled to the pump in either a horizontal or a vertical arrangement in which the pump impeller is fastened directly to the shaft of the motor. Wherever possible, however, avoid configurations that lead to high maintenance or difficult replacement problems. Close-coupled pumps are more difficult to maintain because most units must be disassembled to service the seals. Many units utilizing close-coupled designs have shafts and bearings that are inadequate for severe service in wastewater pumping. If the motor is separated from the pump by a spacer or by a frame mount (which allows a coupling to be installed), seals can be changed without moving either the motor or the pump. Such arrangements are used in both horizontal and vertical configurations. Furthermore, for ease in maintenance the electrical system should not have to be disturbed to remove a pump, nor should the pump or piping have to be disturbed to remove a motor.

It is common to support the shaft and impellers for a deep well pump from a thrust bearing located atop a vertically mounted motor. Some of these motors have two thrust bearings. It is also common to mount a deep well turbine pump above a small-diameter motor and submerge the entire unit in the well.

In some wastewater pumping stations, a vertical pump is located in a dry well. It is good practice to place the motor and other electrical equipment (such as starters and panelboards) on an operating floor that is:

- above grade and safe if the pumping station is flooded;
- accessible only from the outside with no direct access to the wet well and completely sealed from the wet well to prevent contamination with sewer gases; and
- above the 100-yr flood level.

Any motors and electrical devices in locations classified as Class 1, Division 1, by NEC Subarticle 500, 5(B) must be explosion-proof. Motors and electrical devices in locations classified as Class 1, Division 2 must be rated for this classification and must be nonsparking, so there can be no open switches or open contacts; hence, wound-rotor motors with standard open enclosures are prohibited, although polyphase open or totally enclosed motors with no switches are permissible. Neither type of hazardous location should be allowed to occur in the design of new facilities. If an existing facility containing a hazardous area is to be remodeled, it is necessary to consult the proper inspection authority in the formative stage of design and obtain written agreement that the designer's interpretation of the classification is correct. Explosion-proof motors and other electrical equipment are very expensive, but some savings are possible in the overall construction costs of equipment in Class 1, Division 2 locations compared to Class 1, Division 1 locations. There may be some savings in lighting costs, little or none in wiring methods, and a significant amount if there are a number of motors.

If the station is safe from flooding, an alternative arrangement for a vertical pump in a dry well is an open drip-proof motor frame-mounted above the pump. This arrangement avoids a long shaft and intermediate supports. If flooding is a possibility, submersible motors can be installed, but the problem of ambient air cooling for normal operating conditions usually requires substantial derating of the motors. Note that the ordinary pump cannot be operated while submerged without damage to seals and bearings. The addition of bearing isolators, however, especially the pressure-compensated type, mitigates this concern. A better solution for the flood-prone dry wells is apt to be the use of immersible, rather than submersible, motors.

Still another alternative is the close-coupled submersible pump and motor mounted on slide rails in the wet well. These units are increasing in popularity because of the overall low initial cost due to savings in the superstructure. The motors, however, are subject to loss of seals, penetration of moisture, and overheating if the pumping level drops below the motor. Furthermore, the efficiencies of the motors in submersible pumps are less than those of the premium efficiency immersible machines. Maintenance costs (1) depend on the size and operational procedures of the wastewater utility, (2) depend on the number and size of such pumping units, and (3) may be either less or more than those for the wet well–dry well configuration.

A careful study of the alternatives should be made before any of these configurations is selected.

13-3. Fundamentals

The discussion in this section is elementary, and only a hint of the complexity of electrical machinery is given. For a more extensive exposition, see Richardson [3, 4] or other good texts.

When a conductor is moved in a magnetic field, electricity (voltage) is induced in the conductor as shown in Figure 8-1. Because the voltage is able to force current through the conductor into a resistance (or other load), work is done, so energy is required to move the conductor in a generator. If a conductor is formed into a closed rectangular loop and mounted on end pivots so that it is free to rotate in a magnetic field, electrical energy can be converted back into mechanical energy, and the coil becomes a motor. In dc motors, a small part of the electrical input is used to create a strong magnetic field through which the conductor moves. When electrical connections are made to the conductor loop, there is a reacting force between the magnetic field of the conductor itself and the stationary magnetic field. Because the conductor is fastened to a shaft, it can move only by rotating and thus it becomes a motor, as shown in Figure 13-2. By means of a segmented commutator a number of coils are fed with current in a manner that produces a nearly constant rotational force.

The commutator type of motor with a series-connected field can operate on alternating current. The stator, as well as the rotating armature, must be laminated to reduce internal power losses. A constant positive torque is produced because the current in the stator and armature coils reverse together with the alternations of the supply current. This kind of motor typically powers small hand tools.

Squirrel-Cage Induction Motors

The squirrel-cage induction motor operates on alternating current but has no physical electrical connections to the rotor. The rotor consists of a stack of steel laminations with evenly spaced conductor bars around the circumference. The laminations are stacked together to form a rotor core. For motors 250 hp and less, aluminum is die cast in the slots of the rotor core to form a series of conductors around the perimeter of the rotor. Copper, and sometimes brass, is also used as a bar material. Tangential bars, however, are used in large motors. Current flow

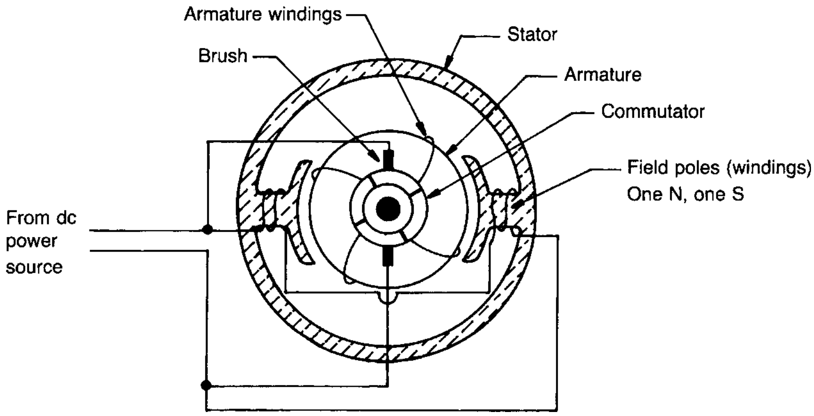


Figure 13-2. Schematic diagram of a direct-current motor.

through the conductors forms the electromagnet. The conductor bars are mechanically and electrically connected with end rings. The rotor core mounts on a steel shaft to form a rotor assembly. There are many designs that provide the shaft-driven ventilation fan in the same integral casting, and one is shown in Figure 13-3.

The stationary part of the motor (stator) is made of laminations of special low-loss steel and is provided with slots to accommodate the stator windings. A small air gap for mechanical clearance separates the rotor and stator. The stator windings are arranged so that when three-phase voltage is applied, the magnetic field advances around the periphery at a rate proportional to the frequency of the ac voltage (see Figure 13-4). This advancing field induces voltages (and thus electric currents) in the rotor conductors. The magnetic field due to these currents reacts with the revolving magnetic field of the stator and produces torque on the rotor. At standstill, or start, the induction motor is identical in operation to a transformer with its secondary winding (see Figure 8-13) shorted together or connected through very low impedance. But as rotation begins, the relative motion between the stator field and the rotor conductors diminishes, and the induced voltage and frequency decrease proportionately. If this machine could rotate at synchronous speed there would be no voltage induced in the rotor conductors because there would be no relative motion between the field and conductors and, hence, no torque to maintain that speed. In practical motors, the difference in rotational speeds between the field and the rotor (slip) can be as low as 2% at the rated load, but 3 to 5% is common in motors smaller than 75 kW (100 hp). Higher-slip motors are manufactured for special applications. The amount of slip depends on the load and on the rotor resistance.

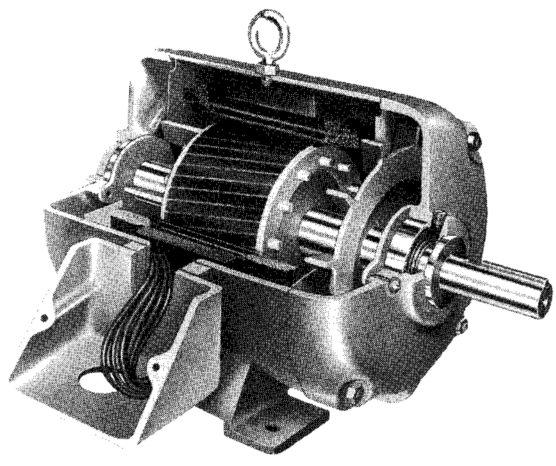


Figure 13-3. Ventilation fan in a squirrel-cage motor. Courtesy of Louis Allis.

When a squirrel-cage motor is started, the inrush current is 5 to 10 (typically, about 6) times the full-load operating current. Some approximate inrush values are given in NEC Tables 430.151(A) and (B) for an assumed locked rotor current of about 6 times the NEC-listed motor full-load current values of the variously rated motors.

Wound-Rotor Motors

The wound-rotor motor is different from a squirrel-cage motor, basically in the design of the rotor and the addition of brushes and slip rings. The rotor windings on a squirrel-cage motor are short-circuited

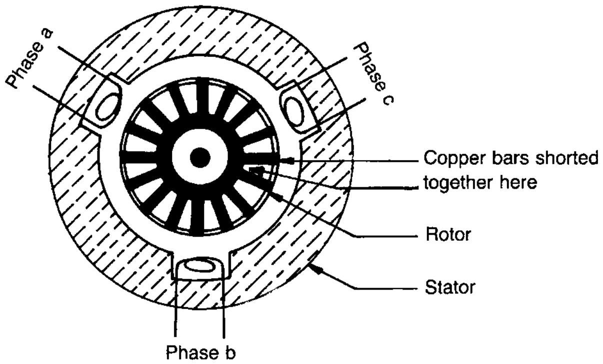


Figure 13-4. Schematic diagram of a squirrel-cage induction motor.

by an end ring. The windings on a wound-rotor motor are not short-circuited, but are brought out via leads to three separate slip rings, which are mounted on the end of the shaft. Stationary brushes ride on each slip ring, forming an external "secondary" circuit in which resistance can be inserted. The resistance of the secondary circuit can be varied by means of switching arrangements external to the motor (see Figures 13-5 and 13-6). The rotor resistance is reflected to the primary (stator) by transformer action, which affects the motor current. Adding appreciable resistance in the rotor circuit at start is desirable to limit the inrush current to the motor while still producing a reasonably high torque, and it is practical to reduce inrush to low values such as 125% of full-load current. Starting a wound-rotor motor without resistance in the secondary circuit produces high inrush currents to flow in the motor windings, causing damage, so direct across-the-line starters cannot be used.

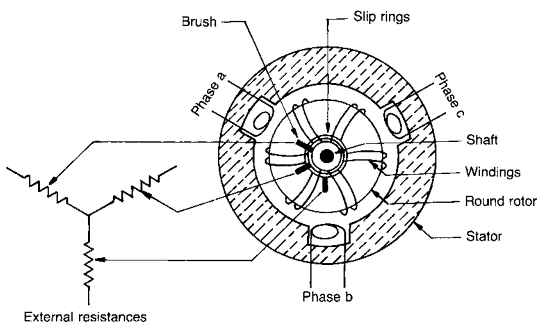


Figure 13-5. Schematic diagram of a wound-rotor induction motor.

Synchronous Motors

Synchronous motors represent the third type of ac motor widely used for large loads (see Figures 13-7 and 13-8). A synchronous motor requires a rotating magnetic field of fixed polarity, which is produced by a permanent magnet (in very small machines) or by a dc electromagnet (in larger machines). The direct current is fed to the rotor through slip rings and is usually derived from external rectifiers. In a more modern version of the synchronous motor (called a "brushless synchronous motor"), a small, externally powered exciter field is used to control a shaft-mounted exciter–alternator. Integrally mounted diode rectifiers are installed to provide the direct

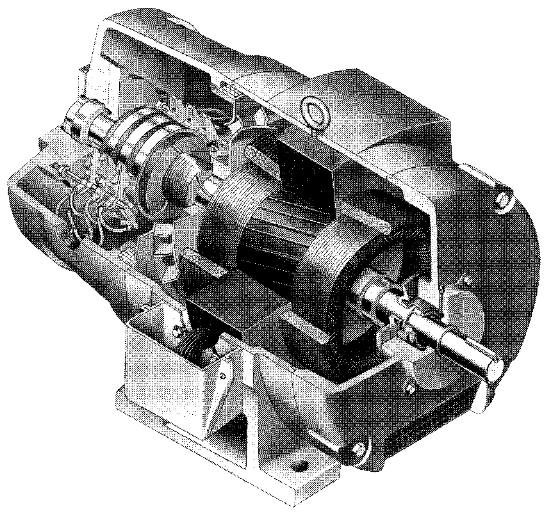


Figure 13-6. A wound-rotor induction motor. Courtesy of Marathon Electric Manufacturing Co.

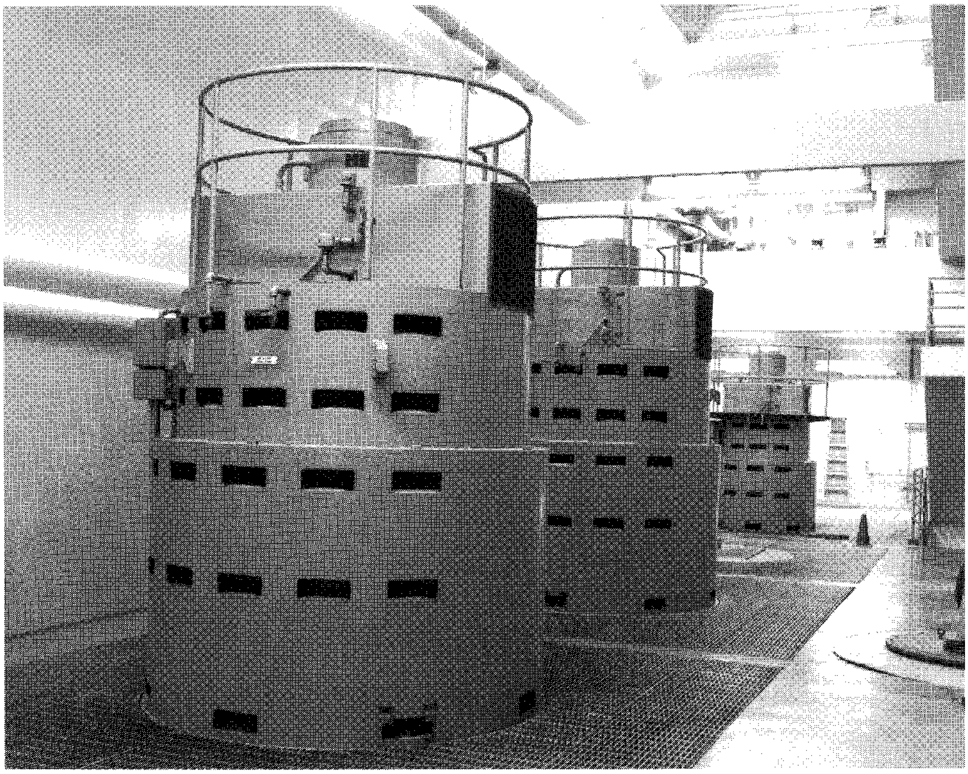


Figure 13-7. Six hundred hp synchronous motor and eddy current coupling for raw sewage influent pump installed in 1962 at Renton, Seattle Metro. Photo by R. L. Niclas.

current to the main motor field. The stator is then connected to the ac power source through contactors and switchgear.

In a typical salient-pole, synchronous induction motor, squirrel-cage type windings are provided in slots in the pole faces (Figure 13-8). These windings serve two purposes: (1) to produce the starting torque required to accelerate the rotor from zero speed to near synchronous speed; and (2) after synchronization, to produce a stabilizing torque that prevents "hunting" of the rotor, which is sometimes the result of changing or pulsating loads. The pole face winding is termed the "damper" (or "amortisseur") winding. A similar winding is also used in the synchronous alternator for stability.

13-4. Types of Motors for Pump Drivers

Only the most common types of motors (squirrel-cage, wound-rotor, and synchronous) used in pumping stations are discussed here.

Squirrel-Cage Induction Motors

Of the three types of motors, the squirrel-cage induction motor is by far the most commonly applied

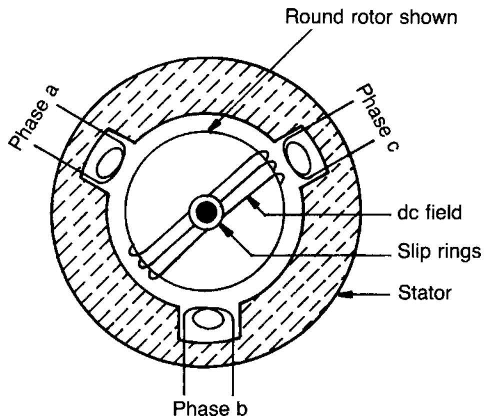


Figure 13-8. Schematic diagram of a synchronous motor.

driver for pumping service because of its (1) simplicity, (2) ruggedness, (3) low maintenance, (4) relatively high efficiency (when well loaded), (5) low first cost, and (6) availability in many standard forms. Because the speed of rotation of the shaft of an induction motor varies only a few percentage points from no load to full load, it is considered a constant-speed driver.

The principal disadvantages of the standard induction motor are: (1) the high starting current (usually about 600% of rated full-load current), (2) the low efficiency at light loads, and (3) the poor power factor, which worsens as the load is reduced. Because each start causes a voltage dip on the line and lights flicker, the high starting current (inrush) may be an annoying problem where a large motor is to be started frequently in a residential area.

Efficiency

An energy-efficient motor is defined as a motor whose performance is equal to or in excess of the nominal full-load efficiency values shown in Table 12-11 of NEMA MG-1 wherein specific full-load nominal efficiency values are given for each horsepower, enclosure type, and speed combination. As shown in that table, percent efficiencies:

- increase from about 80 at 1 hp to about 96 at 500 hp,
- above 5 hp, are nearly the same for speeds from 900 to 3600 rev/min, and are nearly the same for open and enclosed motors.

The Energy Policy Act of 1992 (EPACT) requires that most general-purpose motors manufactured for sale in the United States after October 27, 1997, meet minimum efficiency standards shown in that table between 1200 and 3600 rev/min. The Act applies to 1-through 200-hp general-purpose, T-frame, single-speed, foot-mounted, continuous-rated, polyphase, squirrel-cage induction motors conforming to NEMA designs A and B. Covered motors are designed to operate with 230- or 460-V power supplies, and have open or closed enclosures.

When comparing motor efficiencies, a consistent measure of efficiency should be used. Nominal efficiency is an average value obtained through standardized testing of a population of motors and is the measure recommended. Minimum guaranteed efficiency, which is based on nominal efficiency, is slightly lower to take into account typical population variations and is less accurate because the value is rounded. Other efficiency ratings, including apparent and calculated, are not recommend for use.

In June 2001, NEMA revised the motor standard MG-1 and introduced a "Premium™" efficiency motor standard to meet industry demands for standardization in the categorization of motors with efficiencies that exceeded the levels set by the Energy Policy Act. A premium efficiency motor is defined as a motor whose performance is equal to, or greater than, the nominal full-load efficiency values in Table 12-12 (random-wound) and Table 12-13 (form-wound) of NEMA MG-1-2003.

The extra cost of a premium efficiency motor over a standard motor or standard high-efficiency motor can be quickly offset by the energy savings. Prime candidates for quick payback are motors that run 2000 h or more a year, motors that run at near full load, and motors that run at a constant load (not intermittent, cyclic, or fluctuating). To evaluate efficiency economics, the same method of evaluation should be used on all the alternatives. Two such evaluation methods, simple payback analysis and present worth life-cycle analysis, are explained in great detail in NEMA MG-10-2001.

Wound-Rotor Induction Motors

The wound-rotor motor was particularly effective in the past on applications where a squirrel-cage motor might result in a starting current too high for the capacity of the power system. It was also effective with high-slip loads as well as adjustable-speed installations not requiring precise speed control or regulation. Today's electrical power distribution systems usually have the capacity to start squirrel-cage motors directly across-line, and adjustable-frequency drives (AFDs) have taken over most of their duties. Wound-rotor motors may still be advantageous when large drives (more than 750 kW or 1000 hp) are required. The production of wound-rotor motors has been on the decline for years and will probably continue to decline. Some manufacturers no longer make them, and reduced demand has made them even more expensive.

The wound-rotor motor is now used principally where high starting torque requirements must be met. By using a starter arrangement with a number of steps (each with progressively less resistance in the secondary windings), a wide variety of starting and accelerating currents and torques can be attained. The last step of the starter shorts out all external resistance and results in slip that is only slightly higher than that of an equivalent squirrel-cage motor.

Although the typical starting torque required for centrifugal pumps is usually low, the wound-rotor motor can still be useful in two ways: (1) with the type of starter just described, the starting current can

be kept below a predetermined maximum value (such as 125% of full-load current); and (2) by leaving resistance in the external circuit, the starter can be used as a speed controller because the speed in any step varies considerably with load changes. An adjustable-speed drive is obtained merely by making the control system compensate for speed variations caused by the loading of the motor. Wound-rotor motors are not rated for across-the-line starting; some resistance must be in the external circuit to prevent excessive starting current.

Synchronous Motors

The availability of synchronous motors has been affected by a reduction in demand due to the price premiums required and the increase in the availability of economically priced AFD/squirrel-cage induction motor drives in the higher horsepower ranges. Considering power factor, efficiency, and the more expensive synchronous motor starter, the life-cycle cost is usually less for squirrel-cage motors of up to about 750 kW (1000 hp). Consequently, synchronous motors are rarely applied in drive systems smaller than about 370 kVA (500 hp) because they are not cost-effective in small ratings. The synchronous motor starts the same way an induction motor starts and has the same problems of high inrush during the acceleration, so the same solutions used for induction motor starting must be undertaken to control high starting current. The unique characteristic of a synchronous motor is that it runs at a constant synchronous speed that is determined by the supply frequency. Also, if the field strength (amperes) is externally controlled, the power factor of the line current of the motor with respect to the supply voltage can be varied over a wide range, from lagging to unity to leading. Machines can be specified with unity, 0.8, or 0.6 leading power factors, for which the suitable machine current rating is provided by the manufacturer's design. In large pumping stations, the synchronous motor can sometimes be advantageous due to its ability to be operated with a leading power factor, which thus obviates the need for further power factor correction in the form of capacitor installation or at least reduces the size of the capacitor bank that may be required.

Multispeed Motors

Operating a pump at different specific speeds allows the same unit to be used during dry-weather flow as well as during normal- and high-flow periods. Multispeed induction motors are available for applications requiring more than one speed. There are practical and design limitations on both the number of specific speeds and the speed ratios. Constant-horsepower (not used for pumps), constant-torque (reciprocating pumps), or variable-torque (centrifugal pumps) characteristics are available. Multispeed motors are made with either one or two windings. The speed of the revolving magnetic field is determined by the frequency of the alternating current (expressed in hertz) power supply and by the number of poles on the stator.

$$\begin{matrix}\text{Motor synchronous} \\ \text{speed in rev/min}\end{matrix} = \begin{matrix}\text{Frequency in} \\ \text{Hertz/pole pairs}\end{matrix}$$

By electrically changing the number of magnetic poles (always in pairs), the speed of the motor can be changed. The ends of the stator winding leads are brought out of the motor so they can be connected in multiple arrangements. Single-winding motors are restricted to speed ratios of 2:1 (i.e., 1800/900 rpm, 3600/1800 rpm). Reasonable speed ratios can be obtained with a two-winding arrangement. For example, one winding may provide a four-pole speed of 1800 rev/min while the other provides a six-pole speed of 1200 rev/min with a speed ratio of 3:2. Closer ratios may be obtained by using a lower base speed, such as 1200 or 900 rev/min. One note of caution: The motor starter needs to be matched with the multispeed motor.

Constant- and Adjustable-Speed Systems

There are many ways to provide speed-control devices for constant-speed motors, including:

- Direct current drivers with field control
- Wound-rotor induction motors with rotor circuit resistance control (controlled slip)
- Variable-pitch sheaves (mechanical)
- Hydraulic couplings that produce controlled slip
- Eddy-current couplings that provide controlled slip within the coupling
- Hydraulic pump/motor combinations
- Adjustable-frequency drives.

All of the slip-producing devices, except the expensive wound-rotor slip-recovery system, result in considerable energy loss at the lower speeds. In contrast, the energy losses in the adjustable-frequency and direct-current drives are low. However, reducing the speed of the pump by whatever means often entails a loss of hydraulic efficiency unless the pump is selected according to the principles explained in

Chapter 15, in which variable- and adjustable-speed systems are discussed at length.

13-5. Characteristics of Squirrel-Cage Induction Motors

A number of designs that have significantly differing starting current and torque characteristics are listed by NEMA. The most useful in pumping stations are Design B and Design E motors. Design B is termed "normal starting torque, normal starting current, and normal slip." This is the most commonly applied motor design and is generally used for driving centrifugal pumps, fans, blowers, compressors with unloading start controls, and machine tools. The normal slip at full load is approximately 3 to 5%, and the starting current varies from 500 to 600% of normal operating current. The NEMA Design E motor was created to meet both the needs of higher efficiency and the international standard promulgated by the IEC (International Electrotechnical Commission). Design E motors have higher efficiency than Design B motors but have greater starting current and may not be able to start across the line. A larger starter size and larger branch circuit wiring may be required. Consult the starter manufacturer before applying these motors.

Design D motors are not used for centrifugal pumping operations but may be necessary for auxiliary systems where positive displacement pumps are used. Design D is termed "high starting torque, normal starting current and high slip." The rotor in Design D has a higher resistance than the rotor in Design B. The high resistance in the rotor produces high starting torque, but it also causes high slip (5 to 8% or even 8 to 13%) at full load; the inrush current is the same as for Design B. High slip in Design D causes greater internal loss and, therefore, less efficiency. Nevertheless, hard-starting loads (such as reciprocating pumps and progressing cavity pumps) may require a high-slip motor for satisfactory starting. Typical torque curves for two standard designs are shown in Figure 13-9. Manufacturers can frequently provide special designs to match a particular loading characteristic, but that may mean delayed delivery and expensive or hard-to-find replacements.

13-6. Motor Speed

The speed of the rotor depends on the speed of rotation of the magnetic field produced by the stator currents. The conductors can be wound in the stator slots to produce any desired number of resultant north–south pole pairs around the periphery of the stator. One pair of poles in a three-phase winding is illustrated in Section 13-3. One cycle of sinusoidal alternating current represents 360 electrical degrees and during this time the magnetic field rotates 360 mechanical degrees or completely around the stator. Because 60 Hz ac produces 60 revolutions of the field per second (or 3600 rev/min), a one-pole pair

LIVE GRAPH
Click here to view

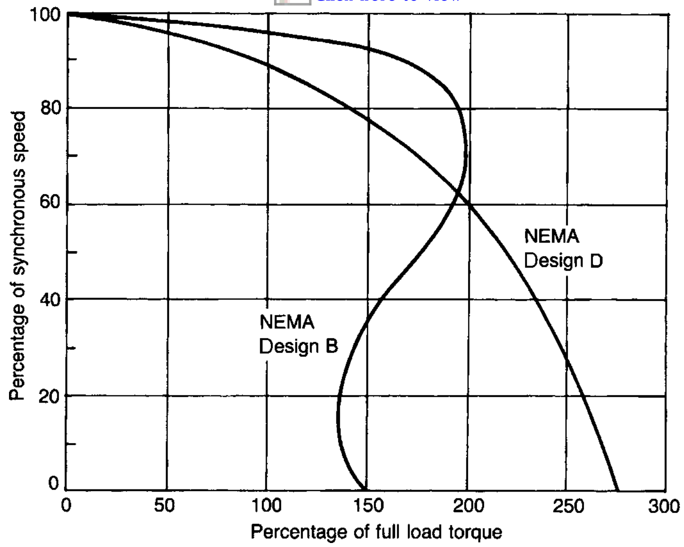


Figure 13-9. Speed-torque curves of two NEMA design motors.

machine (commonly termed a two-pole machine) rotates 3600 rev/min at zero slip. An equation for synchronous speed in SI units is

$$\omega = \frac{2\pi Hz}{n} \tag{13-1a}$$

where ω is radians per second, Hz is current frequency in cycles per second, and n is the number of poles (which is twice the number of pole pairs). In U.S. customary units,

$$\omega = \frac{120Hz}{n} \tag{13-1b}$$

where ω is revolutions per minute and the other terms are as defined above.

Preferred motor speeds for pumping station equipment depend on the characteristics of the driven equipment as well as the overall costs—in particular, the life-cycle costs of the various practical alternative speeds. As a general recommendation, the following guidelines apply to pumping station equipment:

- Blowers and fans: use 1800- or 3600-rev/min motors with speed increaser gears when higher speeds are required. Note, however, that high speed produces a high noise level and 3600-rpm fans do not last nearly as long as 1200-rpm or 1800-rpm fans.
- Water, wastewater, or sludge pumping: use 900-, 1200-, or 1800-rev/min motors for medium to small pumps and speeds up to 600 rev/min or less for special, low-head, large pumps. Motors that run at 1800- and 1200-rpm are usually stock items, while 900-rpm and slower motors are not.
- Progressing cavity screw pumps: use 900-rev/min motors (or less). Consider connecting the motor to the pump through a belt or gear speed reducer.

Note that European motors are designed for 50 Hz. Operating these machines on 60 Hz will probably cause them to overheat. Likewise, do not operate a 60-Hz motor on a 50-Hz system unless the manufacturer specifically warrants the motor at the reduced frequency at some specific voltage and load.

13-7. Motor Voltage

Typical distribution system voltages are given in Table 8-6 as single-phase (120 or 240 V) and three-phase (208, 240, 480, 600, 2400, 4160, or 12,000 V). These voltages are the levels that the utility attempts to maintain at its point of service.

Voltage drops occur in the user's electrical system, and these are divided into two parts according to the NEC. The first drop is in the feeders and includes the service switchgear and the cables delivering electrical energy to local distribution centers, which are typically motor control centers (MCCs) in pumping stations. The second voltage drop cited in the NEC is in the branch circuit, which includes all of the power wiring between the circuit protective equipment (typically the MCC) and the motor. A 5% voltage drop is allowable in the total system and a maximum drop of 3% in either the feeder or the branch circuit is recommended by the NEC. For example, on a system rated at 480 V line-to-line, a 5% drop is 24 V and the net voltage delivered is 456 V line-to-line (at the full rated load). The designer may divide the drop in accordance with the NEC rules or choose to put in cables heavy enough to make the drop somewhat less. It is good design practice to keep the total calculated drop and the individual parts of that drop well below the specified levels discussed. At light loads, the terminal voltage at the motor is likely to approach the service voltage or to be approximately 480 V. The motor manufacturer usually rates the motor (full-load current and horsepower) at 460 V with a $\pm 10\%$ tolerance at full load. Thus, the motor guarantee would be valid at 506 V, but the life of the motor at 506 V may be shortened. A motor rated with a service factor may not be guaranteed at a voltage other than the nameplate voltage.

Typical rated voltages for motors are 115 or 230 V (single-phase), and 200, 230/460, 460, 575, 2300, and 4000 V (three-phase). The supply voltage for a very small pumping station may be 240 V, three-phase, three-wire from a utility delta-connected system, and a 120/240-V supply can be provided from a center tap between two phases on the utility's transformer. Sometimes the utility may supply a 208Y/120-V, three-phase, four-wire system, particularly if the pumping station is located in a network area having this supply. It is also frequently available where an individual transformer is required for the pumping station. This multivoltage system is preferable to a 240-V, three-phase system because it enables the load to be balanced on all three phases. Pumping stations of medium size—for example, 75 to 600 kW (100 to 800 hp)—need a full 480-V supply and usually derive their own 120-V circuits for lighting, small power outlets, and control devices from dry-type transformers located in the station.

Ordinarily, the service for a large station is most economical at the highest voltage available, and large pumping stations require 4160-V service. Because motors are usually rated at 4000 V maximum,

however, there is no advantage in utility service above 4160 V unless the utility rate schedule for higher voltage service is particularly appealing to the owner. Higher service voltage in such situations would reduce cable size for power distribution, but transformers would be required to provide an operating voltage of 4160 or less, and they are costly. Note that some 230-V motors are guaranteed by the manufacturer for operation on 208 V, but some manufacturers then qualify this rating as not necessarily meeting NEMA standards. Therefore, use 200-V motors for 208-V systems.

A study of alternative voltage levels and sources should precede the selection of both service voltage and utilization voltage. For example, 480 V is nearly always the preferred and most economical level for both service and distribution in medium-sized pumping stations with short feeders and service runs. However, if the cable run from the service point to the pumping plant is, say, 150 m (500 ft) or longer, the excessive cable costs at 480 V can make 2400- or 4160-V service more economical. The cost of in-plant transformers versus utility-furnished transformers should be studied carefully on a life-cycle basis.

Service from a utility circuit is usually obtained through pole-mounted fuses, but in the largest stations, service may be obtained from a utility circuit breaker. In addition to the loading recommendations given here, make sure that the utility provides a balanced system voltage of rated value. In addition, require overload relays in each phase of the motor branch circuit to remove the motor from operation in the event of "single-phasing" (which occurs when a fuse on one leg blows). The prevention of single-phasing begins in the utility's system and carries throughout the pumping station electrical system.

Wherever single-pole switching equipment is used, install current-balance relays and circuit breakers with interpole tripping to cut station power in the event of an unbalance greater than the relay setting allows. Never install circuit breaker handle ties, even in the smallest stations. Some unbalance is always present due to lighting or single-phase auxiliary equipment, and the current-balance relays must be set to accommodate this unbalance. If single-phase loads are large enough to cause voltage unbalance, distribute these loads between the phases as equally as possible.

Voltage unbalance causes excessive heating of motors. A 2% unbalance, according to IEEE Standard 141, adds about 8% to the heat in the motor windings and causes the temperature to rise from 80 to 86.4°C. A 3.5% unbalance causes an approximate 25% increase in heating, so the temperature rises (in a Class B insulation, T-frame motor) from 80 to 100°C.

13-8. Enclosures

The motor enclosure provides the environmental protection for the motor, a mounting space for the stator windings, the magnetic circuit for the stator field, a ventilation space, and a mounting for the bearings on most models. Motor enclosures, which are defined in Section 13-1, are:

- Drip-proof
- Splash-proof (if not available, substitute weather-protected Type I)
- Totally enclosed (TE)
- Totally enclosed, fan-cooled (TEFC)
- Totally enclosed, nonventilated (TENV)
- Totally enclosed, pipe-ventilated
- Weather-protected, Types I and II
- Submersible
- Immersible
- Explosion-proof.

The open drip-proof and totally enclosed fan-cooled enclosures are the ones most commonly used in pumping stations. Small- to medium-sized open motors may have epoxy-encapsulated windings, and they give excellent performance in quite severe conditions. They are the least expensive to put back into service in the event of flooding. Nonencapsulated, drip-proof motors are not usually applicable for pumping station environments. Horizontal machines are particularly vulnerable to hosing-down operations.

The totally enclosed, fan-cooled motor usually has the fan mounted integrally with the motor under a shrouded-end bell arrangement. Air is drawn into the end of the motor and distributed over the outside of the motor enclosure by the fan. The cooling ability of a TEFC motor is less than that of a similar open, drip-proof motor. The TEFC motor may be used in both indoor and outdoor environments. Horizontal TEFC motors should be provided with drain holes or a breather drain in the bottom of the enclosure to permit the escape of condensation should it collect inside the enclosure. The shaft, of course, extends through and beyond the enclosure.

The totally enclosed, nonventilated motor is usually used only for small, noncritical loads.

Totally enclosed, pipe-ventilated motors are likely to be used in adjustable-speed systems where, at low speeds, insufficient cooling would be available from the ordinary shaft-mounted fan. This type of ventilation may also be appropriate where the motor is subjected to higher than normal ambient temperatures but where cooler air is available.

Weather-protected Type I motors should always be considered for large drivers in ratings where TEFC is

not a standard option. It is also applied indoors in some installations where pipe breaks may spray the equipment. Use weather-protected Type II motors in outdoor applications subject to high wind and rain.

Only explosion-proof motors can be used in areas that are classified as Class 1, Division 1 hazardous locations in Article 500 of the NEC. These motors are similar to the TEFC in cooling, but are much more expensive than the TEFC motor.

Submersible motors are often used in water wells and in many wastewater pumping stations. Most models must be totally submerged to cool properly, and because the motors are "ordinarily" submerged, a submersible rating is adequate. However, in the event of a malfunction and subsequent overheating combined with possible explosive gas accumulation in the manhole, an extreme hazard may be created. Some motors can be run intermittently or continuously in air. For additional discussion, see Section 13-2 and Chapter 25.

Immersible motors are TEFC motors in special housings that, unlike submersible motors, are designed to run in air. They can, however, be operated for at least two weeks submerged 10 m (30 ft) under water. In 2004, immersible motors from 11 to 1300 kW (15 to 1750 hp) were available [5]. Larger motors may become available. See Section 13-14 for a more complete description.

13-9. Insulation

Motor windings, by necessity, are closely packed into slots that are designed to accommodate them. The methods used to maintain conductor-to-metal, conductor-to-conductor, and phase-to-phase insulation vary greatly depending on the size and application of the motor. Pumping station designers and motor specifiers are more concerned with the class of insulation than with the details of packaging.

In addition to other considerations, the type of insulation that should be specified for a pump driver motor depends on its suitability in a possibly moist atmosphere, its ability to be dried readily in the event of a submersion, and the ambient temperature conditions that will prevail in its location. Insulation is ordinarily classified by letter designations that infer its ability to withstand continuous operation at a defined total temperature (or some rise above a standard ambient temperature). The letter designation is also defined in terms of the types of materials used in the insulation system.

The allowable temperature rises above a 40°C ambient temperature for the various standard letter designations and enclosure types (taken from NEMA MG-12.42) are summarized in Table 13-2. All of the temperatures in Table 13-2 are measured by resistance in accordance with NEMA standards. If the ambient temperature is higher than 40°C, a derating equation is given in NEMA MG-1 for the allowable temperature rise (see the latest revision of NEMA MG-1). The insulation materials for the several classes are:

- *Class A:* Combinations of materials such as cotton, silk, and paper impregnated or immersed in a dielectric liquid, no longer specified for pumping station applications and now encountered only in very old drivers.
- *Class B:* Combinations of materials such as mica, glass fiber, asbestos, and so on with bonding substances.
- *Class F:* Similar to Class B, but designed for a higher operating temperature at the same thermal life.
- *Class H:* Similar to Class F, except for the use of component materials (such as silicone elastomers) rated for a higher operating temperature at the same thermal life. Note that Class H insulation may be unavailable, and its thickness causes design problems. In the southwest, idle motors in the open reach temperatures of 71°C (160°F) due to direct sunlight. Because motors with standard insulations (Class A, B, or F) can burn out quickly, either (1) protect motors from sunlight, (2) use water-cooled motors, (3) use Class F insulation and derate the motors, or (4) best of all, use Class H insulation regardless of its cost.

Epoxy insulation was used in the past to permit an inundated motor to be put into service almost immediately after the dry pit was dewatered. But epoxy is

Table 13-2. Allowable Temperature Rise for Insulation in Motors[a]

	NEMA insulation letter designation			
Motor enclosure	A	B	F	H
Totally enclosed, fan-cooled	60	80	105	125
Totally enclosed, nonventilated	65	85	110	135
Encapsulated with 1.0 SF	65	85	110	–
All other motors, with 1.15 SF	70	90	115	–
All other motors	60	80	105	125

[a]Degrees Celsius above 40°C ambient from NEMA MG-1 Table 12.42.1.4.

short-lived because of the cracks that develop. Use Class F or H insulation and take the time to dry a previously submerged motor properly. Alternatively, use immersible motors.

Although thermal life is not quantifiable by aging time alone, modern insulations (Class B, F, or H) can be expected to last 25 years at conservative temperatures unless they are subject to extreme vibration or extreme thermal cycling.

Always select water or wastewater pump motors for long life. Furthermore, carefully control the temperature rise aspects of the insulation system by specifying the best of materials (within reason). Then take advantage of every means available to obtain an efficient motor with a low temperature rise at the specified loading conditions. A practical and economical way of buying a long-lasting motor is to specify Class B temperature rise but require Class F or H insulation.

Along with the insulation specification, the motor (if available in the enclosure specified) should have an applicable service factor (SF)—typically 1.15 for most sizes. Also specify that the maximum pumping horsepower load must not exceed 85% of the service factor rating (about 98% of the nameplate rating). If the motor type does not have a service factor rating available, then the maximum loading should not exceed 90% of the nameplate rating. This recommended maximum loading is not low enough to cause undue efficiency or power factor loss in a well-designed machine.

The overheating of insulation, whether due to overloading, insufficient ventilation, high ambient temperature, or too-frequent starting, can contribute rapidly to the shortening of insulation life. An old rule of thumb is that insulation life is halved for every 10°C increase above the rated value. The rule is not precise and may not apply to present-day insulation, but it is a good guideline when a decision must be made on whether to use a motor above its rating during an emergency or to replace it immediately with an adequately rated unit. Note that the rule applies to a continuous load. It does not mean that an occasional overload of short duration will halve the life of the insulation. Common sense dictates that conservative loading practices should be maintained in pumping station operation and design. See the IEEE standards for more information on insulation aging.

13-10. Squirrel-Cage Motors

Service Factors

Although the service factor of a motor implies that the entire design of the motor allows for the SF loading, the most sensitive factor is the stator winding temperature. Because the allowable temperature of the conductors is limited by the type of insulation used, the insulation system is the determining factor in the allowable temperature rise. A motor with a service factor rating of 1.15 (standard for most motors for which an SF is available) must therefore be designed for the nameplate load operation at an insulation temperature of somewhat less than the limiting value given in Table 13-2. The service factor stated on the motor nameplate is based on a sinusoidal voltage source. For a motor used on a nonsinusoidal voltage source such as an inverter drive, the service factor must be derated (e.g., from 1.15 to 1.00).

If the motor is selected according to the advice given in Section 13-9 and loaded only to 85% of the service factor rating, the operating temperature of the insulation will probably be low enough to allow a long life for the windings.

Insulation and Service Factor

Insulation systems and service factor are very closely related. The maximum loading of the motor can be easily controlled by the specifier and pump manufacturer. However, it is not ordinarily in the best interests of the owner to load motors too lightly under maximum operating conditions because of the inherent decrease in motor efficiency and power factor. The major considerations that provide long insulation life are those required to keep the temperature of the insulation below its maximum rated value.

Ambient Conditions

Do not place the motor in a closely confined space. It needs “breathing room.” If the motor must be in an inherently dirty area, specify the proper motor enclosure for the conditions and describe in the O&M manual the advantage and necessity of frequent inspection for clogged air passages and accumulations of dirt and debris around the motor.

Responsibility

If a single manufacturer is made responsible for the entire pumping unit (pump, frame, shafting, motor, and controller), the motor is selected on the basis of the complete operating conditions presented by the project engineer (see Chapter 16 and Appendix C). The responsible project engineer must always make independent calculations, as shown in Example 13-1.

Example 13-1
Motors for a Wastewater Lift Station

Problem: Preliminary investigations for a wastewater lift station have led to the following conditions:

- Flows: 800 gal/min minimum, 1600 gal/min design, 3600 gal/min maximum, and 4500 gal/min future maximum
- Static lift: 25 ft at low wet well level, 20 ft at high wet well level
- Friction head losses at 4500 gal/min: 15.7 ft at Hazen–Williams $C = 120$ and 13.4 ft at $C = 145$ (including "minor" losses within the pumping station).

Four (three duty, one standby) 705-rev/min pumps were selected to meet the various conditions of head discharge in the following tabulation. Each constant-speed pump is rated at 2100 gal/min for 30 ft TDH at 79% efficiency.

	Low wet well ($C = 120$)		High wet well ($C = 145$)		
Pumps operating	Head, ft	Flow rate, gal/min	Head, ft	Flow rate, gal/min	Minimum cycle time, min
1	28.0	2300	24.0	2700	6.7
2	33.5	3500	31.0	4100	10
3	37.3	4100	35.3	4800	17

Select the drive motors and size the cables.

Solution: For public works, design for a generic product. Standardization in manufacturing makes it easy to design and specify on the basis of NEC, NEMA, IEEE, and ANSI criteria.

Required power. From Equation 10-6b, the fluid horsepower is

$$hp = \frac{qH}{3960} = \frac{2300 \times 28}{3960} = 16.3 \text{ or } hp = \frac{2700 \times 24}{3960} = 16.4$$

But the efficiency of the pump is $\approx$ 79%, so the output (shaft) motor horsepower must be

$$hp = \frac{16.4}{0.79} = 20.8$$

The motor power requirements for each of the conditions in this tabulation range from 16.4 to 20.8, so choose 25-hp motors, which can meet all conditions. Confirm selection against the manufacturer's pump curves. Confirm that motor is nonoverloading throughout the entire design flow range.

Required torque. Horsepower at full speed is not the only criterion. The motor must develop enough torque at all speeds to exceed the resisting load by a comfortable margin; otherwise the motor will never reach full speed or will stall. The equipment suppliers usually provide the pump and the motor, so they will match the motor performance with the load characteristics of the pump.

To compare the torques, plot the speed-torque curves of the pump and the motor on the same graph, as in Figure 13-10. If motor torque everywhere exceeds pump torque by, say, 15%, the motor can develop full speed quickly. Usually this step is not necessary if the pump manufacturer is responsible for selecting the motor and has been provided a definitive set of hydraulic conditions.

Motor selection. Speed is determined by load speed unless some form of speed reducer is used. For certain custom motors, performance data are not published and must be obtained directly from the motor manufacturer (e.g., full-load amps of a low-speed motor). To determine whether a Code F motor is available as a standard unit (Code G may be standard),

consult motor manufacturers. The motor is to be mounted high above the floor, so an open, drip-proof enclosure is satisfactory. Alternatively, a TEFC motor might be preferred for some protection during washdown—certainly if the motor is mounted at floor level.

Starting. Obtain written approval from the electric utility for frequent line starting of one motor at a time (assuming there are controls for automatic sequencing) and ascertain the maximum motor horsepower they will permit for across-the-line starting. Consider the following scenarios:

- Direct across-the-line starting
- A limited number of across-the-line starts per day
- Across-the-line starts not permitted, but reduced-voltage starting is allowed
- No across-the-line starts and only a limited number of reduced-voltage starts are allowed, per day.

The last three scenarios may govern motors of 50 hp or more. If the final scenario applies, consider adjustable-speed drives.

Load calculation. Assume the service available is 480 V, three-phase, and 60 Hz, and that the length of service run is 100 ft. A Code F motor has an inrush value in the range of 5.0 to 5.59 kVA/hp from NEC Table 430.7(B).

First, size the feeder cable for current. Assume (1) 5 kVA for miscellaneous (balanced three-phase) loads, and (2) three 25-hp duty motors. For the 5-kVA load, the current is calculated from Equation 8-6. Assume the power factor is 1.0.

$$P_{3\phi} = \sqrt{3} \times V_L \times I_L \times Pf$$
$$5 \text{ kVA} \times 1000 = \sqrt{3} \times 460 \times I_L, \text{ so } I_L = 6.3\text{ A} \qquad (8\text{-}6)$$

Motor full-load current (FLA) for 3600- and 1800-rpm, single-speed motors is selected based on NEC Table 430.150. In this instance the motor rpm = 720 rpm (synchr onous) and therefore nameplate current may be used. As this is design, there is no nameplate to read. The motor manufacturer will need to be contacted to obtain full-load amps at this speed. Assume FLA = 39.0 A at 720 rpm.

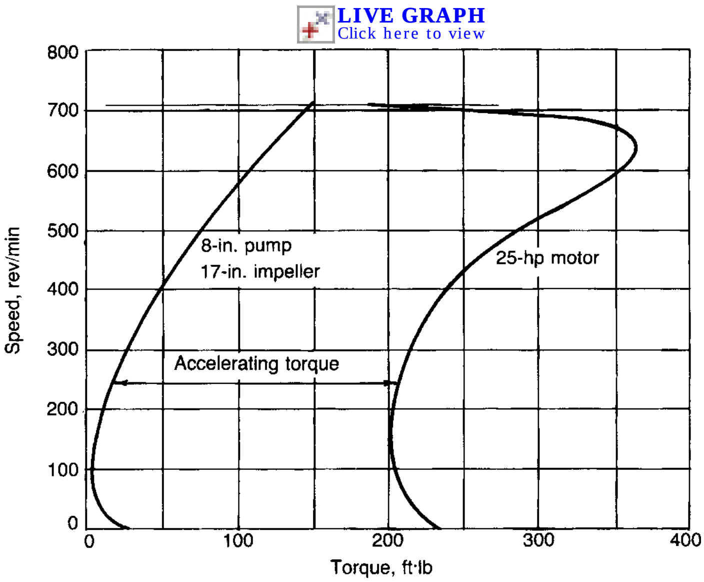


Figure 13-10. Torque curves of the motor and pump in Example 13-1.

At this point it is appropriate to note that choice of a 720-rpm motor may not be the most cost-effective. Note that 720-rpm motors are considered custom and, compared to an equivalent 1800-rpm motor, cost four times as much, are larger, are less efficient, and have a poorer power factor. Another disadvantage of the custom motor is the long lead (purchase to delivery) time.

From the principles given in Chapter 9, the total branch circuit load is:

• Miscellaneous:	6.3 A
• Motor No. 1:	39.0
• Motor No. 2:	39.0
• Motor No. 3:	39.0
• Motor No. 4 (standby):	0.0
• 25% of the largest motor:	9.75
Total:	133.05 A

The feeder would be designed for 150 A (next size circuit breaker, see NEC 240.6) at full load if no expansion were contemplated, but a future expansion of 25% would increase the amperage to 133 A × 1.25 = 166 A, so select a cable to carry 175 A (next size circuit breaker, see NEC 240.6). Inrush persists for such a short interval that it is ignored in the load calculations. However, inrush does produce voltage drop and cannot always be ignored. Conservative engineers may limit the voltage drop at the motor terminals to 12 or 15% on starting, which provides a good, solid electrical system throughout. The drop is approximately 4% in this example.

The motor branch circuit cable would be designed for a motor FLA = 39 A plus an overload of 25% (i.e., minimum cable ampacity = 39 A × 1.25 = 48.8 A).

Second, size the cables. From NEC Table 310.16, choose THHW-, XHHW-, or RHW-rated copper cable (75°C rating). From the "Copper/75°C" column, a 175-A load requires three #2/0 AWG conductors (one per phase) for the feeder cable, and a 48.8-A load requires three #8 AWG conductors for the motor branch circuit. Grounding conductors are selected from NEC Table 250.122. From the "Copper" column, the 175-A circuit requires an accompanying #6 AWG conductor and the 48.8-A circuit requires a #10 AWG conductor.

Third, check for voltage drop. Find the voltage drop from Equation 13-1 (a close approximation):

$$\boldsymbol{\Delta V_{\mathrm{L}}} = \sqrt{3} \times \boldsymbol{I_{\mathrm{L}}}(R \cos \theta + \boldsymbol{X} \sin \theta)\boldsymbol{L} \tag{13-1}$$

where ΔV_{L} is the line-to-line voltage drop, I_{L} is line current in amperes, R is resistance in ohms per 1000 ft, cos θ is the power factor, X is inductive reactance in ohms per 1000 ft, and L is length in 1000-ft units. If the power factor is 80%, cos θ is 0.80 and sin θ is 0.60. From Table 9 in Chapter 9 of NEC, R at 75°C is 0.10Ω/1000 ft and X is Ω/1000 ft for 2/0 AWG in steel conduit. Either 133 or 150 amp can be correctly substituted for I, but at the ultimate load of 175 A

$$\mathbf{\Delta V_1} = \sqrt{3} \times 175 \times (0.10 \times 0.8 + 0.054 \times 0.6) \times 100/1000 = 3.4\,\mathrm{V}$$

The voltage drop percentage = 3.4/480 = 0.7%.

Double-check using IEEE Standard 141, Table 11 for copper conductors in magnetic conduit to find $1.9\,\mathrm{V} \times 10^{-4} \times \mathrm{A} \times \mathrm{ft} = 2.3 \times 10^{-4} \times 175 \times 100 = 3.32\,\mathrm{V}$ (a fair check with 3.4 V).

The voltage drop of 0.7 % is negligible compared with the NEC 215.2 FPN No. 2 and 210.19 (A) FPN No. 4 limit for feeder or branch circuits of 3%. Because a 2% voltage drop is the maximum for good design, the maximum length of the service run should not exceed $100(2/0.7) \approx 300\,\mathrm{ft}$.

Critique of Example 13-1

The maximum permissible length of the service feeder in Example 13-1 is about 300 ft if the voltage drop is limited to 2%. Because of the high cost of the cable and conduit (concrete-encased duct), it is wise to negotiate with the utility to place transformers closer to the load (say, 30 m or 100 ft maximum), regardless of whether the utility or owner provides the service cable.

Starting a 25-hp motor is not likely to cause problems unless the utility's power lines are loaded to the limit. But if the motors were, say, 250 hp, the utility might object to frequent line starts, so consider reduced-voltage ("soft") starting (which might reduce the inrush to 300 or even 150% of the full-load current) or consider adjustable-speed drives.

If repeated reduced-voltage starting is allowed by the utility, compare costs and space requirements of autotransformer start with those of solid-state, soft-start controls. Solid-state controls have a much greater selection of starting currents than do other methods, have reduced space requirements, and costs are becoming more competitive. For a more detailed discussion on motor starters, refer to Section 8-3. The addition of soft starting only affects the installed costs, whereas the addition of adjustable-speed drives is likely to involve a complete redesign of the station using fewer pumps (two duty and one standby, for example) and smaller wet and dry wells. Resulting cost savings derived from this redesign could partly offset the cost of the adjustable-speed drives. Adjustable-speed operation might be more cost-effective because it extends the life of the motors and less energy is used (see Example 29-1). However, compared to constant-speed operation, it is somewhat less reliable and there is increased maintenance of the adjustable-speed devices. For a more detailed discussion on variable-speed drives, refer to Chapter 15.

13-11. Frequency of Motor Starts

Motor life is determined primarily by the temperature of the stator winding insulation and the length of time it is sustained. From motor manufacturers' data, a motor operated continuously at rated load with normal voltage and frequency in a 40°C (104°F) environment will last about 40,000 hours. A general rule is that motor insulation life is halved for every 10°C (18°F) rise in temperature. The converse is also true. For example, a TENV motor running at full load with Class F insulation is rated to have a temperature rise of 110°C (198°F) with a maximum ambient temperature of 40°C. If, however, the internal temperature rise is limited to 100°C, the insulation life is approximately doubled.

Motor stresses (thermal and mechanical) are high during the starting period and affect motor life. When an induction motor is started across-the-line, the stress on the windings is increased, and the inrush current in the rotor is about six times normal current until operating speed is reached—usually 3 to 6 s for a NEMA Design B motor driving a centrifugal pump. The inrush current heats the rotor and stator windings and causes the stator winding insulation temperature to increase. If the motor is started too frequently and is not allowed to cool down between starts, heat buildup in the stator dries out the insulation and causes it to become brittle, leading to ultimate failure.

When designing a pumping system, the decision needs to be made as to the maximum number of starts per hour required (i.e., whether to stop the pump or let it continue to run at no load). To make this decision, a number of other factors must be considered:

- Type of motor
- Motor horsepower
- Motor full-load speed
- Power utility restrictions on inrush current or frequency of starts
- Power utility demand (kVA) charges
- Reductions in expected motor life due to the frequent starting.

The frequency of motor starts is given by NEMA in both MG-1 and MG-10. The two publications approach permissible frequency of motor starts differently. MG-1 provides guidance on the number of successive starts (i.e., two starts from ambient or one start from rated load operating temperature). MG-10 provides guidance on the repetitive start-run-rest cycles applicable to pumping station operation. Because pumps are low-inertia machines, and because there must be a balance between first cost and service life, it seems reasonable to use about two-thirds of the starting frequency calculated from MG-10. The allowable frequency of motor starts is, however, far more complicated than is indicated by such a calculation, because (1) an under-loaded motor may be started more frequently than a fully loaded one; (2) the required frequency of starts in a pumping station with constant-speed motors is based on the assumption that inflow to the wet well is exactly one-half of the pump capacity, whereas half of the time, the required starting frequency is nearly 40% less (see Figure 12-26); (3) the specified severe

ambient temperature conditions may persist for only short periods of time; and (4) motors can be custom designed and built to withstand many more frequent starts than standard motors can. The most important factor relating to the frequency of pump starts is whether (for multiple pumps in a station) automatic sequencing is used. Some engineers do not favor automatic sequencers and prefer manual selection of lead and follow pumps for better control of pump wear. Programmable logic controllers (PLCs) can be programmed to alternate the lead pump at every cycle reliably, and the reliability can be increased by automatic self-testing and switching to a backup PLC if a malfunction occurs.

Soft (reduced-voltage) starters typically induce an inrush current that is only about 250% of normal operating current. Most solid-state starters today come with a built-in bypass contactor to bypass and turn off the solid-state controller when the motor is near full speed. This contactor is not to be confused with a backup bypass contactor, which is sometimes used in the event of adjustable-frequency drive failure. The pumping process has to be capable of constant-speed operation; otherwise, the use of a back-up bypass contactor is worthless. The built-in bypass contactors are not rated to start the motor direct across-the-line, whereas the back-up bypass contactors have to be. Note that soft starters cost considerably less than adjustable-frequency drives (AFDs), but AFDs not only provide soft starts and controlled acceleration and deceleration (reducing water hammer), but also make it possible to reduce the size of the wet well, usually reduce the power lost in overcoming pipe friction, and prevent the sudden changes of flow that tend to upset downstream wastewater treatment.

Motors 224 kW (300 hp) and larger are usually custom engineered to whatever requirements are specified. Hence, large motors can be specified to provide whatever frequency of starts is needed for the pumping station. However, if the frequency of starts is not specified, the motor will be designed to provide the same frequency of starts as small and medium-sized motors (i.e., standard).

Submersible motors, cooled by the pumped liquid, can usually withstand very frequent starts—sometimes much more frequently than 20 starts per hour. For such motors, it is generally the starter (not the motor) that limits the frequency of starts.

Because objective decisions concerning allowable frequency of motor starts must be based on such a myriad of site-specific factors, the pumping station designer's best course is to consult the pump manufacturer, who deals with such problems constantly and can temper theory and calculations with practical experience. The same cannot be said of motor manufacturers because they rarely seem to be cognizant of the needs of pumping stations.

13-12. Miscellaneous Motor Features

Miscellaneous, but important, features to be considered in the selection of a motor include the shafts, bearings, space heaters or winding heating, temperature sensors, and (possibly) vibration monitors.

Shafts

Motor shafts are usually solid and extend beyond the enclosure in order to accept the coupling. The shaft may be ordered with standard NEMA dimensions in either short or long shaft extensions. For municipal work it is usually unnecessary to specify shaft length unless the installation is for a nonstandard arrangement.

Hollow shafts are usually supplied for the vertical motors for deep well pumps. The shafts extend through both ends of the motor with an adjustable thrust bearing above the motor frame to the deep well turbine pump below it. A nonreverse ratchet is required for deep well pumps where threaded-end sectional drive shafts are used. The ratchet prevents back-spin caused by the discharge water column if the drive should ever stop against a high head and the check valve fails. Back-spin could unscrew the connections between shaft sections and the deep well turbine pump below it. Nonreverse ratchets are also useful for other pumping arrangements, such as horizontal pump motors, because energizing a back-spinning drive can break the shaft, tear windings loose, or overheat the motor. When a ratchet is set, the locked rotor causes large line currents to continue beyond the normal starting period, so the overload relays must operate to prevent damage to the motor.

Bearings

Antifriction ball bearings are used in most motors. Recently, improvements in materials have added to the life expectancy of what was already an excellent product.

Bearings for vertical motors may be angular-contact, grease-lubricated, or oil-lubricated for higher-speed applications. Spherical roller bearings can withstand very high thrust. For the highest-thrust

application on very large drives, the Kingsbury thrust bearing [6] is available. It is expensive but has an extremely long life.

Insulated bearings should be specified on all AFD motors. Consult the motor manufacturer for availability.

Space Heaters and Winding Heating

Space heaters are often applied to motors located in damp areas or outdoors where the ambient temperature and humidity vary over a wide range. The space heater is sized to maintain a reasonable temperature in the motor enclosure to ensure that the windings and insulation cannot collect moisture. A motor in standby service or one that operates only at infrequent intervals must have the heater on almost all of the time. Heater circuit continuity is seldom checked, and sometimes it seems that there are as many strip heaters that do not operate as ones that do.

An alternative to the space heater (but initially much more expensive) is a low-voltage heating circuit for the motor winding. The most modern motor winding heater is a solid-state device that does not use a transformer [7]. The safety of this type of circuit, however, may be questioned by the Authority Having Jurisdiction (AHJ), so consult that entity prior to completing the design. A UL listing of such equipment usually ensures the AHJ will approve the circuit if installed according to the listing stipulations. If the low voltage is to be produced by a transformer, the transformer is placed within the motor starter enclosure, and a separate contactor and spare contacts on the motor starter are also required [8, 9]. A timing relay is needed to ensure that the motor voltage has had time to decay before connecting the low-voltage heating circuit to the stator of the motor. The temperature of the windings should be kept about 10°C above the ambient temperature. Low-voltage heating of the motor windings has proved to be very practical for motors less than about 150 kW (200 hp) where there is often little or no room for mounting a conventional space heater.

Temperature Sensors

Winding temperature sensors provide back-up protection for over-temperature conditions within the motor. The simplest kind is a direct-acting, bimetallic element that snaps from one position to another and either opens or closes a contact. Usually this contact is used as a control-circuit-stop contact (similar to the overload relay contact and stop pushbutton). This contact resets automatically on a decrease in temperature, so arrange the circuitry to prevent automatic restarting of the motor. The worst thing that can be done to a motor already in trouble from overtemperature is to cycle it on and off at short-duration intervals.

A more sensitive device that may be applied to any size of motor is the positive temperature coefficient (PTC) thermistor. It is a very small resistance element taped in the interstices of the end turns of the motor windings. It can be retrofitted in most motors—even those of small horsepower ratings [10,11]. PTC units installed in each phase stator winding may be connected in series with two leads brought out of the motor. As the name suggests, the PTC has a positive temperature-to-resistance coefficient (i.e., the resistance increases as the temperature increases). A sensitive control monitoring relay is provided in the control circuitry of the motor, and this relay is energized while the motor is in operation and the winding temperature is lower than the critical value of the sensor. Once the motor winding temperature increases to a predetermined trip temperature, the sensor resistance increases several orders of magnitude for a correspondingly small change in temperature. This sharp increase in resistance causes the control module to de-energize an internal relay that in turn opens the contactor/starter coil circuit. Because the system places thermal sensors at the precise point where protection is needed (in the motor windings), equipment is protected against heat damage regardless of cause—including external faults such as blocked ventilation. A critical pump that stops due to the operation of any safety device (such as the over-temperature detectors or the overload relays of the motor starter) should *always* activate an alarm at the monitoring site—a fire or police station, a supervisor's home, or a central control station.

On large motors, it is more typical to use resistance temperature detectors (RTDs). The RTDs have a linear resistance/temperature response and are monitored by an analog instrument or directly by the PLC. The temperature of the winding being monitored can either be read directly at the monitor or through the PLC output display, and various alarm points are usually set in the readout unit or in a separate monitor. Motor starters need thermal overload relays in each leg, so three are required.

Vibration Monitors

To be effective, vibration monitoring systems are complex, and their high cost must be weighed against

the probable value of the information obtained. In high-speed machinery applications, the information obtained from sensors located radially on quadrature axes can be recorded and analyzed at intervals. Changes in these values, as well as changes in the axial movement of the shaft, may give an early warning of impending failure. In addition, the readout units provide an automatic alarm for out-of-limits radial or axial vibration.

In remote installations, particularly in unstaffed stations, there may be a good argument for a complete vibration monitoring and analysis system. The decision to install a vibration monitoring system should be based upon a careful evaluation of perceived risk if a bearing failure causes a catastrophic failure that could compromise the entire station or result in costly repairs. The ultimate user of such a system must be committed to the regular collection and interpretation of the readout data to make the additional investment worthwhile.

In unattended stations that have smaller pumping units than inferred in the above discussion, simpler vibration monitors are recommended to shut down the unit and to signal an alarm. If the vibration is due to a bearing failure, the bearing is already lost, but early shut-down may prevent further damage to the unit.

The purpose of monitoring systems is to ensure the safety of the pumping station, but a complex system that is inoperable because of poor maintenance or lack of regular testing is a hazard in itself because it creates a false sense of security. Pumping stations should be visited regularly (daily, if possible) so that vibration can be easily heard or felt.

Moisture Sensors for Submersible Motors

The most common detector is the capacitance probe (or two-point electrode) in a moisture leak sensor assembly filled with transmission oil. Water forms an emulsion that conducts electricity and completes a circuit containing a warning light or other alarm. A resistor can be added in parallel with the electrode to make it possible to check the circuit.

Moisture sensors are always placed between the inner and outer seals of submersible motors, and sometimes they are also placed at the bottom of the motor housing. The probes are effective in detecting moisture leakage through the shaft seals, although they tend to give false indications of moisture intrusion. Moisture detectors do not indicate leakage through the power cable or its connection to the motor housing, and occasionally motors are burned out by such leakage—a hazard with any submerged motor.

13-13. Specifying Pumping Unit Drivers

To obtain a driver completely coordinated with the pumping equipment, it is advisable for a single responsible manufacturer to provide the motor and the driven equipment. Usually the pump manufacturer will take this responsibility and purchase a motor to meet both the owner's specifications and the load requirements and starting frequency of the pump. If an adjustable-speed system is specified, the speed-control equipment should be included in the drive package. See Section 16-1 for more about this subject.

Motors for Water Pumping

Motors of about 10 to 60 kW (15 to 75 hp) are common in water pumping stations, but relatively large-size motors are not unusual. Even farm water pumping systems may be of several hundred horsepower if the irrigation system is extensive or the pumping head is large. The electric utility may advise or require large motors to be provided with a reduced-voltage starting system. Either an autotransformer starter or a part-winding start motor can be satisfactory, but solid-state starters should be considered first. Motors for well pumps usually drive a multistage pump often several hundred feet below the motor. The hollow-shaft motor discussed in Section 13-12, as well as the non-reverse ratchet feature, is usually required. A high-thrust (or very-high-thrust) top bearing on the motor is required due to the high loading involved in supporting the long shaft, the impellers, and the TDH of the water column.

Other features to be considered for the specifications include:

- Enclosure type
- Special cooling provisions
- Bearing type
- Bearing and winding temperature monitoring systems
- Special insulation system
- Special temperature rise limitations
- Stator winding temperature monitoring
- Voltage and frequency (for an adjustable-frequency drive, both the speed and frequency range and constant-volts-per-hertz ratio must be specified)
- Vibration monitoring, if applicable
- Brass nameplate
- Painting (manufacturer's standard, special, or prime coat only).

Motors for Wastewater Pumping

Motors for wastewater pumping range up to 600 kW (800 hp) or more and are likely to be indoors and even below grade where they may be subject to flooding. Immersible motors and pumps should be considered for these situations. Raw wastewater is usually pumped at heads of less than 30 m (100 ft) by single-stage pumps with open impellers. Several (at least two) units are installed to provide back-up and to handle the widely fluctuating flow range. In motor sizes below 75 kW (100 hp), direct across-the-line starting is almost universal. Reduced-voltage starting may be required in pumping stations located several miles or so from main electrical substations. Reduced-voltage starting may also be desirable to limit voltage dip. In adjustable-speed systems, raw wastewater pumps usually require only a moderate speed range, which simplifies the speed-control equipment.

Features that should be considered when writing the specifications are the same as those listed for water pumping, but with special emphasis on the ability of the unit to be returned to service quickly after flooding.

13-14. Definite Purpose Induction Motors

Motors for two definite purposes are described: (1) those for use with adjustable frequency (AF) converters, and (2) those subject to flooding or inundation.

Inverter-Duty Motors

Motors operating from ac AF converter power sources are called inverter-duty motors. Standard induction motors can run on an AF source, but without proper component selection and design precautions, only for a short service life. Standard induction motors are designed to run on a power source with a sinusoidal waveform. The output waveform of an AF converter is nonsinusoidal. Most modern ac AF converters use voltage-source, pulse-width modulated (PWM) inverters with very fast-switching power semi-conductor devices such as insulated gate bipolar transistors (IGBTs). With the increased popularity of these fast-switching, solid-state drives came the increased failure of standard design motors. Subsequent investigations revealed that the failures were caused by:

- High transient voltages caused by the high dV/dt (rate of voltage rise) of the AF converter.
- High terminal voltages caused by the reflected wave of the transient voltage along the motor feeder cable.
- Stray currents circulating through the motor bearings on their way to ground potential and caused by buildup of voltage induced in the rotor. Early bearing failure was the result.
- Additional heat generated in the stator and rotor caused by the nonsinusoidal waveform of the AF converter output. The temperature rise of the motor operating at a particular load on a source with a fundamental frequency plus harmonic frequencies (nonsinusoidal) is greater than for a motor operating under the same conditions on a source with only the fundamental frequency (sinusoidal) applied.

The industry's response to preventing or mitigating these failures was two-fold. The first approach was improved design using special materials and fabrication techniques. The second approach was the development of preventive measures (application guidelines).

Responding to the need for an improved motor design, NEMA revised the motor standard MG-1 in 1993 to include specifications for a definite purpose inverter-duty motor. NEMA also revised their guidelines for operating standard motors under adjustable-frequency applications.

Bearing failure mitigation has not been addressed in any great detail in NEMA MG-1 and has largely been left up to the individual motor manufacturers as to what method of prevention is employed. Repeated bearing failures can be an indication of electrical stress. Diagnosis of the problem is usually difficult because it is customary to operate motors with noisy bearings until bearing failure is so severe that any signs of shaft currents are destroyed. Indications include "fluting" or "picket-fence" marks on the bearing race. Initial damage is relatively minor and begins with the formation of small pits or craters in the bearing race. As damage progresses, a fluting pattern develops as the bearing balls run through the craters.

Common manufacturing practice is to insulate the nondrive end shaft bearing journal with a ceramic (aluminum oxide) coating. Insulated sleeve bearings are purchased with the outer diameter insulated by the bearing manufacturer. Insulated bearings are included as a standard feature of inverter-duty motors by most motor manufacturers. Other recommended measures for protecting bearings from stray currents include:

- Insulate both bearings
- Install ground brushes
- Insulate both shaft journals
- Install in-line filters between the motor and AF converter
- Improve grounding of the AFD system.

Two of the most popular design methods are insulated bearings and a shaft grounding system. The decision on which bearing protective system to use is often up to the end-user, as cost is usually the deciding factor.

Design

Basic electromagnetic design changes such as stator slot configuration, effective winding turns, improved insulation materials, and improved manufacturing techniques help to mitigate bearing failure. Improved magnetic wire with greater resistance to high voltage stresses postpones insulation breakdown, but does not prevent it.

Inverter-duty motors should be specified for all new adjustable-frequency applications. Recommended minimum requirements include:

- Speed range 10:1 on variable torque loads (for centrifugal pumps); 2:1 on constant torque loads (for positive-displacement pumps). On a centrifugal pump a speed range of 10:1 is more than adequate as maximum flow range is usually less than 3:1.
- Service factor 1.00 on a nonsinusoidal source.
- Definite purpose NEMA inverter-duty design (to meet NEMA MG-1, Part 31) is an ideal motor for AFD applications. Note that inverter-duty motors do not conform to Design A or B starting characteristics and therefore may not be capable of direct across-the-line starting, so a reduced-voltage starter may have to be used if bypass operation is required.
- Class F or H insulation with Class B temperature rise.
- Motor insulation shall meet NEMA MG-1, Part 3 for:
  - a voltage pulse withstand of $> 3.1 \times V_{\text{rated}}$ volts, rise time of $> 0.1\ \mu$s for motors rated $\leqslant 600$ V.
  - a voltage pulse withstand of $> 2.04 \times V_{\text{rated}}$ volts, rise time of $> 1\ \mu$s for motors rated < 600 V.
- Thermal protection sensors integral to the motor.
- Bearing protection features to prevent damage by high-frequency stray currents.

Operating Standard Motors with AF Inverters

Wherever possible, use inverter-duty motors for AF applications. When that is not practical, standard design motors may be used within their design limitations. Motors operating under AF applications are subjected to:

- Reduction in efficiency
- Derating of output horsepower
- Derating of thermal capacity due to reduction of cooling due, in turn, to reduction in operating speed.

Recommended minimum requirements for standard design motors under AF applications are:

- Speed range 4:1 on variable torque loads (for centrifugal pumps); 2:1 on constant torque loads (for positive-displacement pumps). A speed range of 4:1 for a centrifugal pump is more than adequate as the maximum flow range is usually less than 3:1.
- Service factor 1.15 on a sinusoidal source (derated to 1.0 for a nonsinusoidal source). For motors with 1.0 service factor, a further derating of 15% of the full-load output is recommended.
- NEMA Design B is a good choice of standard design motor for AFD applications. Design A, C, and E motors may also be used. Design A and E motors may not be suitable for bypass operation because of the high starting current. Design D motors are not encountered in centrifugal pumping applications and are not appropriate for AFD application.
- Class F or H insulation with Class B temperature rise. Old motors with Class B insulation need to be derated to provide the necessary thermal capacity. Rule of thumb is one horsepower size.
- Insulation with a voltage pulse withstand of > 1000 V with a rise time of $> 2\ \mu$s for motors with a V_{rated} of $\leqslant 600$ V, and $> 2.04 \times V_{\text{rated}}$ with a rise time of $> 1\ \mu$s for motors with a V_{rated} of < 600 V.
- Thermal protection sensors integral to the motor.

Alternative Preventive Measures

In situations where it is not feasible to employ motors that meet the withstand capability achieved with current standard or inverter-duty motors, some form of alternative solution is required. Examples where these alternative solutions may be required include old motors that do not meet the voltage withstand requirements or motors with undefined characteristics. In such situations, some form of motor terminal voltage modification technique is necessary to mitigate the voltage surges occurring at the motor terminals. The following techniques involve placing additional devices between the motor and the AF converter to limit the peak voltage level. These techniques are summarized as follows:

- Output reactors
- Line output *dV/dt* filters
  - Sinusoidal filters
  - Motor termination units.

Output Reactors. These reactors are specially designed units that can accommodate the PWM waveform without causing undue reactor heating and can also provide the necessary inductance values over the frequency spectrum needed. They are used to reduce the *dV/dt* and peak voltage. However, care is needed in their selection as reactors can theoretically extend the duration of overshoot and contribute to the problem. They should be mounted as close as possible to the AF converter output terminals. Normally the output reactor is mounted within the inverter cabinet of the AF converter in new installations. The disadvantages of this technique are increased cost, the need for extra space, and reduced efficiency (less than approximately 0.5%). Output reactors can also be used to compensate for cable charging currents (balances cable capacitance) and may be used for motor cable lengths up to many hundreds of feet on larger drives.

Line or Voltage Limiting Filter (dV/dt Filter). A filter design consisting of capacitors, inductors, and diodes or resistors may be used to limit the *dV/dt* drastically, thereby reducing both the amplitude and the rate of rise of the peak voltage. These filters allow the use of most motors and are therefore recommended if the data of a motor are unknown (e.g., for a retrofit), particularly on higher-voltage supplies (> 480V). Filters can be mounted within the inverter cabinet of the AF converter or in a separate cabinet. The disadvantages are the need for extra space (if in a separate cabinet), increased cost, and reduced efficiency. The increased losses of 0.5 to 1 .0% must be accommodated.

Sinusoidal Filter. A sinusoidal filter is a special design of low pass filter that allows the high-frequency currents to be shunted away. The disadvantages of these types of filters are that they are the most expensive and also they prevent the motor voltage from exceeding 90% of the supply voltage (thereby derating the AF converter). They do, however, have the following additional advantages:

- Reduced motor noise
- Reduced motor losses
- Simplified hazardous area motor certification
- Allows use of standard motors and long motor cables.

Motor Termination Unit. Some manufacturers produce series resistive/capacitive filters that may be locally connected at the motor terminals (usually as an extra box mounted near the motor). The fast-rising voltage pulse sees the capacitor as a short circuit and the resistive element is temporarily connected across the end of the cable. Disadvantages: If the motor termination unit is not sized correctly, over-voltages may occur, thereby contributing to the problem. These filters add losses of about 0.5 to 1.0%.

These devices have not been widely used. One concern is that the parallel connection could be compromised, subjecting the motor to the high transients without any warning. Another problem has been in matching the AF converter current rating to the motor rating and maintaining the same level of protection inherent with the converter. Termination units must not be used with motors designed for use in classified atmospheres. Some engineers apply these filters even when inverter-duty motors are used. Under certain conditions (e.g., long cable lengths) they may be required, in addition to the output reactor, to protect the motor.

Care must be exercised in applying these filters. There must be 100% compatibility between the AF converter (including output reactor), motor, and filter. It is important that the AFD system be investigated by an experienced engineer knowledgeable in motor and AF converter construction, the effect of the components on one another, and the mitigation of these effects. If this is not possible, a joint statement from the AF converter (including output reactor), motor, and filter manufacturers stating that their equipment is compatible with the specific installation should be obtained.

Cost Comparisons for Preventive Measures. In considering the relative merits of the competing solutions, costs should be considered. Table 13-3 is only used for rough relative comparison purposes. For actual costs, consult equipment suppliers.

Table 13-3. Cost Comparisons of Preventive Measures

Typical relative costs (480-V motor = 100%)					
Rating	Drive, %	Output reactor, %	*dV/dt* filter, %	Sinusoidal filter, %	Motor termination unit, %
3 hp	350	75	440	330	170
100 hp	220	15	100	150	10
300 hp	120	5	65	110	3

Other Considerations for AFD Installations:

- Power factor capacitors are *not* recommended on the load side of an AFD.
- There is reduced availability of inverter motors for use in classified areas. The motor manufacturer should be consulted for their use in hazardous classified areas.
- Keep the AF controller and motor close together to minimize the length of cable between them. Use high-quality insulation on cables. Avoid using THWN and other "thin" types of insulation. For best results use RHW with insulation rated 2000 V for 460-V motors, 1000 V for 230-V motors. Where long cable runs cannot be avoided, install an output reactor on the AF controller output or a motor termination unit (see precautions on their use). It is recommended that on immersible and other critical AFD applications, continuous corrugated aluminum-sheathed cables be used between the AF converter and the motor. Also recommended are cable connectors that provide 360-degree surface contact between the cable sheath and the connector and the connector and the motor frame to ensure a solid ground return path for stray currents.
- Adjust the AF controller carrier frequency to its lowest setting where noise is not a problem. Frequency setting must be compatible with any filters applied.
- Motor insulated bearings and/or a shaft grounding system are recommended on all inverter-duty motors. Consult the motor manufacturer for availability.
- Output reactors are recommended on all but the smallest (<10 hp) AF converters. They comprise the best prevention for the investment.

Immersible Motors

Immersible motors are definite-purpose, squirrel-cage induction motors designed to be used in dry well applications where there is a possibility of submergence (flooding).

Design

Immersible motors are designed to withstand up to 30 ft of submergence depth for a two-week period. Although NEMA MG-1 has no rating for an immersible motor, this design exceeds the requirement for immersible motors that is described in the IEC standard IP67 that requires that the motor be protected against effects of immersion to depths of between 0.15–1 m (0.49–3.28 ft) above the motor.

The basic design of the immersible motor is a premium efficiency, inverter-duty, TEFC motor. Under normal operating conditions the motor is cooled by a blower on top of the motor. When the motor is submerged, the blower is switched off by an external level switch and the motor is cooled by the surrounding water, thus allowing uninterrupted operation.

Bearings are either double-shielded or open-construction, deep-groove Conrad type. The motor is designed to prevent infiltration of water along the shaft by utilizing a tandem lip seal arrangement with pressurized grease. The completed motor assembly must successfully withstand salt spray tests for corrosion for 96 h.

The motor main conduit box is either of fabricated steel or cast-iron construction with a bolted and sealed cover. See Figure 13-11. Power leads enter into the conduit box through either a sealable water-tight gland or a potted hub that is bolted and gasketed to the conduit box to allow removal of the power cable. The conduit box and covers are designed to prevent moisture leakage.

To guard against occasional flooding, the motor termination, conduit, and cable systems are designed to prevent failure due to moisture ingress. Continuous corrugated aluminum-sheathed cables are recommended between the AF converter and the motor. Cable connectors that provide 360-degree surface contact between cable sheath and the connector and the connector and the motor frame are also recommended. Together with the potted hub, the installation is a water-resistant assembly that will remain in operation during temporary submergence, and it also provides a solid ground return path for stray currents.

Protection

Motors are equipped with space heaters appropriate for the frame size and thermal protection by thermostats connected in series. The motor is also equipped with humidity/moisture detectors internal to the motor and used in the control circuit to either warn of the problem or to disconnect the motor. If a separate conduit box is used for humidity/moisture detector leads, the leads must be sealed in a manner similar to that in the main conduit box.

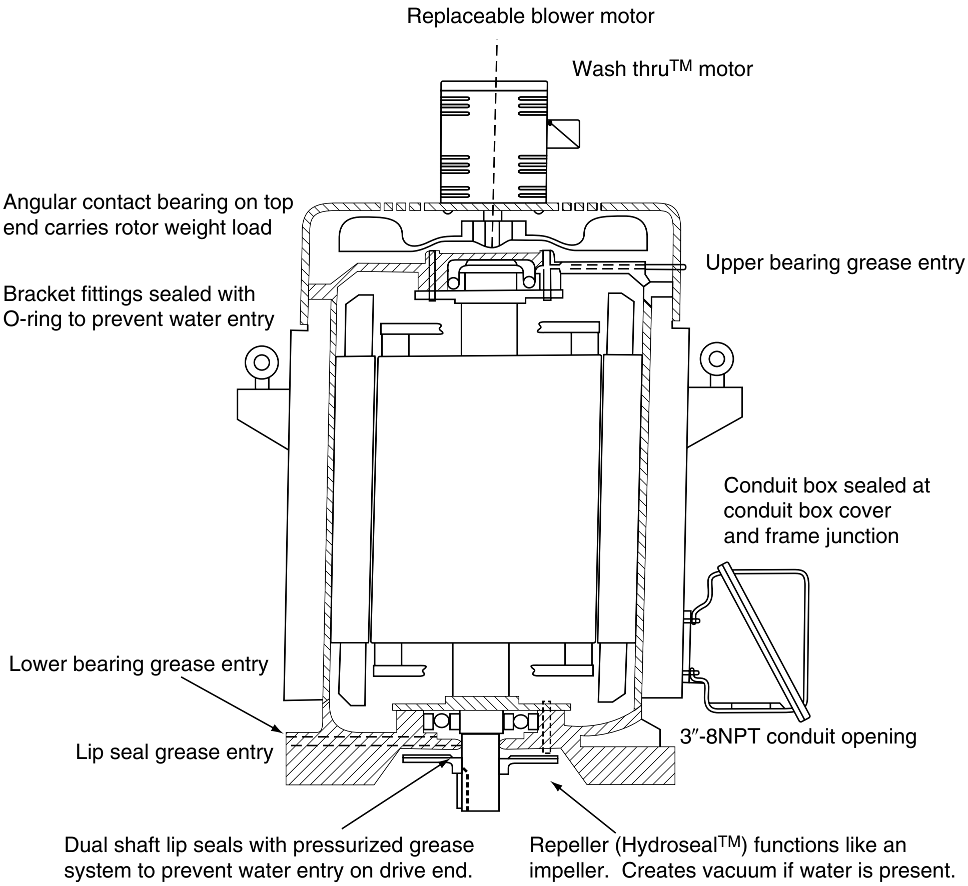


Figure 13-11. Immersible vertical pump motor. Courtesy of Cornell Pump Company.

13-15. Design Checklist

The following checklist contains aspects of the characteristics desirable in motors used for pumping service. The typical pumping station service is severe and warrants the expense of the more rugged motor designs. Such motors, manufactured in NEMA standard frames from size 286 through 445 (up to about 150 kW or 200 hp at 1800 rev/min), typically have cast-iron frames, whereas smaller motors with rolled steel or aluminum frames may be available from the same makers.

1. Motor frame (cast iron or, above 200 to 400 hp, fabricated steel)
2. Enclosure to suit application
3. Antifriction bearings (a Kingsbury bearing may be desirable on deep well pumps or large vertical pumps)
4. Copper stator conductors
5. Lifting eyes or welded-on hooks
6. All surfaces of the frame treated with corrosion-resistant epoxy paints
7. Nonsparking vent fan
8. Breather drain or (on small motors) two ¼-in. drain holes in totally enclosed motors, explosion-proof breather drains on explosion-proof units
9. Premium insulation (Class F or H materials)
10. Class B-rated temperature rise
11. All conductors brought out to the conduit box, lugged, and identified
12. Nonwicking insulation on motor leads
13. Large cast-iron or fabricated steel conduit box for motor leads
14. Grounding lug within conduit box
15. Separate cast-iron box for auxiliary circuits (temperature and vibration monitoring, etc.)

16. Special motor guarantees for severe applications:
    – Submersibles
    – Very-high-thrust vertical motors
    – Nonsinusoidal voltage supply (rectifier/inverter supply)
    – Unusual ambient temperatures or elevation
17. Submittal requirements for nameplate data
18. Submittal requirements for efficiency data at various loads and power factor at the same loads from "like-motor tests" or from factory testing of the first unit of the order
19. Space heater or a provision for winding heating of infrequently run motors and motors in damp locations
20. Over-temperature, vibration, and (for submersible motors) moisture monitors
21. Protection from dirt, rodents, and insects
22. Protection from moisture and flooding
23. Protection from vehicular traffic
24. Protection from sun and weather, if applicable
25. Balanced voltage supply at the rated value
26. Rated frequency supply (utility and local generation)
27. Protection from single-phasing for utility and in-plant systems
28. Rating of motor(s) and frequency of starts within the utility's system capabilities
29. Starting voltage drop reasonable
30. Adequate controls to limit the frequency of starts
31. Controls designed to prevent energizing a back-spinning motor
32. Prescheduled maintenance and frequency of inspections by adequately trained personnel written into the O&M manual
33. Inverter duty requirements

13-16. References

1. Linders, I., "Effects of power supply variations on ac motor characteristics," *IEEE Transactions on Industry Applications,* IA-8, 383–400 (Jul./Aug. 1972).
2. Woll, R. F., "Effect of unbalanced voltage on the operation of polyphase induction motors," *IEEE Transactions on Industry Applications,* IA-II, 38–42 (Jan./Feb. 1975).
3. Richardson, D. V., *Handbook of Rotating Electric Machinery,* Reston Publishers, Reston, VA (1980).
4. Richardson, D. V., *Rotating Electric Machinery & Transformer Technology,* 2nd ed., Reston Publishers, Reston, VA (1982).
5. Elshareif, A. "Cornell immersible pumps. A great alternative for conventional dry-pit submersible installations," Cornell Pump Company, Portland, Oregon (n.d.).
6. Karrasik, I. J., J. P. Messina, P. Cooper, and C. C. Heald, *Pump Handbook,* 3rd ed., McGraw-Hill, New York (2001).
7. Yuen, M. H., "Low voltage heating of motors in refineries and chemical plants," *IEEE Transactions on Industry and General Applications,* IGA-5, 300–309 (May/June 1969).
8. Dikinis, D. V., and M. H. Yuen, "Solid-state control—Low voltage heating of motors," *IEEE Transactions on Industry Applications,* IA-II, 287–290 (May/June 1975).
9. Lukitsch, W. J., "Methods of protecting against damaging effects of moisture buildup in motors used in the petroleum/chemical industry," IEEE Conference Paper PCI-80-15, Petroleum and Chemical Industry Conference, New York (September 1980)
10. Sheffer, K. W., et al., "Application of inherent thermal protection to industrial motor systems," *IEEE Transactions on Industry Applications,* IA-II, 14–23 (Jan./Feb. 1975).
11. Obenhaus, R. E., "Sensor retrofitting of motors for protection against overtemperature," *IEEE Transactions on Industry Applications,* IA-II, 24–32 (Jan./Feb. 1975).

Chapter 14

Engines

GARR M. JONES
JAMES W. SCHETTLER

CONTRIBUTORS
Robert A. Daffer, Jr.
Michael A. Devine
Stanley S. Hong
Philip A. Huff
Richard A. Malesich
Loran D. Novacek
Thomas O. Williams

Drives based on engines (either through electric generators or directly connected) should be considered for virtually every water, storm water, and wastewater pumping station installation. Pumping stations are required to protect public safety, health, and property, and operate reliably under all conditions of weather and manmade or natural disasters. See Section 9-9 for compelling reasons to install engines or engine-generators. This chapter is intended for those who may be unfamiliar with engine drives and their needs. It is an introduction, not a definitive work. Rely heavily on the advice of qualified manufacturers' representatives (especially in application engineering departments) and those with adequate experience in engine-based facility design, and consult them early in the design process.

References to a standard or code are given in abbreviated form, such as NFPA 37, as that designation is sufficient for identification.

14-1. Selecting an Engine Drive

Internal combustion engines are used only sparingly for pumping unit prime movers. To apply this type of equipment properly, a greater understanding of a wider variety of technical concerns is needed than for electric motor-driven installations. This section is intended only to outline the multitude of issues that must be addressed by the designer. To design the engine installation and specify the equipment properly, consult reputable engine manufacturers' engineering and sales representatives. Become completely familiar with the limitations, capabilities, and installation requirements for candidate equipment.

The following factors may lead to the selection of engine drives in lieu of electric motors.

- Greater reliability, if a reliable source of fuel is available and multiple engine-driven pumps are provided.
- Economy of operation (especially at wastewater treatment plants where the availability of biogas and the need for process heat almost always makes cogeneration attractive).
- Improved protection against surge damage due to (1) higher rotating moment of inertia, (2) greater reliability, and (3) a source of pumping energy that is independent from any electrical power supply. Variable-speed operation for capacity control and slow, surgeless starts.

- Remote location of installation and unreliable electrical supply.
- Fuel (natural gas, biogas, diesel oil) available at prices competitive with electricity.

Factors that may cause rejection of engine drives include:

- Concern over air pollutant emissions
- Concern over noise emissions
- Lack of adequate maintenance capability on the part of the owner
- Small (less than 60 hp) unit pump requirements.

To determine whether an engine drive is more economical than an electric motor, the best approach is to base the cost comparison on a unit of energy (such as kilowatt-hours or British thermal units). The costs must be modified to include the efficiency of the pump-driver combination. Do not forget to take into account obscure items such as transformation and motor efficiencies and slip, heat losses for motor drives, the cost of compressing fuel gas, gear losses, and power-absorbing accessories for engine drives. The costs for the various forms of energy depend on many factors, including location and, sometimes, time of day of consumption (particularly for electricity). These factors should be explored thoroughly, and then base prices should be determined. The following multipliers for U.S. customary units can be used to obtain the cost per British thermal unit:

- *Biogas:* free, except for the cost of treatment to remove contaminants that could foul the engine's compression, transportation, and standby fuel, if any
- *Crude oil:* dollars per barrel (19,000 Btu/lb, 7.5 lb/gal, 42 gal/bbl)
- *Diesel fuel:* dollars per gallon (19,000 Btu/lb, 7.5 lb/gal)
- *Electricity:* dollars per kilowatt-hour (3413 Btu/ $kW \cdot h$)
- *LPG:* dollars per gallon (2316 Btu/ft^3, 36.5 ft^3/gal)
- *Natural gas:* dollars per therm (1000 Btu/ft^3, 100 ft^3/therm).

Once the difference in energy costs has been calculated, it can be combined with adjustments in maintenance costs and a comparison can be made against capital cost difference to arrive at an initial assessment of economic viability. After an engine drive has been selected, a multitude of decisions is required. The primary categories for decisions include:

- Duty cycle
- Fuel
- Aspiration
- Combustion type (lean-burn or rich-burn)
- Type of engine.

Secondary categories include:

- Starting method
- Cooling method
- Controls
- Governors
- Accessories
- Combustion air
- Exhaust piping and silencing
- Pollution control
- Vibration isolation.

Finally, the following peripheral systems must be designed:

- Lubrication oil storage and supply system
- Fuel oil or fuel gas storage and supply system
- Service piping
- Building envelope
- Ventilation systems.

Most of the decisions listed in the primary and secondary categories dictate the equipment to be supplied as a part of the basic equipment contract. Engine drives are complex because of the number of components that must be coordinated properly. A unit responsibility provision, with a single manufacturer responsible for providing the driven equipment, the engine, all engine-related accessories, any gear reducers or increasers, shafting, and equipment supports is a recommended approach for such systems. (See Section 16-1 and Appendix C, Section 1.02B for unit responsibility.)

The third group of issues is designer-oriented and relates to systems to be designed by the pumping station engineer and to be specified and supplied independently. In addition, be aware of code requirements governing stationary engine installations, particularly NFPA 37 and local fire codes.

14-2. Duty Cycle

The term "duty cycle" means both the specifics of the application (duty), such as generation, direct drive, emergency, or standby, and the time-based utilization of the equipment.

Direct Drive

Direct-drive systems require the smallest engine to drive a given load because the engine does not have

to sustain the loads imposed by squirrel-cage induction motors' inrush currents. If an adjustment (gear reducer or increaser) is required between the engine and pump speeds, specify gears with a nonreversing mechanism to prevent reverse rotation on pump shutdown.

When using turbocharged engines, compare the torque requirement of the pump to the torque and power output of the engine, especially at startup when the engine exhaust stream is insufficient to drive the supercharger to full output.

Electric Generator Duty

Generator duty requires a careful analysis of the applied loads and generator response. Inrush currents require the generator to be sized to 120 to 150% or more of the nominal running load. The best approach is to apply loads in increments, thereby limiting the amount of the starting load to be sustained by the engine-generator at any one time. A careful analysis of load increments (type, size) is required to ensure that engine capability is not exceeded. A complete description of the conditions under which load will be applied to the standby generator must be furnished to the engine-generator (gen-set) manufacturer. The information to be furnished should include:

- Size and NEMA code letter for each pumping unit or other large motor to be started.
- Magnitude of miscellaneous loads (electrical resistance loads, lights, small motors).
- Special features such as variable frequency drives that could produce destabilizing effects in the system to be powered.
- Motor start sequence after the standby generator has started and reached stable operation.

Ask the engine manufacturer to consider this information and recommend specific equipment for the proposed application. The information thus received can be used for the *preliminary* allocation of space within the pumping station. Note that low-emissions features can compromise an engine-generator load acceptance stability. Be sure to be generous with space requirements because manufacturers may require added accessories or may change their minds in a bidding situation. If competitive bids are sought, include all of the information and the provision for a field test of the standby generator.

An analysis for the required size of generator and engine is given in Example 9-10.

Continuous Duty

Continuous-duty engines should be selected using conservative rating factors (Section 14-6). The principal objectives are reliability, low operating costs, and a long service life. Just as with electric motor-driven units, continuous-duty applications require multiple units to permit maintenance operations without a loss of capacity.

Standby Duty and Standby Generators

Standby engines used to drive pumps or generators require rapid starting and the ability to assume the load soon (within 10 s) after starting. They should include special starting aids, such as jacket water and lube oil heaters. Consult the engine manufacturer for recommendations. Engines for such service should have a relatively high torque versus speed relationship. Some lean-burn engines might not be suitable for standby generators, or might only be applicable if carefully loaded.

Standby generators are considered in Chapter 9, which states that standby engine-generator sets need to meet and sustain the high starting loads imposed by squirrel-cage induction motors. One effective technique is to size the generator for the worst-case condition of running loads and start-up, to provide the generator with a field-forcing regulator, to size the engine for the maximum steady-state operating loads, and to equip the engine with a large flywheel to assist in sustaining the starting loads. It is important not to oversize diesel engines for standby generators. Underloaded diesel engines develop carbon deposits in the exhaust gas passages. These deposits reduce the reliability of the engine. Reduced reliability is particularly a problem with standby generators because of the following:

- Standby generators are typically selected for starting loads under worst-case conditions—not running loads.
- Standby generators are exercised frequently, usually under less than peak load conditions.

In pumping stations, solutions may include the following:

- Recommended operating procedures that force the O&M staff to operate the station under peak loads. These might include: (1) shutting down a wastewater pumping station and storing the wastewater (if possible) until all pumps can be started and operated under power supplied only by the standby

generator for long enough to clear the carbon deposits from the engine; or (2) operating a water pumping station at a maximum rate under standby power.

- Make it easy to connect a portable electrical load bank at the motor control center or at the generator breaker.
- A fixed load bank with a suitable cooling system connected to the station electrical distribution system.
- Feed the power generated to the electric utility. The current generated must, however, be synchronized with the utility's current, requiring additional complexity in the station electrical equipment.

14-3. Fuel for Engines

Factors that govern the choice of fuel for an engine installation include:

- Engine exhaust emissions and compliance with required air quality
- Duty cycle
- Application of the equipment. (For example, diesel fuel may not be the best selection for a standby generator because of the duty cycle. Conversely, natural gas may not be available on a reliable basis.)
- Fuel storage and distribution
- Economics.

Fuels available for fixed-engine installations are gaseous (natural gas, liquid propane gas, biogas) and liquid (diesel, gasoline).

Combination fuel systems are available that allow some engines to function on more than one fuel, which improves economy or reliability. These include the following:

- Bifuel carburetion systems. These permit the utilization of more than one gaseous fuel. As a rule, these operate on an either/or basis; blending is not possible. Each fuel has its own pressure regulator and carburetor. Switchover to the standby fuel occurs on low pressure of the preferred fuel, and usually with momentarily reduced engine loads.
- Blended fuel systems. These are external to the engine and require mixing equipment to blend two gaseous fuels together to meet engine fuel demand. Typically, air is used to reduce the fuel value of a standby, higher-Btu-value fuel to that of a lower-Btu-value primary fuel. The advantage is that the blending system can be used to maximize utilization of the primary fuel when engine demands are greater than the fuel availability. Fuel blending systems can be costly, but once set up they have proven to be reliable and economical to operate. Blended fuel systems usually cannot be used on low-exhaust-emission, lean-burn applications.
- Dual fuel or gas/diesel engines. This category of engines is limited to relatively large [550 kW (750 hp) and larger], slow-speed (1200 rev/min or less) diesel engines. The engine is typically started as a compression ignition diesel. Once in operation, up to 99% (heating value) of the fuel is provided in the form of gas (natural gas or biogas), with the remainder provided as diesel fuel oil. Ignition is by compression rather than by spark.

Natural Gas

Commercially available natural gas, with a net higher heating value (HHV) of 4.45×10^4 J/L or $12.4\,\mathrm{kW \cdot h/m^3}$ ($1050\,\mathrm{Btu/ft^3}$), is a good engine fuel for the following reasons:

- Ignition takes place uniformly over time and avoids high momentary loads on engine parts.
- Natural gas has a nearly uniform fuel value of predictable quality.
- Gas is clean and requires minimal, if any, conditioning or fuel treatment.
- Natural gas is much more reliable than electricity (also, a limited fuel supply can be stored).
- Gas is usually available at a wide range of pressures.
- Engines are available from 20 kW (30 hp) to 3700 kW (5000 hp).
- Higher compression ratios for improved fuel economy may be used in certain situations.
- Clean-burning or lean-burn engine designs that meet air quality requirements and restrictions and that have substantially improved efficiencies are available.

Disadvantages include the following:

- Natural gas may be more expensive than alternative fuels, especially if a long transmission pipe is needed to reach the site.
- Natural gas may not be available under emergency conditions if it is not stored on site. Thoroughly explore its availability with the gas utility. Compressed natural gas is an option in some applications with smaller engines or with limited fuel storage needs.
- Gas may not be available with adequate pressure for some new-technology lean-burning engines,

and gas compressors with their accessories may be required.
- Regulatory agencies may require on-site fuel storage or may prohibit the use of gas for a standby engine fuel.
- Gas, in the right concentration in air, can create an explosive mixture.

Propane

Propane, available as a liquid under pressure (LPG), is useful as a reserve fuel for spark-ignited engines. The advantages of propane include the following:

- Propane has a relatively high fuel value of $9.32 \cdot 10^4$ J/L or 25.8 kW · h/m^3 (2500 Btu/ft^3).
- A large quantity of fuel can be stored in a relatively small volume [7.11 kW · h/L (91,900 Btu/gal)].
- Propane can be stored for long periods without deterioration.
- Propane has a low liquid specific gravity (0.51).
- Propane vaporizes at −43°C (−42°F) and is, therefore, useful in cold climates.

The disadvantages of propane include the following:

- Unless the rate of fuel consumption is relatively low, liquid must be vaporized with an external heat source prior to transport to the engine carburetor.
- The high gaseous specific gravity (1.52 relative to air) is hazardous at low points in structures, and it disperses slowly if leaks occur.
- Propane is heavily regulated by fire authorities and propane storage, and delivery facilities often have a negative public perception. Fire authority approval can be extremely difficult to obtain in heavily populated urban areas.
- The sharp peak in ignition characteristics can cause detonation and may increase maintenance. This tendency to knock or detonate limits the compression ratio to 8.0:1.
- Propane, if obtained from a poor-quality supplier, can be dirty or contaminated.
- Propane cannot be used in a compression-ignition engine.
- Propane burns with a higher temperature and thus produces more NO_x than does natural gas.
- LPG often contains small amounts of butane that can confuse sensitive air/fuel ratio controllers and possibly cause engine detonation.

Propane may be too expensive for continuous use, but given the advantages listed here, it can be a reserve fuel or fuel for a standby generator. In some engines, the timing may need to be changed.

Butane

Butane is somewhat similar to propane, but its combustion characteristics are erratic and it ignites extremely quickly with a very high flame temperature. It is not acceptable for most modern engines. Butane does not vaporize below −0.5°C (31°F)—a problem in cold climates.

Biogas

Biogas, a product of anaerobic decomposition of organic materials, may be readily available for use as an engine fuel at wastewater treatment plants and at sanitary landfills. As a rule, digester gas consists of 40 to 65% methane, 35 to 65% carbon dioxide, and small amounts of hydrogen sulfide, nitrogen, and halogenated hydrocarbons, and it is saturated with moisture. Landfill gas is similar but it contains about 30 to 55% methane, 45 to 65% carbon dioxide, nitrogen, and halogenated hydrocarbons, but no hydrogen sulfide. Both biogas fuels may also contain many other contaminants including siloxanes, arsenic, vanadium, and chromium. The moisture and hydrogen sulfide or halogens and, perhaps (depending upon the engine manufacturer's fuel and oil treatment requirements), the siloxane contaminants must be removed before it is introduced into the fuel system. The fuel heating value varies from about 38 to 62% of that for natural gas.

The advantages of using this gas for a fuel are:

- The combustible constituent (methane) makes it a good fuel for internal combustion engines. Methane is often described as an ideal engine fuel due to its knock or detonation resistance.
- The gas is a byproduct of wastewater treatment plants with anaerobic digesters and sanitary landfills; thus, it is free except for the low capital investment required for the collection and treatment systems.
- Carbon dioxide in the biogas reduces the combustion zone temperature so that biogas usually produces only 55 to 60% of the NO_x as does natural gas, the traditional clean fuel.
- With the proper selection of engines and engine cooling systems and for biogas with a modestly low H_2S concentration and an appropriate gas delivery system design, the only treatment required to make the gas suitable as a fuel is trapping and draining the condensate.

The disadvantages of biogas are:

- Only modest fuel pressures are available without resorting to complex and expensive gas pressurization systems.

- The low fuel value requires special carburetion systems.
- The saturated gas is warm as it leaves a landfill or a digester and can contain 2 to 4% water vapor. As it cools, the water vapor condenses and the condensate must be trapped and drained safely from all low points in the piping system.
- Contaminants such as water vapor, siloxanes, hydrogen sulfide, and particulates may cause corrosion and/or fouling or solids buildup on engine parts and accessory appurtenances, including turbochargers and exhaust heat recovery equipment.
- Exhaust catalytic converters required in some air districts (and a common compliance technique with rich-burn engines) are quickly fouled by the contaminants in biogas.
- Most landfill gases contain a wide spectrum of corrosive constituents. Gas conditioning and treatment along with engine maintenance can be quite costly.
- The gas flow from wastewater treatment plants fluctuates greatly (3:1 peak to average or greater). This fluctuation nearly always mandates an alternate source of fuel—usually natural gas.

Fuel supply systems must be designed and engines must be specified to accommodate the special characteristics of biogas. Some engineers provide gas scrubbing systems and fuel filters to protect biogas fuel engines against corrosion and contaminants. Some engine manufacturers have been able to apply naturally aspirated and some turbocharged engines to biogas fuels without conditioning the fuel to remove contaminants other than condensate. Turbocharged lean-burn engines with draw-through (low fuel inlet pressure) systems have a good record and are available from most engine manufacturers. Necessary precautions include high-temperature engine cooling systems and special bearings to guard against corrosion. However, virtually all engines (particularly lean-burn engines) require extensive and sometimes costly contaminant removal systems. Engine manufacturers should be consulted and required to furnish fuel quality specifications before considering them for candidates for projects where biogas will be the fuel.

Diesel Fuel

Diesel oil is an excellent choice for standby systems, but in 2005, it is too expensive for continuous operation. Its advantages are:

- Diesel engines are very reliable and stable. When properly equipped in standby generator applications, diesel engines can start up and accept their full generator load in less than 10 s.
- The high flash point and low volatility make diesel oil safer to store than most fuels.
- Higher compression ratios compared to gas engines permit greater horsepower per cylinder and thus smaller engines overall, along with improved efficiency.
- It is common and easily obtainable, even in most emergencies.

The disadvantages of diesel fuel include the following:

- Diesel engines usually produce more NOx, particulates (PM_{10}), and volatile organic compounds (VOCs) than gas-fueled, spark-ignition engines.
- Because diesel fuel oil deteriorates in storage at 6- to 12-month intervals, the tank might need to be drained and refilled. An alternative is to plan operating cycles and limit fuel tank size to ensure a full turnover of fuel inventory every six months.
- The fuel storage, handling, and supply systems are more complex than most. Engine-driven diesel fuel pumps supplied with the engines typically withdraw more fuel than the engine needs and thus the "excess" fuel must be piped or returned back to the diesel fuel day tank. It may be necessary to cool return fuel from some high-pressure diesel engines to prevent deterioration of the fuel.
- Repeatedly filling tanks and consuming diesel fuel causes moisture from the air to enter the fuel tank. This moisture condenses and allows bacteria to grow at the fuel/water interface in the fuel storage tank. These bacteria clog fuel filters, strainers, and fuel injectors if not removed or prevented from growing in the first place by proper system design and operation. (See ASTM D 975, Appendices X1 and X2, for some valuable information and recommendations on diesel fuel storage systems.)
- Both above-ground and below-ground fuel storage tanks include added spillage and protection risks that many owners find unacceptable.
- Engine and fuel system maintenance costs are high.
- Diesel engines are costly (but less so than gas engines).
- Exhaust emissions requirements may mandate the use of larger engine auxiliary cooling systems (for engine intercooler water), and exhaust particle/soot traps or even low-emissions diesel fuel.

Gasoline

Gasoline is often used for engines powering small- to medium-sized portable pumps, but the disadvantages for stationary gasoline engines are overwhelming.

- Gasoline is a fire and explosion hazard due to its low flash point and high vaporization rate.
- Gasoline engines generally are more temperamental than other types.
- Gasoline engines are relatively high-speed designs, and they are limited to approximately 375 kW (500 hp).
- The allowable storage period for gasoline is very short.

14-4. Aspiration

Engines may be either naturally aspirated or pressure-charged.

Pressure charging is typically accomplished by an exhaust gas-driven turbocharger or an engine accessory train-driven blower. For most engines, the combustion air is pressurized and the fuel is then added at the combustion chamber. This method of operation requires significant [140 to 420 kPa (20 to 60 lb/in.^{2})] pressure for the introduction of fuel into the combustion chamber. Diesel fuel is usually injected with an engine accessory train-driven pump. A variety of means, including fuel injection systems, are used for gas-fueled engines. A simpler system, available from some manufacturers, uses an arrangement wherein the fuel/air mixture is established at atmospheric pressure through a carburetion system, then compressed and delivered to the combustion chambers. This latter arrangement has proven particularly attractive for dirtier fuels and for control of exhaust emissions. Draw-through designs have also demonstrated much higher efficiency than other pressure-charged gas-engine designs. Turbocharged engines can more easily achieve exhaust emission requirements than do naturally aspirated engines, and virtually all low-emissions, lean-burn engines are turbocharged. The fuel gas passages and metering devices for naturally aspirated engines are large—an advantage when using biogas or other dirty gas fuels. Naturally aspirated engines may not be capable of meeting air pollution code restrictions in degraded air basins, even with currently available control technology.

Pressure charging is accomplished by an exhaust gas-driven turbocharger or by an engine accessory train-driven blower. With pressure charging, fuel gas pressures are high [from 140 to 550 kPa (20 to 80 lb/in.^{2})], engine efficiency is increased by 10% or more, and cooling requirements increase because more heat is wasted.

Pressure-charged designs that incorporate clean-burn or lean-burn technology are capable of complying with air pollution code restrictions in many locations without additional emission control technology.

14-5. Types of Engines

Engines useful in pumping stations can be categorized by ignition, cycle, and configuration, along with engine combustion type or air/fuel ratio (either rich-burn or lean-burn).

Ignition

Based on ignition, engines can be subdivided into those using spark ignition and those using compression ignition.

With spark ignition, the fuel charge is ignited (as in most automobile engines) by a spark arcing between two direct-current electrodes. Medium- or slow-speed spark-ignited engines should be specified with breakerless, electronic capacitor, discharge-type ignition systems. If available, dual spark plugs should be specified for larger-bore engines.

Compression ignition engines employ the heat generated by the compression of the fuel-air mixture to ignite the fuel charge. Compression ignition engines include both straight diesel and diesel-gas (dual-fuel) engines.

Cycle

Stationary engines are available in both two- and four-stroke cycle designs. Two-stroke engines require a separate scavenging blower (usually powered from the engine accessory train) to remove exhaust gases from the cylinder and introduce combustion air into the cylinder. In general, two-stroke cycle engines are limited to compression ignition designs and are less efficient, more complex, lighter in weight, less expensive, and noisier than four-stroke cycle engines.

Four-stroke cycle engines are far more common and are available as both spark-ignition and compression-ignition types.

Configuration

Stationary engines are available in either in-line or vee-block configurations. Compared with in-line-

engines on the same power basis, and same displacement, vee-configuration engines are generally more complex, more expensive to maintain, smoother operating, more compact, and require less floor space.

14-6. Rich-Burn or Lean-Burn

Spark-ignition engines can either be rich-burn or lean-burn engines depending on their air-to-fuel ratio.

For ideal or perfect stoichiometric combustion, a spark-ignition engine fueled with natural gas requires about 16 kg (35 lb) of air for every kg (2.2 lb) of fuel. At perfect stoichiometric combustion the mass air-to-fuel ratio is thus about 15.9 or 16 to 1 and the excess air ratio or lambda value is 1.0.

- Rich-burn engines are engines with a relatively rich fuel-to-air ratio and thus a lambda value of 1.0 or lower. Usually this lambda value is about 0.96 to 0.99.
- Lean-burn engines are engines with a lean fuel-to-air mixture and with a lambda value of about 1.06 or higher. Common lean-burn engine lambda values are in the range of 1.4 to 1.8. Lean-burn engines are almost always turbocharged.

Both rich-burn engines and lean-burn engines have a place in pumping station applications and each type has advantages.

- Rich-burn engines are simpler and do not have turbochargers.
- Rich-burn engines are more tolerant of small changes in fuel composition, ambient air temperature, and barometric pressure.
- If equipped with exhaust catalytic converters, rich-burn engines can, however, have precise air/fuel ratio sensors and controllers, and can meet very stringent air quality regulations.

By comparison, most lean-burn engines are of newer designs. In the last 20 or 25 years, nearly all new gaseous-fueled, spark-ignition engines have been lean-burn types.

- Lean-burn engines produce 90 to 95% less NO_x without post-combustion or "tailpipe" emissions control devices such as exhaust catalytic converters. Emissions from many lean-burn engines are within the allowable air quality limits in most locations.
- Lean-burn engines require 3 to 10% less fuel than rich-burn engines. This improved fuel economy is a valuable feature in a life-cycle cost comparison. The more complete combustion process translates into lower VOC (unburned hydrocarbon) emissions.
- Lean-burn engines produce more power from a smaller package and their compact size makes them good candidates for retrofit and capacity-upgrade projects.
- Lean-burn engines do require more combustion air, larger turbochargers and intercoolers, and larger-diameter exhaust stacks than do rich-burn engines, including turbocharged rich-burn engines.
- Many lean-burn engines, because of their higher power density, are not suitable for ebullient cooling systems, described later in this chapter.

14-7. Application Criteria

Application criteria (the rules by which a designer establishes how engine manufacturers may apply their products to a given power requirement) include the following:

- Excess air or lambda ratio
- Brake mean effective pressure
- Piston speed
- Rotational speed.

Recognize that not all engine manufacturers use the same criteria for furnishing the buyer with output power ratings for their products. Some manufacturers' ratings are very conservative, whereas others tend to overstress their products. Instead of accepting manufacturers' ratings, provide a set of criteria applicable to all manufacturers as a means of establishing a fair basis for bidding. The following recommended application criteria will help to produce a consistent approach to the selection of similar types of internal combustion engines.

Brake Mean Effective Pressure

Brake mean effective pressure (BMEP) is a widely used empirical measure of the load imposed on an engine. Although not universally accepted by all engine designers, it does provide a useful tool for controlling the size of the equipment to be used for a particular application. In SI units, BMEP for a four-cycle engine is given by:

$$\text{BMEP} = \frac{bkW}{LAn} = \frac{60\,bkW}{LAN} \tag{14-1a}$$

where the BMEP is in kilopascals, bkW is the brake kilowatts required at the output shaft, L is length of

piston stroke in meters, A is net piston area in square meters, n is number of power strokes per second, and N is number of power strokes per minute. In U.S. customary units,

$$\text{BMEP} = \frac{33{,}000\, bhp}{VN} \tag{14-1b}$$

where BMEP is in pounds per square inch, bhp is the brake horsepower required at the output shaft, V is displacement in liters, and N is number of power strokes per minute. Note that the coefficient 33,000 in Equation 14-1b reduces to 1.00 in Equation 14-1a.

When applied to two-cycle engines, Equation 14-1a reduces to:

$$\text{BMEP} = \frac{60{,}000\, bkW}{VN} \tag{14-2a}$$

where V is the displacement in liters and N is number of power strokes (or engine revolutions) per minute. In U.S. customary units, Equation 14-1b reduces to:

$$\text{BMEP} = \frac{396{,}000\, bhp}{VN} \tag{14-2b}$$

where V is the displacement in cubic inches. At one time BMEP was an effective parameter for comparing similar engines. Today, however, designs have changed significantly among different engine suppliers and within different engine lines from a supplier. Other factors such as fuel economy and exhaust emissions have become much more important. Using BMEP as a sole basis of engine selection is a blunder. A suggested set of BMEP limits for four-stroke cycle engines and various types of engine applications is given in Table 14-1.

Another approach advocated by some authorities is to limit BMEP to a percentage of the BMEP at the manufacturer's listed maximum power rating. The following are suggested limits to the engine's maximum power requirement:

- Continuous duty, relatively constant power demand—70 to 100%
- Continuous duty, variable power demand—75 to 100%
- Standby duty, incremental application of load—85 to 100%
- Standby duty, full-load application after starting—80 to 100%.

Recognize that BMEP impacts other factors, such as fuel economy and part load operation. As shown later, fuel cost is often about 80% of the overall cost of operation (per Table 14-8) [G3], and low BMEP can result in poor fuel economy

Derating for Altitude

Some engines, particularly turbocharged designs and lean-burn engine designs, may not be capable of achieving acceptable performance if the criteria in this section (14-7) are used. Note that the BMEP limitations should be adjusted downward for increasing altitude and increasing ambient temperature per the engine manufacturer's guidelines.

Table 14-1. Recommended BMEP for Four-Cycle Engines[a]

	kPa			lb/in.^{2}		
		Turbocharged			Turbocharged	
Duty	Naturally aspirated	Gas	Diesel	Naturally aspirated	Gas	Diesel
Standby	660	1100	1600	95	160	170
Continuous						
Constant load	550	790	900	80	115	130
Variable load	590	860	930	85	125	135

[a]Sea-level pressure and air temperature below 32°C (90°F).

The above rules are simplified. A better approach is to use all of the application criteria in this section.

Piston Speed

Piston speed, which is actually the *average* piston speed, is calculated as:

$$v = 2LN = \frac{LN}{30} \tag{14-3a}$$

where v is piston speed in meters per second and L, n, and N are as defined for Equation 14-la. In U.S. customary units,

$$v = \frac{LN}{6} \tag{14-3b}$$

where v is piston speed in feet per minute, L is stroke in inches, and N is number of power strokes per minute.

Piston speed is an indicator of (1) the rate of wear on cylinder walls and piston rings, (2) the magnitude of inertial forces at the top and bottom of the stroke, and (3) the class of engine design (i.e., "low," "medium," or "high" speed). Item 3 refers to rotating speed only indirectly. The important factors are the length of piston stroke and the given speed. The following limitations are suggested for average piston speed:

- Continuous duty—6.1 to 7.1 m/s (1200 to 1400 ft/min)
- Standby duty—6.6 to 8.1 m/s (1300 to 1600 ft/min).

Rotative Speed

Using rotative speed to compare competing engine designs can be somewhat misleading. Rotative speed does relate, however, to the frequency of piston reversals and, hence, to the rate of maximum stress occurrences.

The suggested limits for rotative speed are as follows:

- Continuous duty, 115 kW (150 hp) or less—20 Hz = 1200 rev/min
- Continuous duty, greater than 115 kW (150 hp)—15 Hz = 900 rev/min
- Standby duty, up to 115 kW (150 hp)—30 Hz = 1800 rev/min
- Standby duty, up to 375 kW (500 hp)—20 Hz = 1200 rev/min
- Standby duty, 375 kW (500 hp) or more—15 Hz = 900 rev/min.

Other Criteria

Other generalizations that may be useful for comparing competitive designs include the exhaust valve port area (a larger area requires less valve maintenance); the number of pistons (fewer pistons mean less complexity); and the experiences of other users, particularly in similar applications.

14-8. Starting Methods

Stationary engines may be started using one of the following stored-energy systems: compressed air, electricity (direct current from storage batteries), or hydraulic fluid.

When selecting a starting method, place substantial importance on storing sufficient reserve energy for a series of starting attempts without resorting to commercial power. A list of advantages and disadvantages of each starting system is given in Table 14-2. If either electric or hydraulic starting is chosen, the engine manufacturer usually supplies all of the starting equipment. The design of starting air compressor, receiver, and piping systems for compressed air starting is usually the responsibility of the pumping station designer. Suggested design criteria for the various starting methods are given in Table 14-3.

14-9. Cooling Methods

Because the cooling requirements of each engine design are unique, the equipment should be furnished by the engine manufacturer. Cooling methods suitable for use with stationary engines include radiators, heat exchangers, and ebullient (boiling water) cooling. Air cooling is available on some smaller engines, but it is not a viable system for the engines used in most pumping stations. Radiator cooling is different from air cooling. A radiator-cooled engine is water-cooled wherein the radiator removes heat from the cooling water.

The advantages and disadvantages of each method are given in Table 14-4 and the details of radiator, heat exchanger, and ebullient cooling are shown in Figures 14-1 through 14-3.

14-10. Controls

The engine manufacturer should be required to furnish all of the controls associated with the routine start-up and shut-down and emergency shut-down of

Table 14-2. Comparison of Engine Starting Methods

Method	Advantages	Disadvantages
Electric		
DC starting motor with rectifer batteries	Lowest cost. Simple system. Can be mounted on engine base.	Effectiveness and amount of stored energy falls with falling air temperature. Batteries require considerable care. Batteries subject to damage if rectifier fails or overcharges; use float-type charger. Limited effectiveness on larger engines.
Compressed air		
Air starter motor with remote air compressors and receivers	Most reliable. Can be used on all but largest engines. Consistent power available until stored air is exhausted.	Most costly. Requires greatest building space. More complex than electric. Requires greatest air-storage volume. Larger volume of air required per start.
Direct injection with remote air compressors and receivers	Very rapid start. Requires less storage volume. Smaller compressors.	Higher pressures required. Available on larger engines only. Most applicable to diesel engines.
Hydraulic		
Hydraulic motor with electric motor-driven pump, hand pump, and nitrogen-charged accumulators	Very rapid start. High-speed cranking. Requires least floor space. May be appropriate for portable generators.	System is complex and expensive. Leaks in hydraulic lines can be a problem. System is costly to maintain.

the equipment. If remote start–stop initiation is needed, require interfacing relays for field connection to manufacturer-furnished logic circuits. Other important features of the control system include the following:

- Use direct current (24, 48, or 125 V) for all engine-related controls. Do not use alternating current because external electric power supplies may fail, thereby compromising the reliability of the system.
- Do not locate the control panel on or near the engine foundation because the vibration causes control system maintenance problems.
- Engine controls must be coordinated with controls for the driven machine. For example, if it is an engine-driven pump with power-operated valves, the valve controls must be coordinated with engine start-up/shut-down controls to avoid sudden surges or reverse engine rotation.
- Specify dust-tight construction for the control enclosure to protect the equipment.
- The recommended control and monitoring system features are listed in Table 14-5.

14-11. Governors for Engine Control

Two basic types of governors may be employed for engine control: (1) isochronous (steady speed regardless of load), and (2) droop (speed varies slightly with engine load).

Isochronous governors are necessary where two or more units are operated in parallel and precise regulation and load sharing are required, such as when paralleling on an electrical bus. Droop mode is usually satisfactory for standby generators.

All governors use some type of detector (to monitor the engine speed) and a means of developing sufficient force to activate the throttle. Hydraulic governors are driven from the engine and use either a self-contained oil system or engine lubricating oil. Electric governors depend entirely on electronic/electric actuators. Pneumatic/hydraulic or electronic/hydraulic governors are the most suitable for variable-speed service. The recommended governor types for various applications are listed in Table 14-6.

Table 14-3. Suggested Design Criteria for Engine Starting Methods

Starting method	Components	Design criteria
Electric	Rectifier	Floating-charge and quick-recovery features with automatic voltage regulation.
	Motor	12 or 24 V DC with Bendix release.
	Batteries	12 or 24 V nickel-cadmium type; sufficient storage capacity for at least three starts at 30xF (lower, if appropriate for project).
Compressed air	Compressor	Two- or three-stage air-cooled (liquid-cooled required for higher pressures).
		Provide two. Size to recharge receivers in 1 h. Set start/stop controls to ensure receivers are fully charged. Allow no more than 10% loss of pressure before starting lead compressor alarm at 15% loss of pressure. Aftercooling is not necessary if discharge piping is insulated.
	Receivers	Provide at least two. Store air at 300 lb/in.^{2} (gauge) or greater to reduce size and provide better starting action. Provide sufficient storage for at least three successive starts—more if more than one engine. Size piping to engines to minimize pressure loss. Use copper piping to prevent corrosion.
	Starting motor (if required)	Provide pressure-reducing station at motor to adjust for motor limitations. Must be fitted with Bendix release. Provide lubricator and exhaust silencer. Solenoid valve located on receiver side of flexile hose connection to engine piping system.
Hydraulic	Pump (electric)	System pressure not more than 3000 lb/in.^{2} (gauge).
	Pump (engine)	Piston type with internal relief and check valves.
	Pump (hand)	Double-acting piston type with lever for hand operation.
	Accumulators	Nitrogen-precharged bladder type. Sufficient storage volume for at least four starting attempts. Not less than four accumulators. Independent means for draining each accumulator with pressure gauge on gas side. Filters required to protect seals and rings in motors, pumps, and accumulators.
	Starting motor	Multipiston type with Bendix release.

14-12. Accessories for Engines

The recommended accessories for engines include the following:

- An electric or compressed air, motor-driven, lubrication-oil priming pump mounted on the engine sub-base. The pre-lube pump should be timer controlled to operate hourly and prior to engine start. On larger engines, a pre-lube pump is often listed as a standard feature and is extremely important in reducing engine wear during start-up.
- A standby engine should be pre-lubed for exercising. Startups on electric power failure are infrequent, and the wear without pre-lube is negligible.
- Dual oil filters with quick shift-over valves for larger, continuous-duty engines.
- A lubrication oil level regulator (in all spark-ignition and some diesel engines) to automatically replace the engine lube oil consumed during operation.
- Crankcase explosion vent valves (on large diesel and dual-fuel engines).
- A thermostatically controlled jacket water heater (on automatically started engines, especially diesels and in colder climates).
- Engine-driven or motor-driven circulating water pumps for external cooling water systems (heat exchanger and ebullient cooling).
- Blow-by emission absorbers for crankcase vents (specify cartridge type).

Table 14-4. Engine Cooling Systems

System	Equipment	Operation	Advantages	Disadvantages
Radiator (Figure 14-1)	Engine jacket water circulating pump Thermostat Lube oil heat exchanger Radiator and fan (if located at engine, fan is usually engine driven. If radiator is remote from engine, fan is motor driven)	Water is circulated through engine, oil heat exchanger, and radiator by pump Thermostat maintains desired system water temperature Fan circulates air through radiator, thus cooling water	Least costly system Simple to operate and maintain All engine heat, including oil, can be accommodated	Fan requires considerable power System is not readily adaptable for reclamation of waste engine heat Radiator is vulnerable to leakage [typical pressure rating $10-15\,\mathrm{lb/in.^2}$ (gauge)] Fans make considerable noise Difficult to provide high-temperature cooling needed when biogas is used as fuel Thermostat failure can cause engine overheating
Heat exchanger (Figure 14-2)	Engine jacket water pump Thermostat Lube oil heat exchanger Waste heat exchanger	Water is circulated through engine, oil heat exchanger, and waste heat exchanger by jacket water pump Water from an external source is circulated through waste heat exchanger to cool engine jacket water	Relatively simple system Modest cost Possible to reclaim waste heat for useful purposes Water or wastewater pumped station can be source of cooling water (through heat exchanger) All engine heat can be accommodated	Two pumped water systems Energy consumption can be significant System is not particularly suited for reclamation of exhaust heat Thermostat failure can cause engine overheating
Ebullient cooling (Figure 14-3)	Engine jacket water pump may or may not be necessary Steam separator Exhaust boiler (if exhaust heat reclamation is desired) Source of softened make-up water Source of condensing water if reclamation of waste engine heat desired Source of cooling water (or radiator) required for engine oil	Water circulates through engine to steam separator (and exhaust heat boiler) thermally if no jacket water pump is required Jacket water flashes to steam. If no heat reclamation, make-up water is added to steam separator to replace lost water; if heat reclamation is desired, cooling water in steam condensing heat steam; cooled water exchanger condenses returns to engine; typical engine water inlet temperature is 225°F, outlet 228°F	Low temperature differential across engine reduces thermal stresses and wear Stable, simple, easy to maintain Ideal for maintaining high exhaust temperatures necessary for biogas fuels Particularly suited for reclamation of waste engine heat Quiet Power consumption nearly negligible; system is safe, impossible to overheat engine	Higher first cost Requires special engine construction (bearings, gaskets, and seals) Full advantage of system realized by utilizing thermal circulation Many engine designs are not suitable

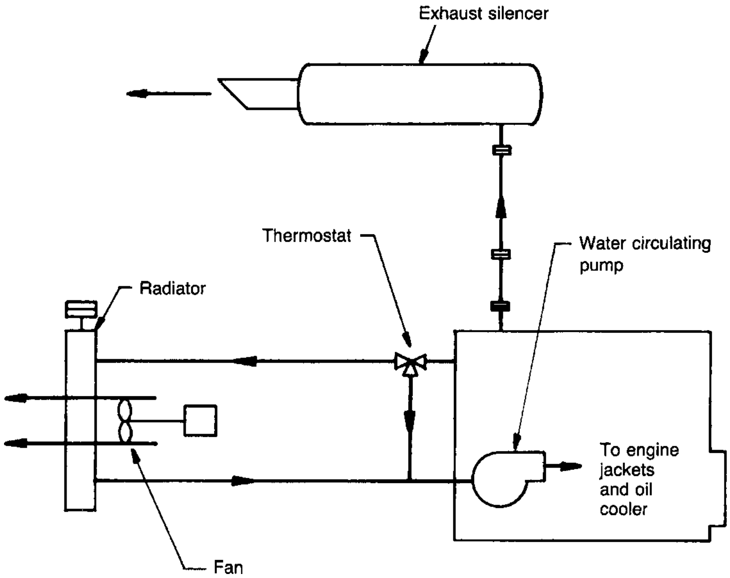


Figure 14-1. Radiator cooling. If the radiator is remote from the engine, the fan is motor-driven; if the radiator is at the engine, the fan is engine-driven. The thermostat controls the temperature of the water leaving the engine. The water circulating pump is engine-driven. After Brown and Caldwell.

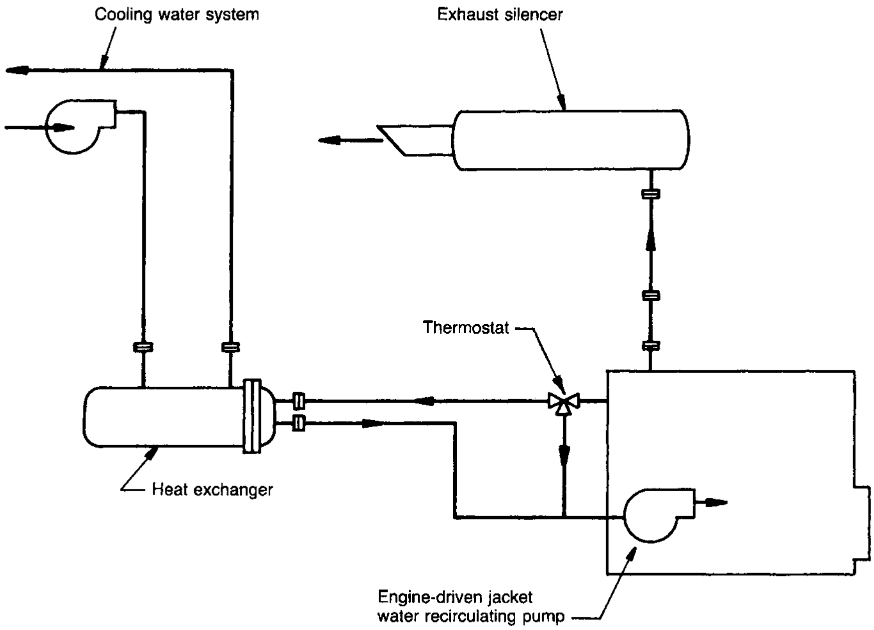


Figure 14-2. Heat-exchanger cooling. With the proper heat exchanger, the cooling water can be the flow from the pumping station and need not go to waste. The thermostat maintains the engine water outlet temperature. The exhaust silencer can be a heat exchanger or a boiler for heat reclamation. After Brown and Caldwell.

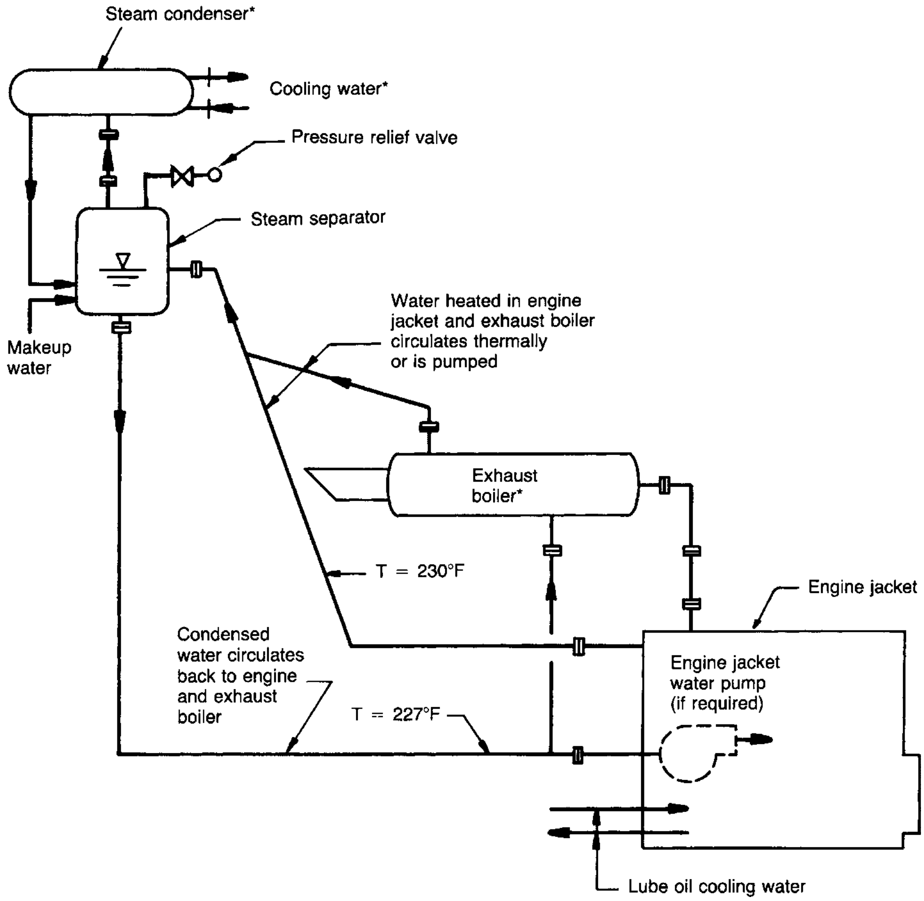


Figure 14-3. Ebullient cooling system. In the steam condenser, steam is condensed and returned to the separator (the conversion is 950 Btu/lb of steam). In the steam separator, heated water flashes to steam (the conversion releases 950 Btu/lb of water). The asterisks (*) indicate the components that are necessary to reclaim engine waste heat. If heat reclamation is not desired, the steam condenser, the steam separator, and the exhaust boiler can be eliminated, and the system will operate by boiling off water, but provide for blowdown of precipitated solids. After Brown and Caldwell.

- An electric motor-driven cool-down pump, particularly with larger engines. Recognize that engine-driven accessories, such as engine-driven centrifugal water pumps, may not develop adequate head when connected to engines operating at reduced speeds or in variable engine speed applications.

14-13. Combustion Air

Small- to medium-sized engines can be furnished with engine-mounted combustion air filters. If the engine room ventilation air is filtered, specify dry-type filter elements. Larger engines and some lean-burn engines require an air intake system ducted outside of the building to avoid upsetting the building ventilation system. If one is required, the pumping station designer should be responsible for such an air intake system. If turbocharged engines are installed, a separate silencer is often needed to control intake noise.

14-14. Exhaust Silencing

Exhaust silencers must be selected specifically for the application because of engine back-pressure limitations and should, therefore, be furnished by the engine manufacturer because allowable exhaust back-pressure levels vary in engines of different manufac-

Table 14-5. Recommended Engine Controls, Monitors, and Alarms

Controls	Monitors	Alarms
Start-up logic Cranking routine Actuation of associated remote devices Status Shut-down logic Actuation of engine-mounted and remote devices Status Emergency shut-down logic Actuation of engine-mounted and remote devices Status Manual start-up/shut-down Timer and logic for lube oil priming pump Timer and logic for remote devices such as cooling fans and pumps	Oil temperature and pressure Jacket water temperature Engine speed Intake manifold vacuum (gas engines) Cylinder exhaust temperature (larger, continuous-duty engines) Main bearing temperature (larger, continuous-duty engines) Vibration (larger engines)	Low oil pressure[a] Jacket water temperature (radiator and heat exchanger cooling only)[a] Low water level[a] High oil temperature Cylinder exhaust temperature (large, continuous-duty engines) Vibration (larger engines)[a] Overspeed[a]

[a]Initiates emergency shut-down.

turers and at different rotational speeds. Chambered, ported types appear to be the best. There are four different grades of effectiveness:

- *Industrial*—minimal sound attenuation
- *Commercial*—suitable for nonresidential areas
- *Residential*—applications where a high degree of silencing is required
- *Critical*—maximum silencing for locations near hospitals and very quiet residential areas.

Do not rely solely on the silencer manufacturer's rating for the equipment. Instead, request specific detailed noise data from the engine manufacturer and specify noise reduction by octave band. In critical applications, specify a result [i.e., the desired numerical noise level on the decibel-A scale (dbA) at a specific distance from the exhaust]. An acoustical engineer should be consulted in such instances (see also Chapter 22).

Because of their design, some exhaust heat boilers provide a high degree of silencing and approach residential quality, while others require an added supplemental exhaust silencer. Specify inspection ports and drain plugs for all silencers and for the gas side of all exhaust heat boilers. (Davited inspection doors are preferred for boilers.) Stainless steel (type 347) may be advisable for silencers serving engines that operate only intermittently. Engine exhaust piping should also contain 25-mm (1-in.) diameter exhaust sample ports located in an unobstructed, straight reach of engine exhaust piping. The sample ports allow testing of the exhaust gases for air quality compliance.

One relatively new technology is active noise reduction, or active noise cancellation. It is a technique

Table 14-6. Recommended Governor Types for Engines

Application	Governor type
Generation	
Standby generator (single unit)	Droop, hydraulic
Standby generator (two, in parallel)	Isochronous, with synchronizing feature electronic/hydraulic
Continuous-duty multiple units or cogeneration with utility	Isochronous, with synchronizing feature electronic/hydraulic
Direct drive	
Constant speed	Droop, hydraulic
Variable speed	Electronic/hydraulic or pneumatic/hydraulic

in which the actual equipment noise waveform is electronically sensed, analyzed, and duplicated 180 degrees out of phase from the noise itself, essentially canceling out the noise waves. This "negative noise" technology is common in special high-tech earphones used by helicopter and other aircraft pilots, and by some armored vehicle drivers. One new offshoot of this technology consists of "smart" exhaust silencers. These devices electronically sense large internal combustion engine exhaust noise and modulate a special damper within the exhaust muffler for maximum noise attenuation.

Fully silencing the combined effects of multiple large noisy engines requires expensive and sensitive equipment and careful analysis, but the technology is advancing and has been successful in a very few pumping stations. Designers might want to consider this technology in some unusual circumstances.

14-15. Pollution Control

Federal regulations and many local air pollution control codes require some form of engine exhaust emission control. Pollutants of interest include nitrogen oxides (NO_x), oxides of sulfur (SO_x), volatile organic compounds (VOCs), and particulates (PM_{10} or $PM_{2.5}$. Many emissions control technologies have been attempted—some with mixed results.

Generally speaking, the early versions of many exhaust converters using noble metal catalysts have not worked well even when using a clean-burning fuel such as natural gas. Turbocharged engines, using a very lean mixture in the combustion chamber, have achieved better (but not reliable) performance. The reasons are:

1. Many of the systems utilized devices such as oxygen sensors that were located in hostile environments including combustion chambers and exhaust gas streams, and they depended on sensitive instrumentation systems. Maintenance costs were high because of the labor required to keep the equipment operating and properly tuned for optimum emission performance.
2. Many of the original catalysts were structurally unsound and were easily damaged or were undersized.
3. The original catalysts were extremely vulnerable to contamination from trace constituents in the fuel, including trace elements from the fuel gas compressor and engine lubricating oil.

Engine emissions control techniques are of two general types:

1. Tailpipe systems, or post-combustion systems to reduce emissions downstream from the engine.
2. Combustion modifications to the engine itself to reduce emissions formation.

Emerging technologies using parameters easily measured outside the combustion chamber and exhaust gas stream, such as fuel charge temperature and pressure, engine torque, and so on, promise improved performance and reduced maintenance costs. Emission control technology used by various manufacturers of turbocharged engines include:

- Lean-burn
- Exhaust gas recirculation
- Variable speed supercharging.

14-16. Vibration Isolation

Depending on the application, the subsoil conditions, and the nature of the building housing the engines, the following alternatives can be used to isolate engine vibrations from the building (in descending order of isolation efficiency). See also Chapter 22.

- Separate foundation, with vibration-absorbing filler between the building and the engine foundations.
- Heavy inertia block, mounted on energy-absorbing material or spring-type isolators.
- Spring-type, energy-absorbing isolators mounted on a heavy engine sub-base (the sub-base may be filled with concrete to lower the center of gravity). Engine manufacturers must provide the proper sub-base for spring-type mounting arrangements. Seismic restraints may also be required.
- Cork and elastomer-type, energy-absorbing sandwich material under the engine sub-base.

Before choosing which vibration isolation system to use, carefully evaluate the natural frequency of the structure and the effect transmitted vibration may have on sensitive electrical and electronic equipment. Because unbalanced forces in engines vary widely among engine types and manufacturers, the vibration isolation system should be made a part of the engine manufacturer's responsibility. Specify the maximum transmissibility and the maximum amplitude for the engine during operation and on start-up.

14-17. Lubrication Oil Storage and Supply

Oil should not be stored in large volumes inside buildings because of fire codes and insurance costs.

The maximum allowable volume of stored lubricating oil depends on local regulations and authorities.

Automatic lubrication oil-makeup units are designed to sustain only modest pressures, so direct pumping is not possible. The solution is to use a small day tank similar to the 30-gal (110-L) day tank for fuel, as illustrated in Figure 14-4. The system operates on a fill-and-draw basis. Waste oil is pumped to a storage tank by means of a portable, air-operated pump. For large engines, use a fixed pump and permanently connected oil drain piping. Do not use galvanized (zinc-coated) piping or appurtenances in any portion of the lubrication oil storage or supply system, because the zinc can cause oil degradation. The building design should incorporate provisions to contain any spills caused by day tank overflow or rupture.

14-18. Fuel Oil Storage and Supply

Diesel fuel injection systems are not designed for pumped supply but must, instead, draw fuel by gravity from a day tank. The system shown in Figure 14-4 ensures the constant supply pressure required for proper injector operation. About 20 to 80% of the of the excess or return diesel fuel must be drained back to the fuel storage tank from the engine fuel oil pump and cylinder distribution system while the engine is in operation. Never use galvanized (zinc-coated) piping or appurtenances in any portion of the fuel oil storage or supply system. Avoid excessively oversizing the fuel oil piping to minimize air bubbles in the diesel fuel. Recall that modern diesel engines with high-pressure fuel injectors operate at pressures of up to 275 to 400 bar (4000 to 6000 lb/in.^{2}), so the return fuel can be hot enough to require fuel coolers. These high-pressure fuel injectors require high-quality, well-filtered diesel fuel.

14-19. Gaseous Fuel Storage and Supply

The design of storage and supply systems for gaseous fuels is governed by NFPA 37 and NFPA

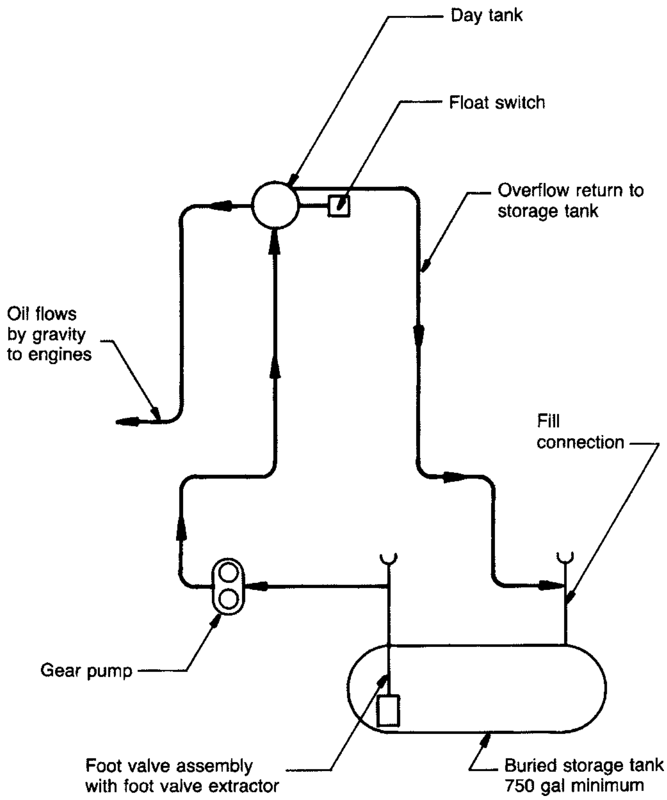


Figure 14-4. Schematic for a lubricating oil supply system. The day tank is less than 30 gal and is located high in the superstructure. The float switch controls the pump on a fill-and-draw basis. After Brown and Caldwell.

58. In general, the local utility's storage and transmission system provides adequate service for natural gas and no separate storage system should be necessary. If a pressure-charged engine is allowed, however, a separate pressure-boosting system may be required.

Be particularly careful when designing gaseous fuel systems because the effectiveness of carburetor systems depends on stable fuel supply pressures. If possible, design biogas systems that do not use compression and storage systems because:

- The power required for compression and the initial cost of compressors and storage vessels reduces the economic advantages of biogas.
- Gas compressor oil can cause problems with catalytic converters and gas purification equipment.
- Biogas is contaminated with liquids, sulfur dioxide, siloxanes, and particulate matter, and the condensate causes corrosion and other maintenance problems in compression equipment.

The gaseous fuel supply to naturally aspirated engines may be troublesome because of the relief valve required in Paragraph 4-21 of NFPA 37. The relief valve is extremely sensitive, may leak, and may lose fuel, especially in low-pressure fuel systems. Be aware of this particular provision and be prepared to deal with it. One obvious solution is to employ a fuel transport pressure of less than 3.5 kPa (0.5 lb/in.^{2}) gauge within the engine room.

Note that NFPA 37 also states that engine fuel piping, by itself, is not adequate cause to designate the engine room as a Classified Area.

14-20. Service Piping

The service piping is the piping and ductwork that connects the engine to various auxiliary system piping networks. Examples of service piping include exhaust, fuel, lubrication oil, starting air, and cooling water. Give the design your full attention.

The engine is a source of vibration, so flexible piping sections (or flexible metal hose) should be installed at each engine connection. The piping on both sides of the flexible section should be firmly anchored to prevent the transmission of motion that could cause the joints or flexible hoses to fail. Also observe the following:

1. Install the engine exhaust flexible connection as close as possible to the engine exhaust outlet.
2. Avoid fittings or lengths of pipe between the engine exhaust connection and the flexible hose; the weight could damage the exhaust manifold.
3. Try to make flexible piping connections such as engine lube oil, fuel, and compressed air (for air motor-started engines) as close as possible to the engine crankshaft.
4. Install flexible sections parallel to the engine crankshaft to increase their service life.
5. Require flexible piping sections to be designed for a specific number of full displacement cycles (10,000,000 is suggested).
6. Install isolating valves and automatically controlled valves, such as fuel and air valves, in the rigid gas piping upstream from any pressurized flexible section to avoid the escape of fuel to the engine room if the flexible section fails.
7. Use braided stainless steel hose rated for a working pressure at least twice as great as the system's maximum pressure.
8. Install flexible duct connectors on the air ductwork between engine-mounted radiators and the built-in wall exhaust louvers.

14-21. Building Envelope

When designing structures for engine installations, consider these caveats:

- Avoid the transmission of engine vibration by conducting a thorough analysis of each building element and eliminating harmonic natural frequencies.
- Build a small room for engine maintenance tools adjacent to the engine installation.
- If permitted by the project budget, conduct a sound spectrum analysis and provide noise-control features. In critical applications where building noise emissions may be objectionable, employ an acoustical consultant and coordinate the acoustical control efforts with the ventilation system and architectural design.
- Avoid crowding engines. A clear passage of 2 to 2.4 m (6 to 8 ft) between adjacent units is suggested.
- Be aware of special engine maintenance clearance requirements. One example is the vertical distance above the heads that is necessary to pull the pistons on some engine designs.
- Install some type of hoisting means (e.g., a bridge crane or a rolling gantry with chain hoist) to aid in engine maintenance.
- Design for explosion venting (see NFPA 37).
- Except for gas compressors, explosion-proof electrical devices for engine rooms are not required by

any code. However, engine rooms should be designed for explosion venting when gas engines are used (see NFPA 37 and NFPA 68).
- Provide a means for draining the crankcase. A portable, air-operated pump to remove the oil from the engine sump is preferred. If supplied from a fixed-service air system, the air can also be used for other purposes (e.g., torque wrenches and other air-driven tools).

14-22. Ventilation

Ventilation systems for engine rooms should be designed for the following:

- Balanced room pressure.
- Powered intake and exhaust for continuous-duty applications.
- A maximum room temperature of 43°C (110°F).
- Positive air flow at a velocity of about 8 km/h (5 mi/h) in a path at least 2 m (6 ft) in diameter along each side of each engine (necessary for personnel comfort).
- Air flow from the driven equipment (pump, gear, generator) and then directed toward (and possibly over) the engine.
- Exhausted 10% of the total air flow through the roof to clear out smoke and fumes; the rest should be exhausted at the wall at the rear of the engine.
- Adequate engine combustion air if engine-mounted air intake systems are provided—a *variable* that depends on the engine load and the number of engines in operation.
- Filtered air for continuous-duty applications, tempered to 13°C (55°F) in the winter; cooling is necessary only in hot climates.

Unless mitigated, ventilation air openings (for both supply and exhaust air) have a clear pathway for transmitting engine body noise to the surrounding neighborhood. Suggestions for minimizing engine noise are:

- Use sound attenuation louvers and sound traps on intake and exhaust ventilation air systems. Change the ventilation air duct direction to avoid a clear unobstructed noise transmission pathway.
- Consider an air plenum between engine-mounted radiators and wall exhaust louvers. The plenum can be both a directional change and a location for installing sound traps.
- Consider roof supply fans or upward-directed exhaust fans to reduce the size or number of wall louvers.
- Seal off all other outdoor wall penetrations.

14-23. Maintenance

The manufacturer's instructions for maintenance should, of course, be followed exactly, but a designer contemplating an engine drive should recognize the owner's involvement. The maintenance of engines depends on whether (1) the engine is a prime mover and operates regularly, or (2) the engine is on standby and operates only about 50 h per year (including exercising).

Prime Movers

The life-cycle costs for a continuous-duty diesel engine are compared with those for a gas engine in Tables 14-7 and 14-8. The total costs are based on an operating period of 20 years and 1997 prices.

Standby Engines

Standby engines operate so seldom that maintenance depends on time as well as wear. Maintenance tasks for a 125-kW diesel engine-generator set are shown in Table 14-9. On a yearly contract, the total cost of maintenance labor and parts would be about $700/yr if an allowance of 2 h driving time is assumed per yearly visit.

There is a considerable difference between the maintenance of light, standby units and continuous-duty prime movers. For example, the service interval for valve overhaul in a heavy-duty engine is 20 times as long as the interval given for the light-duty engine in Table 14-9.

Exercising Engines

Instructions vary, but the following recommendations are good practice for modern engines.

Continuous-Duty Engines

At least once every six months, all engines should be operated at the rated load for at least 5 to 10 min after reaching operating temperatures to ensure that the engine is capable of developing the required rated power.

Standby Service

Every six months to one year, all engines should be full-load tested as follows: Operate at 1 or 2 min at no

Table 14-7. Data for Diesel and Natural Gas Engines[a, b]

Item	Diesel engine	Gas engine
Size of engine, hp	1150	1150
Engine output, kW	820	820
Engine speed, rev/min	1800	1200
Total engine prices, $	158,000.00	268,000.00
Down payment, $	58,000.00	68,000.00
Loan interest rate, %	10.00	10.00
Inflation rate, %	4.00	4.00
Months payment due	120	120
Taxes, $	9,875.00	16,570.00
Insurance, $	0.00	0.00
Operation, h/yr	8000	8000
Fuel	No. 2 diesel	Dry natural gas
Fuel price	1.00 $/gal	0.35 ft^3
Fuel consumption at average load	59.06 gal/h	9500 ft^3/h
Oil	10W30	SAE 30
Oil consumption at average load	0.14 gal/h	0.05 gal/h
Oil change interval	500 h	1000 h
Sump size, gal	81.1	111.7
Overhaul interval		
Overhaul 1 (top end), hours	6530	16,006
Overhaul 2 (major), hours	13,061	48,017
Overhaul 3, hours	—	80,002

[a]Courtesy of Caterpillar, Inc.

[b]Prices effective June 1997. ENRCCI = 5700.

load, 5 min at 25% load, 5 min at 50% load, 5 min at 75% load, and 5 min at rated load.

Engines should be no-load exercised at least once each month. The engine should be run just long enough to ascertain that the starting and control systems are operating properly and that proper oil pressure is obtained. Excessive no-load operation causes the exhaust system to "slobber," a problem that is mostly cosmetic (messy). Slobber, an oily discharge from the manifold stemming from partially burned fuel, is especially noticeable in modern direct-injected diesel engines. The exercising can be done by an automatic timing device installed in the transfer switch. These devices are usually 1-week cycle timers. There are 2-week cycle timers but they may not be available for most switches. Many engineers dislike automatic exercisers because (1) an operator should be listening to an engine while it is exercised; (2) the maintenance crew may depend too much on automation and neglect the trouble-forestalling inspection of the running engine; and (3) very few automated exercising systems can adequately exercise the engine under load. The owner should be consulted about automatic versus manual control of exercising.

Manufacturers' Recommendations

Although the above exercising instructions comprise one manufacturer's recommendation for modern engines, there are many other makes and models and many older engines. Owners should be advised to follow the maker's recommendation for the par-

Table 14-8. Financial Summary for Twenty Years of Operations[a,b]

	Diesel cost			Natural gas cost		
Item	$	$/h	¢/kW·h	$	$/h	¢/kW·h
Selling price	158,000	0.99	0.12	268,000	1.68	0.20
Interest	58,581	0.37	0.04	117,162	0.73	0.09
Taxes	9,875	0.06	0.01	16,570	0.10	0.01
Insurance (omitted)	—	—	—	—	—	—
Fuel	9,450,000	59.06	7.20	5,320,000	33.25	4.05
Oil	288,500	1.80	0.22	151,400	0.95	0.12
Preventive maintenance	241,500	1.51	0.18	153,900	0.96	0.12
Components	567,500	0.36	0.04	107,600	0.67	0.08
Overhauls	937,900	5.86	0.72	295,800	1.85	0.23
Total cost	11,202,000	70.01	8.54	6,430,000	40.19	4.90
With inflation	16,584,000	103.65	12.64	9,393,000	58.71	7.16

[a]Courtesy of Caterpillar, Inc.

[b]Prices effective June 1997. ENRCCI = 5700.

Table 14-9. Typical Maintenance Schedule for a 125-kW Standby Diesel Engine-Generator[a]

	Frequency interval	
Maintenance task	Operating time	Calendar time
Inspect fuel, oil level, coolant	8 h	1 mo[b]
Inspect air cleaner, battery	50 h	1 yr
Clean governor linkage, breather, air cleaner	100 h	1 yr
Clean fuel filter, replace oil filter, change crankcase oil, check switchgear	200 h	1 yr
Clean commutator, collector rings, relays, cooling system; inspect brushes, valve clearances, starting and stopping systems, water pump	500 h	1 yr
Check injectors, grind valves (if required), remove carbon, clean oil passages, replace secondary fuel filter, grease bearings, clean generator	1000 h	—

[a]Courtesy Onan Corporation.

ticular engine model and the particular circumstances of service. Some regulatory agencies impose operating limitations for exercising standby diesel engines.

Batteries

Starting batteries must be reliable and capable of five 10-s start attempts without a loss of cranking power. Maintenance-free batteries are recommended with either a two-stage trickle charger or, better, a float charger that is used to sense the charge on the battery and to limit the charging current accordingly.

Heaters

Moisture is the enemy of electrical equipment, including generators. Install space heaters inside the generators to minimize condensation and to keep them as dry as possible. Install electric heaters on the engine cooling water and lubricating oil to keep the engine warm. Use a diesel fuel that is compatible with the ambient temperature at the storage location.

Chapter 15

Variable-Speed Pumping

MAYO GOTTLIEBSON
ROBERT L. SANKS
ALAN VAUSE

CONTRIBUTORS
Stefan M. Abelin
Robert S. Benfell
Harrison C. Bicknell
Bruce Densmore
Ronald W. Duncan
Stanley S. Hong
Casey Jones
Paul C. Leach
Timothy Nance
Michael R. Olson
E. O. Potthoff
Douglas L. Schneider
Arnold R. Sdano
Richard N. Skeehan
Earle C. Smith
Patricia A. Trager

In this chapter, the uses of variable-speed (V/S) pump operation in water and wastewater pumping stations are presented and discussed. The principal adjustable- and variable-speed drives are reviewed and compared.

The primary goal of V/S *wastewater pumping* is to keep the station discharge rate equal to the influent rate at all times. The station acts as though it were a sewer discharging by gravity. A pumping station that meets this goal requires very little storage in the wet well, does not rapidly cycle any of the pumps, and does not cause abrupt, large changes in the discharge rate.

The primary goal of V/S *booster pumping* is to maintain a nearly constant pressure at all times, regardless of the flow rate. The discussion given here for V/S booster pumping also applies to high-service (high-head) pumping for reducing water hammer in very long transmission mains. Otherwise, V/S is rarely used for water pumping.

In common usage, the term "variable speed" indicates that the speed of the pump is regulated to produce the desired flow rate. The term "variable" infers a change that may or may not be under the control of the user. "Adjustable" infers a change under the control of the user. An example of an adjustable-speed drive is an adjustable-frequency drive (AFD), sometimes mistakenly called a "variable-frequency" drive. An example of a variable-speed drive is a rheostat-controlled wound-rotor motor in which, although speed varies with both load and control, the proper speed can be attained by the external control.

"Frequency" should only be applied to drives with an ac output, while the term "speed" is preferred since it applies to both ac and dc drives as well as to mechanical drive systems. Thus, pump speed can be varied at will by either an adjustable- or a variable-speed drive. In this chapter, both kinds of drives are called "variable-speed" or "V/S" for convenience, but when communicating with electrical engineers, confusion can be avoided by strict usage of correct terminology.

Applications to both water and wastewater pumping are described in Sections 15-1, 15-3, and 15-11. Sections 15-2 and 15-4 through 15-6 are generally devoted to wastewater pumping, and Sections 15-7 through 15-10 are confined to water booster pumping.

15-1. Variable Speed versus Constant Speed

The difference between V/S and constant-speed (C/S) pumping is profound and has far-reaching consequences, so the choice should not be lightly made. Some reasons for the choice of V/S or C/S for a particular pumping station may be compelling while others, which may be significant for a different situation, may be trivial. Nevertheless, all of the reasons should be considered thoughtfully.

Advantages of Variable-Speed Pumping

The major advantages of V/S pumping, stemming from the objective that discharge always equals influent flow rate, are:

- Downstream flow rates fluctuate gradually and do not upset treatment processes caused by surges, nor do the sudden surges caused by C/S pumping pound downstream piping.
- No storage is required, so the wet well can be smaller, shallower, and less expensive than for C/S pumping, and size might compensate for the high cost of the V/S equipment.
- Fewer—although larger—pumps are needed in medium- to large-sized stations with savings in space for the pump room and less equipment to maintain.
- The almost constant wet well surface and short residence time tend to reduce the production of odors and corrosive gases.
- Energy savings are likely due to the higher average wet well surface elevation and the usually lower pumping rates and lower pipe friction losses.
- Motor life is prolonged because of the infrequent starts.
- The high inrush current that characterizes across-the-line starts of C/S drivers is greatly reduced in V/S drives such as adjustable-frequency drives (AFDs) and slip-recovery systems (although not in eddy-current couplings). These reduced inrush current starts are referred to as "soft" starts.
- Voltage dips caused by across-the-line starts for C/S pumps can be substantially reduced with soft start equipment such as reduced-voltage starters. Solid-state, reduced-voltage starters are the most flexible, economical, and space-efficient. They are usually switched out of the circuit when operating speed is attained so that harmonic disturbances no longer occur.
- Throttling the engine of a direct engine drive is a simple, inexpensive, and reliable means of achieving V/S, but read Sections 14-2 and 14-22 and consult the engine manufacturers regarding the problems of operating diesel or gas engines at less than full load.

Wastewater

The reasons for using V/S pumping for wastewater include the following:

- Continuous pumping and the very short liquid residence time in V/S operation reduces (1) the *deposition* of organic solids—a problem in C/S pumping because of the difficulty of re-suspension; (2) *putrefaction*—so the wastewater is easier and less costly to treat; (3) the *production of odors* and corrosive, poisonous gases; and (4) *cyclic rise and fall of the water surface* with the resulting pumping of sewer gases into the atmosphere.
- V/S pumping eliminates the long free fall of wastewater and the entrainment of air bubbles that, if sucked into the pump, greatly reduce pump efficiency. However, this advantage disappears in a comparison with C/S pumping if a sloping approach pipe is used (see Section 12-7).
- The average static head in C/S pumping can be reduced by 0.6 m (2 ft) or more in V/S pumping by keeping the wet well level the same as the normal level in the influent sewer (see Section 12-6 and Figure 12-22). Turbulence is thereby virtually eliminated and the release of odorous gases is substantially reduced. The energy cost savings due to the consistently high wet well level may be significant.
- One important reason for using V/S pumping is to produce gradual changes in flow that do not upset (and thereby reduce the efficiency of) downstream processes such as sedimentation. Sudden flow changes caused by C/S pumps, even at significant

distances from a treatment plant, can upset the entire treatment process. At one utility, it was found that such flow changes propagated through the entire primary/secondary treatment process and were upsetting the dechlorination system. Forcing the operators to use the V/S drives (which had been installed but, due to indifference, were not used) solved the problem.

Water

Water is customarily pumped for long periods (of several hours) at uniform rates (often to fill reservoirs or tanks), so—although they may be just as compelling—there are fewer reasons for choosing V/S for pumping water than for pumping wastewater.

- For booster pumping or pumping into a distribution system without a reservoir, V/S can provide constant water pressure during varying rates of usage. That is the most common application.
- Pumps can be ramped up (or down) slowly so that water hammer is reduced. (Solid-state starters controlling C/S pumps can also perform this function if the pipeline is not too long.) In pipelines so long that $5t_o$ is more than 5 min (about 30 km or 20 mi), surges caused by the startup of C/S pumps can be serious. Solid-state starters or automatically operated control valves can also reduce water hammer, but only in shorter pipelines. The full-speed operation of C/S pumps for long periods (more than several minutes) at near shut-off head causes excessive wear on pump bearings and flexing of the shaft.

Disadvantages of Variable-Speed Pumping

The selection of V/S pumping units should only be made after due consideration of the disadvantages.

- Except for engine drives, V/S adds significant cost and complexity, requires more equipment and more maintenance, and reduces reliability. Troubleshooting and repairs require personnel with on-the-job training. Each type of V/S drive requires a different kind of specialized training.
- If the operating personnel do not understand the equipment or find it difficult to maintain, they are likely to turn off the V/S drive, operate the pumps manually at C/S, and thus effectively lose the investment in expensive equipment plus the advantages it offers. Pump sumps designed for V/S pumping have insufficient storage for C/S pumping, and the frequent starting may burn out the motors and certainly reduces their life.
- V/S drives have less electrical efficiency than C/S motor drives (although V/S drives may use less energy because of lower pipeline friction losses).
- Avoiding vibration is more difficult with variable drive shaft speeds because the "windows" (the ranges of allowable supporting structure frequencies) for avoiding resonance are narrower. This avoidance should always be by design—never by luck (see Chapter 22).
- Instrumentation for regulating pump speed must usually be more refined, accurate, and costly than is required for C/S pumping.
- V/S drives are not well adapted to flat system H-Q curves because (1) good efficiencies cannot be obtained throughout the speed range; (2) small changes in speed produce large changes in discharge; and (3) energy losses (and costs) are high.
- Rapid changes in technology can make particular models obsolete.
- V/S drives may be much noisier than C/S drives.
- AFDs, in particular, are more vulnerable to lightning and electrical disturbances.

The disadvantages of V/S pumping are essentially advantages for C/S pumping. To complete the comparison, consider the following:

- The cost of the large wet wells required for C/S pumping may exceed the cost of V/S drives, but note that excavation costs are site-dependent.
- Pump sizes and wet well design can be easily coordinated so that the number of pump starts per hour (for C/S pumps) is well within acceptable limits.
- A wide range of flow rates can also be accommodated by using several C/S pumps, and the programmers for alternating C/S pumps are more reliable and less expensive than control systems for V/S units. A penalty for using C/S pumps is a larger station, more pumping units, more piping and fittings, and discrete increments of flow.
- Surges due to starting or stopping one pump are not as severe as those caused by a power failure, which may occur when several pumps are running and for which the station and pipeline must be protected from water hammer anyway. Solid-state starters or pump-control valves can eliminate surges in usual circumstances. If the pumping station discharges a small proportion of the treatment plant flow, the sudden change of influent flow rate due to the starting and stopping of a single pump may be negligible. The effect cannot be quantified, so this is a matter of judgment.

- Relatively constant water pressure from a booster pumping station may be obtained with C/S pumps either with flat H-Q curves (if suction pressure is constant) or with elevated or pressurized tanks.

Summary

If objective evaluation of these advantages and disadvantages leads to a decision to use V/S, select the best drive for the particular application.

15-2. Design Considerations

Firm pumping capacity is the maximum station discharge with the largest pump out of service. Similarly, a station with V/S pumps must be able to accommodate any flow (from minimum to maximum) and operate in a normal manner when any one of the pumps or drives is out of service.

A V/S pumping station that includes C/S pumps can provide the same results as a station with all V/S pumps, but only if the relationship between the sizes of the two kinds of pumps is correct (see Sections 15-5 and 15-10).

Pumps

The number of pumps to be installed in a station depends on (1) the range over which the influent flow rate varies, (2) the available pump selections, (3) the optimization of size of wet well versus the size of pumps, and (4) operating costs.

Wet Wells

For V/S pumping, the wet well need only be a sump for pumping at the instantaneous system flow rate. The only storage required is enough to allow pumps to be shut down or started up without an excessive change in water level, and the water surface area needed for that purpose is quite small.

Furthermore, the change of water surface elevation can be reduced to the amount needed for regulating the speed of the pumps. Most V/S control systems start and stop the pumps and regulate speed over a range of about 0.6 m (2 ft) of change in the water elevations in the wet well. (An even smaller range can be used, but a more sophisticated control system may be required.) Hence, some static head can be saved as compared with a C/S pumping system in which the wet well level typically fluctuates over a range of 1.2 m (4 ft) or more. Detailed designs of wet wells for V/S and C/S pumping are given in Examples 12-1 and 12-2.

15-3. Theory of Variable-Speed Pumping

It may be helpful to review the fundamental theory for centrifugal pumps in Section 10-3, which was explained primarily by Equations 10-15 through 10-21 and by Examples 10-2, 10-3, and 10-8.

In pump manufacturers' published data, H-Q curves are usually shown for impellers of several diameters at two or more speeds. Such curves drawn for pump speeds of 880, 695, and 585 rev/min are shown in Figure 15-1. At variable speeds, the pump actually operates on an infinite number of speed curves between the maximum and minimum limits. The minimum speed may be less than the lowest speed for which a published curve is available.

Most nonclog, dry-well wastewater pumps can be furnished with an impeller of any diameter desired between the minimum and maximum shown. For example, if the pump whose curves are shown in Figure 15-1 must deliver a maximum of $0.35\ m^3/s$ at 15.2 m (5500 gal/min at 50 ft), the curve for that impeller would pass through those coordinates and would be represented by the dashed lined labeled 20.1 in. (impeller diameter).

Speed versus Flow Curves

When discharging into a hydraulic system, the pump must operate on the system head-capacity (H-Q) curve at all flows. In Figure 15-2, a typical system H-Q curve has been superimposed on the 20.1-in. diameter impeller curves. The pump delivers $0.133\ m^3/s$ (2100 gal/min) at 10.8 m (35.5 ft) when it operates at 585 rev/min; $0.227\ m^3/s$ (3600 gal/min) at 12.3 m (40.5 ft) at 695 rev/min; and $0.347\ m^3/s$ (5500 gal/min) at 15.2 m (50 ft) at 880 rev/min. A curve of speed versus flow rate can be plotted from these data, as in Figure 15-3. This curve is probably the most meaningful depiction of V/S pump operation for a single pump. Note that the pump efficiencies (obtained from Figure 15-1) vary from 70% at maximum flow rate through 83% at midspeed to 77% at low speed—good efficiencies throughout the usual flow rate rates.

Affinity Law for Speed

To determine a speed versus flow curve for any pump impeller, find the flow rates at which three or more of

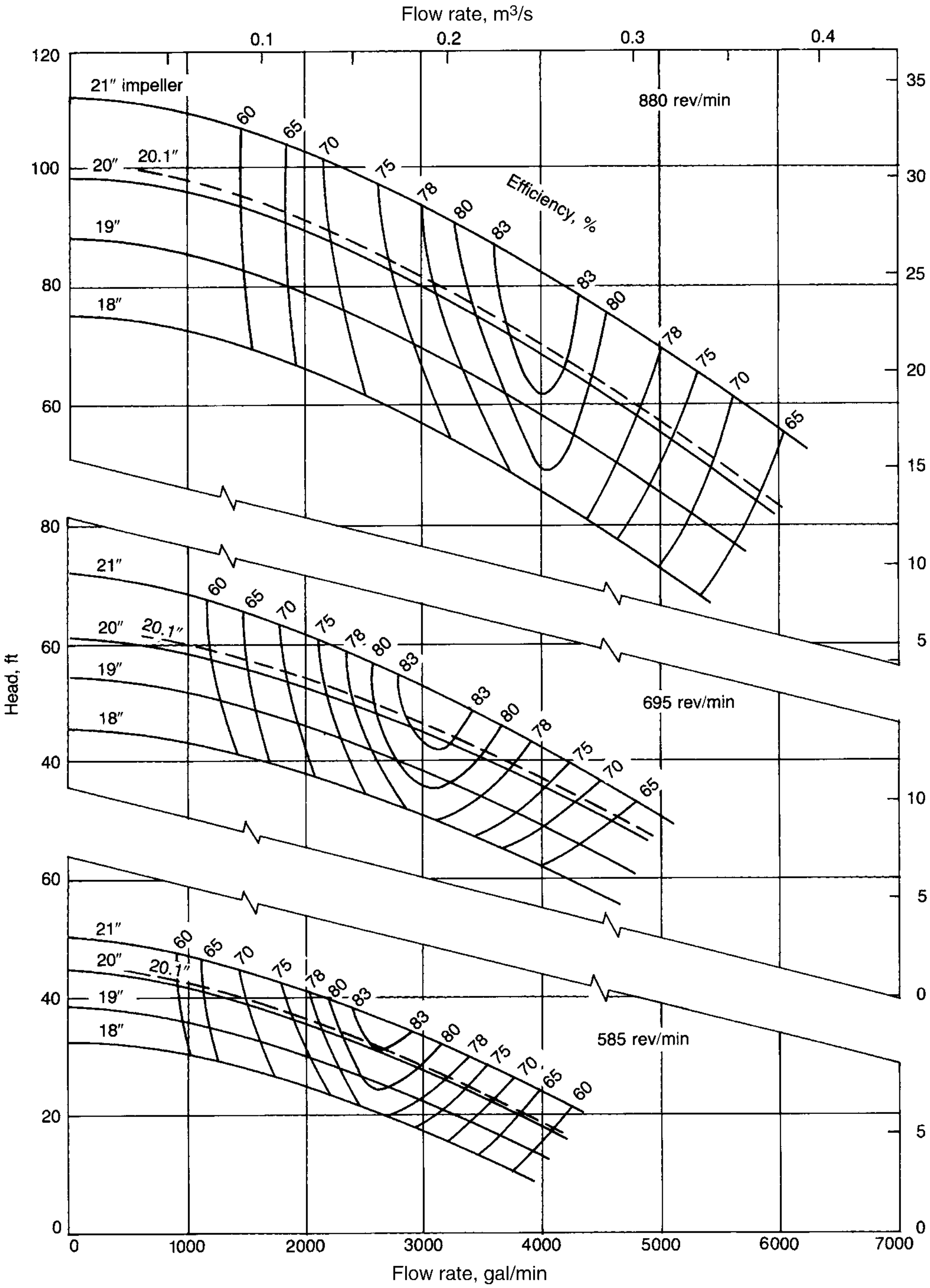


Figure 15-1. Pump characteristic curves at three speeds.

the pump H-Q curves (each at a different speed) intersect the system H-Q curve. If only one or two of the published pump curves intersects the system curve, points on another speed curve can be calculated from the affinity laws as shown in Example 10-2, Section 10-3. Note, incidentally, that points along the system

LIVE GRAPH
Click here to view

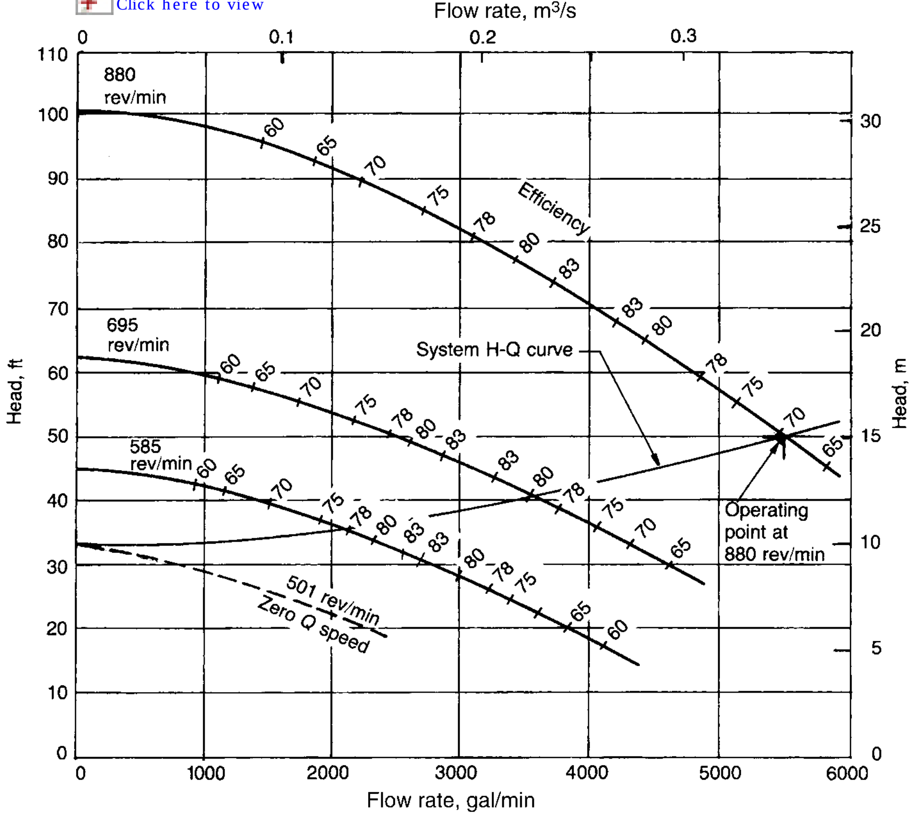


Figure 15-2. Pump characteristic curves for a 20.1-in. impeller.

H-Q curve are *not* "corresponding points" and, hence, the flow rate is *not* proportional to pump speed.

Minimum Operating Speed

The minimum operating pump shaft speed (zero Q speed) is defined as the speed at which the pump is no longer able to maintain discharge against the static head. Hence, it is unique for each pump and each pumping station, and it occurs at the speed for which the pump's shut-off head equals the static head.

At zero discharge (zero Q), and only at zero Q, the pump discharge head can change without a change in the flow rate. Hence, at this unique locus of points, Equation 10-16 is the only one needed for calculating the zero Q speed for a pump discharging into a hydraulic system. Solving that equation for the pump and system H-Q curve in Figure 15-2 gives

$$n_1 = n_2\sqrt{\frac{H_1}{H_2}} = 585\sqrt{\frac{33}{45}} = 501 \text{ rev/min} \quad (15\text{-}1)$$

and a position of the curve for this speed is shown as a dashed line in Figure 15-2. However, most pumps, especially medium and large ones, should not be operated more than momentarily at zero flow. At zero flow and at low discharge rates, the pumped fluid recirculates within the volute, which causes an unbalanced radial thrust against the impeller that results in a flexing of the pump shaft, which can cause bearing failure or fatigue fracture of the shaft (see Section 10-6).

As a rule of thumb, it is usually safe to operate a pump continuously at discharge rates of 30% or more of its best efficiency capacity (BEC), which is its discharge capacity at the best efficiency point (BEP) at the maximum recommended speed. The pump manufacturer's minimum discharge rate, however, should always be met; the minimum recommendation may be 50% or more of BEC for some pumps. By requiring custom-engineered pumps (heavier shafts and bearings), the problem with the minimum recommended operating speed can be avoided with most pump makes. But a custom pump, especially in a

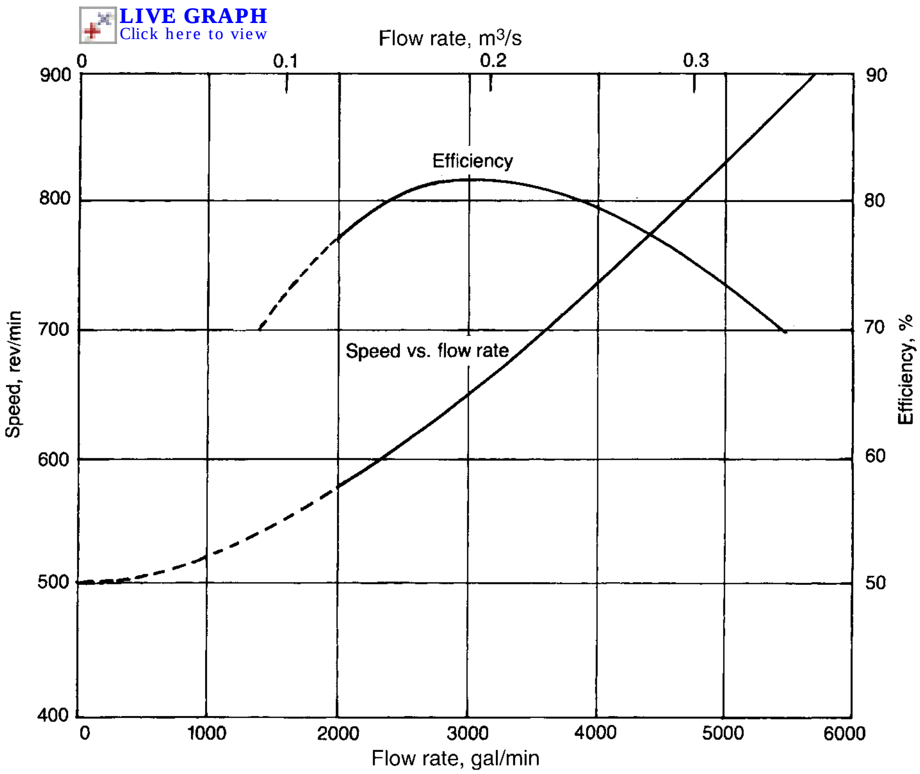


Figure 15-3. Speed versus flow rate and efficiency.

nonclog design, might be expensive, so discuss the subject with one or more manufacturers. Perhaps a larger number of smaller pumps would be a good alternative.

Most V/S drives provide a method for preventing the pump from operating below a preselected minimum speed. When the influent rate is less than the discharge rate of the pump at its preselected minimum speed, the pump is cycled on and off. There is no speed variation during the on-off operation. If the minimum influent rate is less than the minimum recommended discharge rate of a single pump, the wet well (and the sewer pipe if its storage capacity is utilized) should include sufficient storage to prevent cycling the pumps too frequently at this rate. When the influent rate again reaches the discharge rate of the pump at its preselected minimum speed, the pump is returned to V/S operation by the control system.

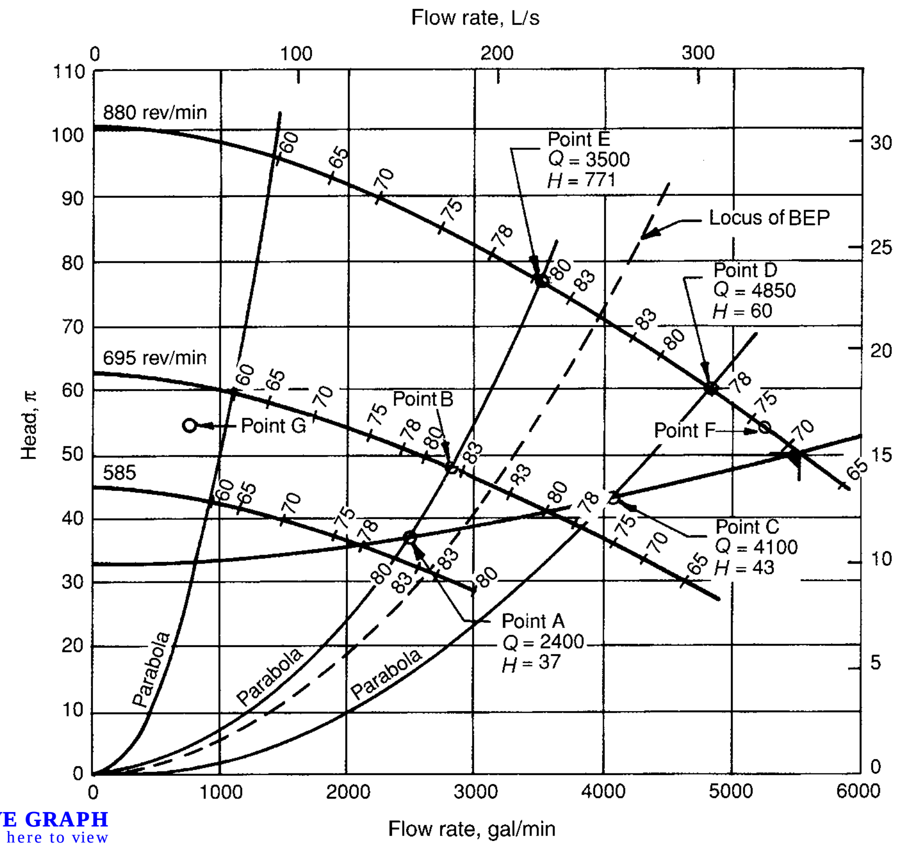
The following two methods can be used to avoid cyclic action of the "lead" pump:

- Use a larger number of smaller V/S pumps. For example, use four 3000-gal/min pumps instead of two 6000-gal/min units. Select the pump capacities so that the recommended minimum discharge rate of each pump is less than the minimum influent rate.
- Add two auxiliary V/S pumps that operate only when the influent rate is less than the recommended minimum discharge rate of the main V/S pumps. Select the capacity of each auxiliary pump to deliver a maximum flow that slightly exceeds the recommended minimum discharge rate of one of the main pumps.

Pump Efficiencies

As speed changes, efficiencies follow parabolic curves with their apexes at the origin of the graph, as shown in Figure 15-4. Thus, except for a slight shift with Reynolds number, efficiency is independent of speed. Consequently, if pump H-Q curves are constructed for different speeds, the efficiencies can be plotted at the same time. By projecting efficiencies from a pump H-Q curve along a parabola to the system H-Q curve, the efficiency at any pump operating point can be found.

An alternative way to find pump efficiencies along a system curve is to superimpose characteristic curves for selected pump impeller curves at two or preferably three speeds (as in Figure 15-2) onto the system H-Q curve and connect points of equal efficiency.



LIVE GRAPH
Click here to view

Figure 15-4. Characteristic curves for a single pump. Points A and B are corresponding points and lie on a parabola through the origin, as do corresponding points C and D. Points A and C on system curve are not corresponding points.

The pump efficiencies along the system H-Q curve can then be plotted as shown in Figure 15-4.

Power Required

When the pump discharges into a hydraulic system, the power delivered by the pump at any point on the system head curve is determined by the flow rate and the head at that point (see Equation 10-6).

According to the affinity laws, the ratio of power *delivered* by a pump (water power) at "corresponding points" (such as Points A and B in Figure 15-4) at different speeds equals the cube of the ratio of speeds. *But points along the system H-Q curve (such as Points A and C) are not "corresponding points," so the power delivered along the system curve does not vary as the cube of the ratio of speeds.* Using Equation 10-6 to calculate the ratio of power delivered at 695 and 585 rev/ min along the system head curve in Figure 15-4 gives:

$$\frac{(3550)\ (40.5)}{(2100)\ (35.5)} = \left(\frac{695}{585}\right)^{x}$$

from which $x = 3.8$. But x varies along the system head curve. For the same pump, the exponent x can vary from less than 2 to more than 4 along the system curve.

Moreover, it is the *required* pump shaft or brake power that interests the designer. The power required by the pump at any point on the system curve is equal to the power delivered at that point divided by the pump efficiency at that point. Pump efficiency can vary considerably over the system curve, and it can cause the pump power requirements to further deviate from a cube relationship. Thus, drive manufacturers' efficiency data (which are based on a cube relationship of speed and required power) are not applicable.

Always compute power requirements from head, discharge, and efficiency (Equation 10-7)—not from the affinity laws.

15-4. Pump Selection

All of the arrangements described here include sufficient pumping capacity to discharge the peak influent rate when any single pump is out of service.

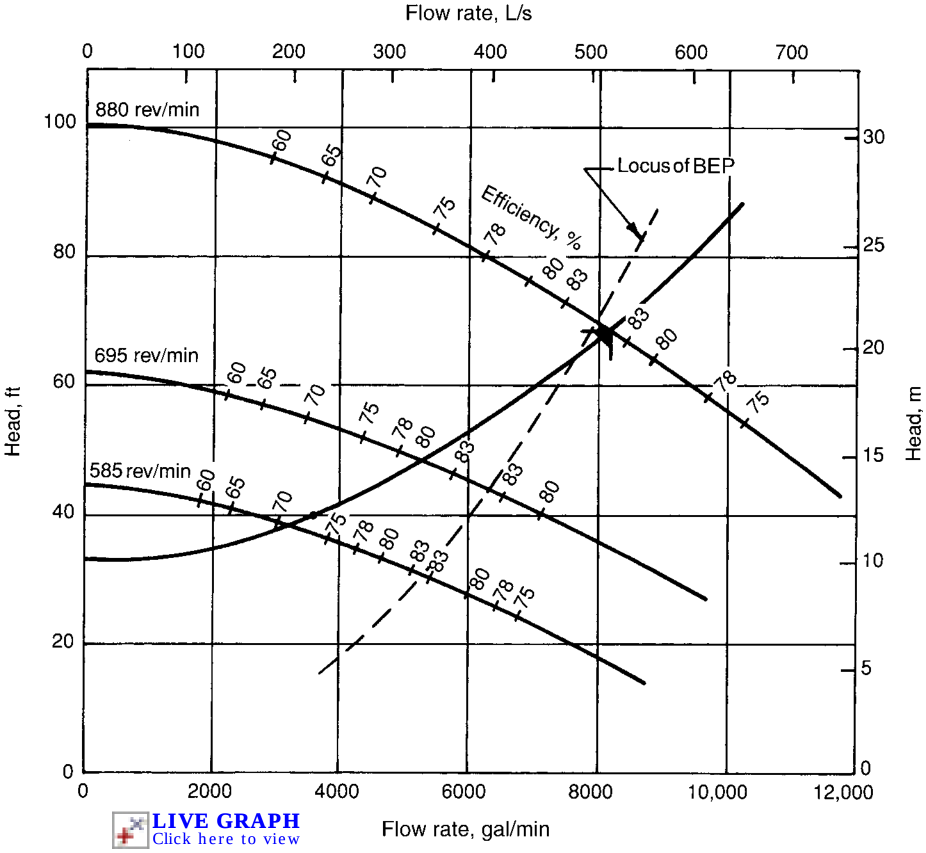
Two-Pump Facility

For an installation with two pumps, each capable of discharging the peak influent rate, a single duty pump with its peak discharge rate well to the right of the BEP provides satisfactory efficiencies throughout the entire operating range, and very good efficiencies in the center of the range. An example of the capability of a single pump is shown in Figure 15-4. Note that the efficiencies are 69% or more over a range of flows equal to 2.75:1.

Three Variable-Speed Pumps

In the usual arrangement of a three-V/S-pump system, two pumps must be capable of discharging the design flow rate, while the third is idle as a standby unit. Unless the system curve is flat, a single pump must be capable of discharging somewhat more than one-half of the peak influent rate. The "lead" pump runs alone during periods of low flow. The "lag" pump is started when the influent rate slightly exceeds the maximum discharge rate of the lead pump, and both pumps operate until the influent rate again falls to slightly within the capability of the lead unit.

Assume, for example, that a pumping station must discharge flow rates from 0.13 m^3/s (2000 gal/min) to 0.50 m^3/s (8000 gal/min) at a static lift of 10 m (33 ft) and a TDH at peak discharge of 20 m (67 ft). The system H-Q curve is shown in Figures 15-4 and 15-5.



LIVE GRAPH
Click here to view

Figure 15-5. Characteristics for two pumps (per Figure 15-4) in parallel.

The friction head is moderate and the curve is relatively flat, so a pump with comparatively steep H-Q curves should be selected. When two pumps are operating, the BEP should be to the left of the operating point at peak flow rate so that the system H-Q curve sweeps across the curves of highest efficiency. Therefore, the BEP of a single pump at top speed (880 rev/min in Figure 15-4) should be at the left of the mid-range discharge [0.35 m^3/s (5500 gal/min)], and for a single pump operating, only the higher efficiency curves should straddle the system H-Q curve. The minimum discharge should be at least 30% of the maximum. The pump depicted meets all of these requirements and thus appears to be a good choice.

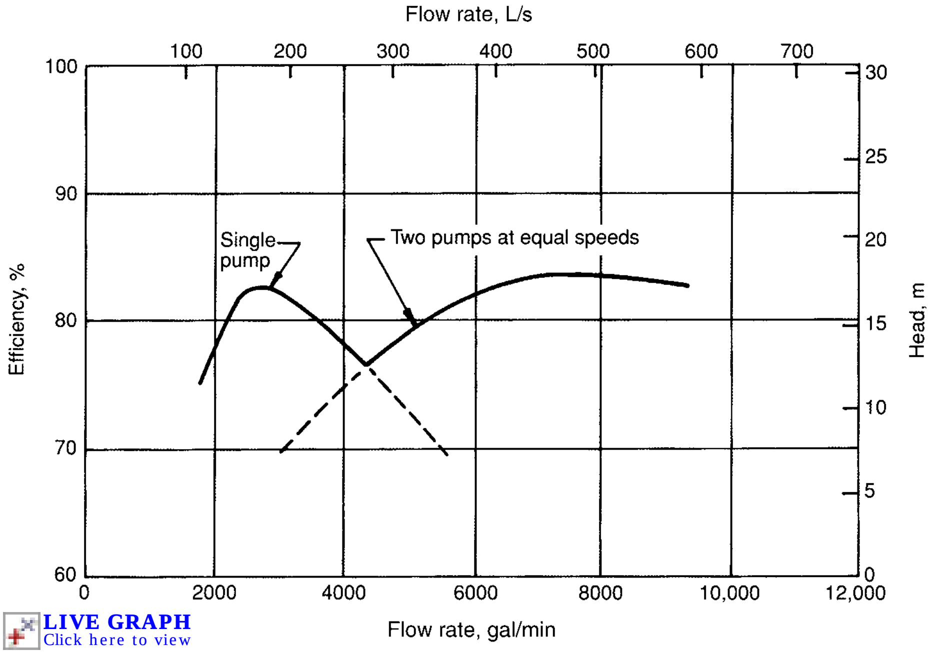
Load-Sharing Operation

When two of the above pumps operate in parallel at the same speed, the discharge (abscissa) is doubled with no change of head (ordinate), as is shown in Figure 15-5. (Review Figure 10-24 and the accompanying text if an explanation is needed.) Note that the locus of BEP for the two pumps at full speed is slightly to the left of the peak discharge. The efficiencies throughout the full range of flows from 0.13 m^3/s (2000 gal/min) to the peak discharge, shown in Figure 15-6, can be kept between 76 and 83%. Throughout the entire range of flow rates, the pumps operate at 36% or more of maximum capacity, and neither pump would cycle on and off frequently. Hence, the pump selection is indeed good.

Usually a lag pump is started when the influent rate slightly exceeds the capacity of the lead pump, and the two pumps then operate at the same speed. The lag pump runs continuously until the influent rate is again well within the capability of the lead pump. In the example of Figure 15-6, however, the lag pump could be stopped or started near the intersection of the efficiency curves for single- and for double-pump operation.

In staggered operation, the lead pump operates at maximum speed while the lag pump operates on reduced-speed curves. Staggered operation is a poor way to operate pumping equipment and should not be used. The complexity of the controls is increased, the energy use is increased, and ragging at the reduced speed of the lag pump is increased. The discharge rates of the pumps at various influent rates can be calculated as follows:

- For any influent rate that exceeds the maximum capacity of a single pump, find the head, H', from the system H-Q curve.
- Determine the capacity of the lead pump at H' from its maximum speed curve.



LIVE GRAPH
Click here to view

Figure 15-6. Efficiencies for one and two variable-speed duty pumps in load-sharing operation.

- The discharge rate of the lag pump equals the influent rate minus the lead pump capacity.

A portion of Figure 15-4 is reproduced in Figure 15-7. As an example of staggered operation, consider a flow rate of $0.38\,m^3/s$ (6000 gal/min) shown as Point E on the system H-Q curve. The head is 16.5 m (54 ft), and at this head the lead pump delivers about $0.33\,m^3/s$ (5200 gal/min) at maximum speed (Point F). The lag pump, therefore, must deliver $0.38 - 0.33 = 0.05\,m^3/s$ ($6000 - 5200 = 800$ gal/min)—also at 16.5 m (54 ft) head, which is plotted as Point G. By interpolating curve GH between the 585- and 695-rev/min curves, the shut-off head is found to be 17.4 m (57 ft). The shut-off head for 695 rev/min is 19.2 m (63 ft), so to produce a head of 17.4 m (57 ft), the speed of the pump would be, from Equation 10-16,

$$n_1 = 695\sqrt{17.4/19.2} = 662\,\text{rev/min}$$

or

$$n_1 = 695\sqrt{5.7/63} = 661\,\text{rev/min}$$

Interpolating between curves may be inaccurate where the curves are far apart (as in Figure 15-7), but accuracy can be improved by using the affinity laws (Equations 10-15 and 10-16 simultaneously).

The combined curve for the lead pump (at 880 rev/min) and the lag pump (at 661 rev/min) is found by adding the abscissas of the lag and lead pump curves (see Figure 10-24 for further explanation). The efficiency of the lead pump operating alone is the same as that shown for the single pump in Figure 15-6. The efficiency of the lead pump when operating with the lag pump is found by projecting the head horizontally from Point E on the system H-Q curve in Figure 15-7 to an intersection (Point F) with the maximum speed curve (880 rev/min), transferring Point F to Figure 15-4 (note difference in horizontal scales), and interpolating between the efficiency lines to obtain an efficiency of approximately 73%. The lag pump efficiency for a total discharge of $0.38\,m^3/s$ (6000 gal/min) is the efficiency at Point G, which (transferred to Figure 15-4) is found to be below the pump manufacturer's data. It is probably about 55%.

Efficiencies at other discharges are found in the same manner. From these efficiencies and the heads and discharges, a plot of required pump shaft power versus discharge can be drawn as shown in Figure 15-8.

Comparison of Operating Modes

The power requirements for "load sharing" by the same pumps as above are also shown in Figure 15-8 for comparison. Staggered operation requires more power than load sharing whenever the flow rate varies between 101 and about 150% of the capability of a single pump. The influent flow rate could remain within this range for many hours at a time, with a consequent waste of considerable energy. Of even

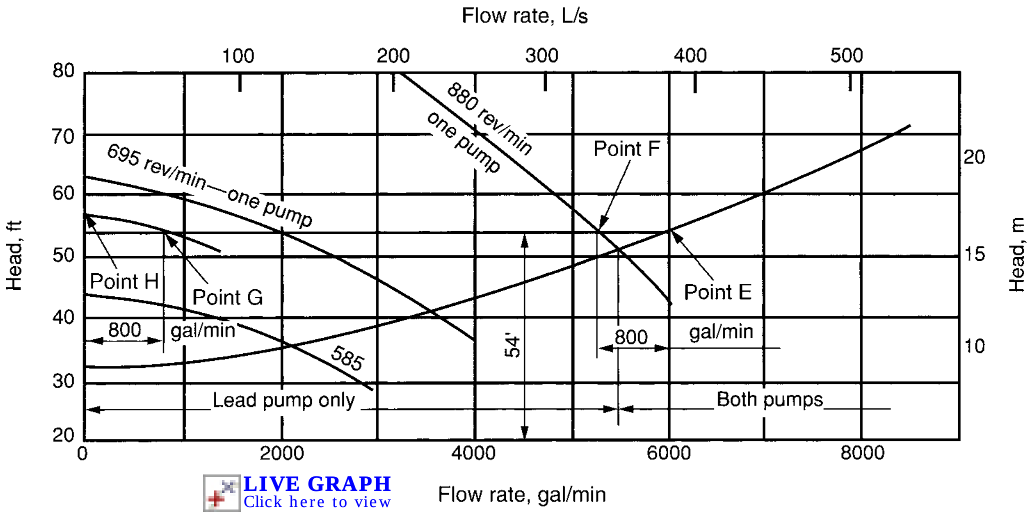
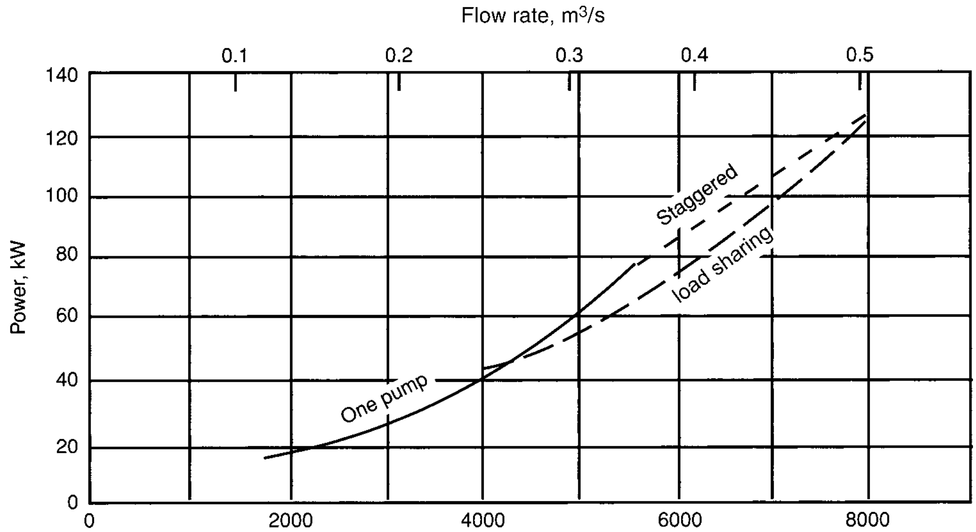


Figure 15-7. Two variable-speed pumps in staggered operation.



LIVE GRAPH
Click here to view

Figure 15-8. Pump shaft power consumption for two variable-speed pumps.

more concern is the operation of the lag pump at very low (and potentially damaging) discharge rates for long periods of time.

Selections for Steep System Curves

If the friction head is large in relation to the static head, the best pump selection would be to have the BEP well to the left of the single pump maximum discharge rate. The pump shown in Figure 15-9 would be an excellent selection to deliver one-half of a peak influent rate of $0.58\,m^3/s$ (9200 gal/min) with the system H-Q curve shown.

Four Variable-Speed Pumps

The same principles of selection apply to systems with four or more V/S pumps. A combination of V/S and C/S pumps, however, may provide some of the advantages of V/S drives at a lower equipment cost, but see Section 15-5 for the disadvantages.

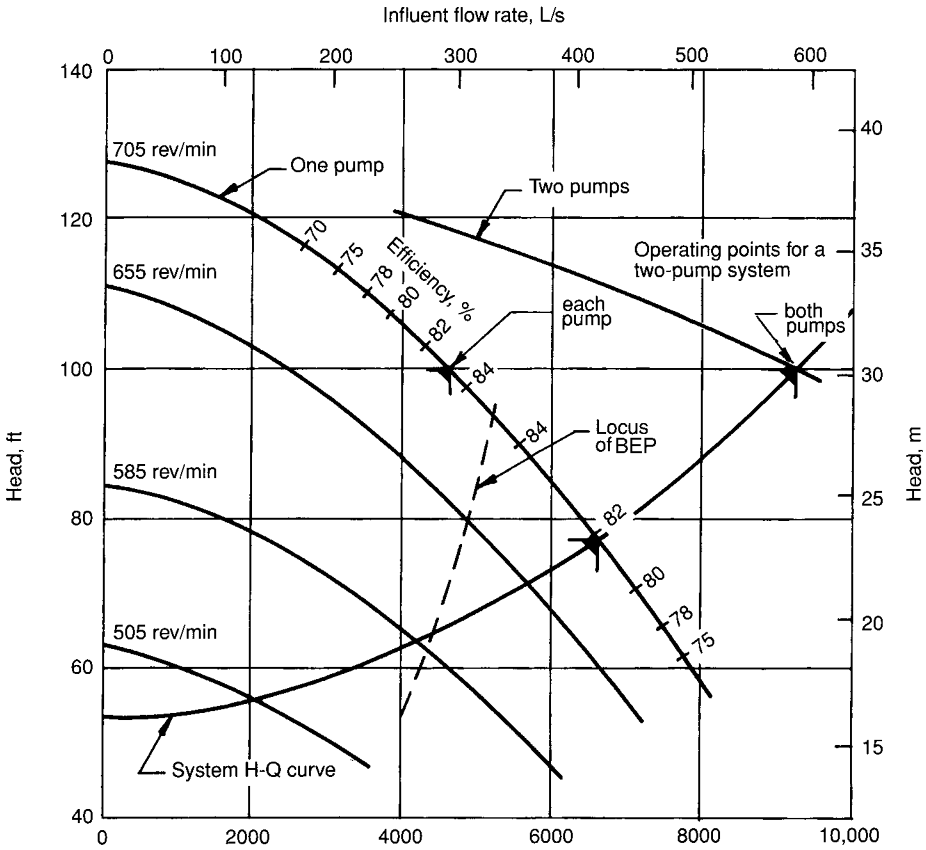
15-5. Variable- and Constant-Speed Pumps in Simultaneous Operation

To avoid the cost of V/S drives for every pump, some designers try to reduce construction costs by substituting a C/S drive for one or more V/S drives. A combination of drives is acceptable subject to the constraint that no pump may operate outside its AOR at any station flow rate. Furthermore, most pumps cannot operate at less than about 35% of their maximum capacity (Q_{max}).

Assume, for example, a station capacity of 200 units, a minimum flow rate of $0.35Q_{max}$ (=35 units), and (for simplicity) a relatively flat station H-Q curve. The following scenarios are possible.

1. Three V/S pumps, each of 100 units capacity. One V/S pump is a standby. Below 35 units, one pump must cycle on and off.
2. Two V/S pumps (one a standby), each of 121 units capacity and one C/S pump of 79 units capacity. Below 42 units, one V/S pump must cycle on and off. Some utility managers might object to a loss of flexibility; others might not.
3. Three V/S pumps and one C/S pump, each of 67 units capacity. Any one is a standby. Any flow rate from 23 to 200 units can be accommodated with any pump out of service. The extra piping and increased size of the wet well would probably make this alternative more expensive than Scenario 1.
4. Three V/S pumps and two C/S pumps, each of 50 units capacity. Any one is a standby. Below 18 units, one V/S pump must cycle on and off.

Now assume the minimum flow rate for a V/S pump is $0.5Q_{max}$. The following scenarios are possible.



LIVE GRAPH
Click here to view

Figure 15-9. Variable-speed pumps for steep system H-Q curves.

1. Three V/S pumps, each of 100 units capacity. One V/S pump is a standby. Below 50 units, one pump must cycle on and off.
2. Two V/S pumps (one a standby), each of 133 units capacity and one C/S pump of 67 units capacity. Below 67 units, one pump (preferably a V/S unit for its soft start ability) must cycle on and off.
3. Three V/S pumps and one C/S pump, each of 67 units capacity. Any one is a standby. Any flow rate from 33 to 200 units can be accommodated with any pump out of service. Below 33 units, one V/S pump must cycle on and off.
4. Three V/S pumps and two C/S pumps, each of 50 units capacity. Any one is a standby. Below 25 units, one V/S pump must cycle on and off.

These explanations are summarized in Table 15-1.

15-6. Special Design Considerations

Maximum Speed Limit

A pump must *never* be operated at a flow rate greater than that shown by the manufacturer's plotted curve, nor should it operate beyond its AOR. In other words, *never* extrapolate a pump H-Q curve. Hence, it may be necessary to limit the speed of a pump to ensure that the pump curve not only intersects the system curve but also overlaps it somewhat. Consider the range of possible system curves, and also note that pipe friction can be lower than anticipated, particularly when the pipe is new, which may further limit the speed range of the pump.

A discharge pressure-control override can also be used when several stations discharge into a common

Table 15-1. Station Flow Rates for Various Combinations of V/S and C/S Pumps

Scenario	V/S pumps	C/S pumps	Standby	Max flow rate	Min flow rate
For minimum flow rate in any V/S pump at 35% capacity:					
1	3 @ 100	0	1 V/S	200	35
2	2 @ 121	1 @ 79	1 V/S	200	43
3	3 @ 67	1 @ 67	1 C/S or V/S	201	23
4	3 @ 50	2 @ 50	1 C/S or V/S	200	18
For minimum flow rate in a V/S pump at 50% capacity:					
1	3 @ 100	0	1 V/S	200	50
2	2 @ 133	1 @ 67	1 V/S	200	67
3	3 @ 67	1 @ 67	1 C/S or V/S	201	33
4	3 @ 50	2 @ 50	1 C/S or V/S	200	25

Note that in Scenarios 2, 3, and probably 4, the C/S pump will operate very little of the time. The total cost of these scenarios (especially Scenario 4) may be greater than that of Scenario 1.

force main where the head at any station can be very low at times.

Control of Pumping Units

A pump-control system should be designed so that the failure of any component or the trip of any single protective device (downstream from the main breaker) does not disable more than one pump. A complete and separate V/S drive for each V/S pump is strongly recommended to ensure that the facility can operate in a normal manner with any single V/S drive out of service.

Pump Failure Detection

Pump failure detection is a highly desirable feature that can usually be included in the pump-control system at little cost. If the pumps are equipped with outside-lever check valves, the valves can be fitted with limit (position) switches to detect check valve arm movement and therefore flow (or the lack of flow). The limit switch is actuated by the lever when the valve is fully closed; it is deactivated when the pump begins to deliver a small flow rate. Regardless of the cause, the check valve limit switch can detect the failure of the pump to discharge.

The electrical circuitry is designed to de-energize the motor of any pump whose check valve remains closed for a predetermined period of time (usually less than 3 min for V/S pumps) when the pump is required to operate. An alarm is signaled, and the standby pump replaces the unit that failed.

Check valve limit switches can also prevent energization of the motor of any pumping unit whose discharge check valve is not fully closed. Some drives can be severely damaged or destroyed and shafts can break if the motor is energized while the pump is back-spinning.

If the check valves do not have outside levers, a pressure switch located between the pump discharge nozzle and the check valve is sometimes used as a flow sensor. Actually, the check valve only indicates pressure, not flow, and is misleading if flow is blocked by a downstream obstruction. A better flow indicator is a vane switch in clean water applications and, for wastewater, a thermal-dispersion switch or a Doppler flow switch (see Section 20-5). When used for wastewater or dirty water, any pressure device (such as a bourdon tube) should be filled with a clear liquid (glycerin, for example), which is separated from the dirty water by a flexible diaphragm. See Figure 20-6.

Power-Operated Check Valves

If the pump discharge check valves are power-operated (hydraulic or electric), it may be necessary to include a relatively sophisticated control system to protect the V/S pump drives against back-spinning while the drives are energized. When a pumping unit is required to operate and the pump drive is activated, the check valve should not begin to open until the pressure on the upstream side of the valve exceeds the pressure on the downstream side. (The pressure can be sensed, as described previously.) When the pump is no longer needed, the check valve should close first, and the pump should not be stopped until the valve is at least 95% closed.

The pumping unit should be de-energized when:

- The pump discharge pressure does not exceed the pressure in the force main within a short interval (say, 60 s) after the pump is started.
- The check valve does not open fully within a few minutes after the pump discharge pressure exceeds the pressure downstream from the check valve.
- The check valve does not close fully within a short interval (say, 3 min) after the pumping unit is signaled to shut down.

The control system should also prevent energization of the motor of any drive whose check valve is not fully closed.

15-7. Analysis of Variable-Speed Booster Pumping

The purpose of a V/S booster pump is to maintain a nearly constant discharge pressure while delivering the variable flow rate needed in a closed distribution network. Pump speed variation controls the pump discharge pressure, and the flow rate is determined by the system demand.

To analyze the operation of a pump that discharges into a hydraulic network, both the V/S pump characteristic curves and the curve of required pump differential head versus flow rate must be considered. The pump must always operate at the intersection of the impeller's characteristic curve (at the speed of rotation) with the required differential head curve.

Pump Characteristic Curves

The curves for a typical horizontal split-case pump, with its impeller trimmed to deliver 681 m^3/h (3000 gal/min) at 33.5 m (110 ft) at maximum speed, are shown in Figure 15-10. Note the flatness of these curves.

Required Differential Head Curves

The significant contrast between wastewater and booster pumping is that the system H-Q curves for the two are derived differently and usually have

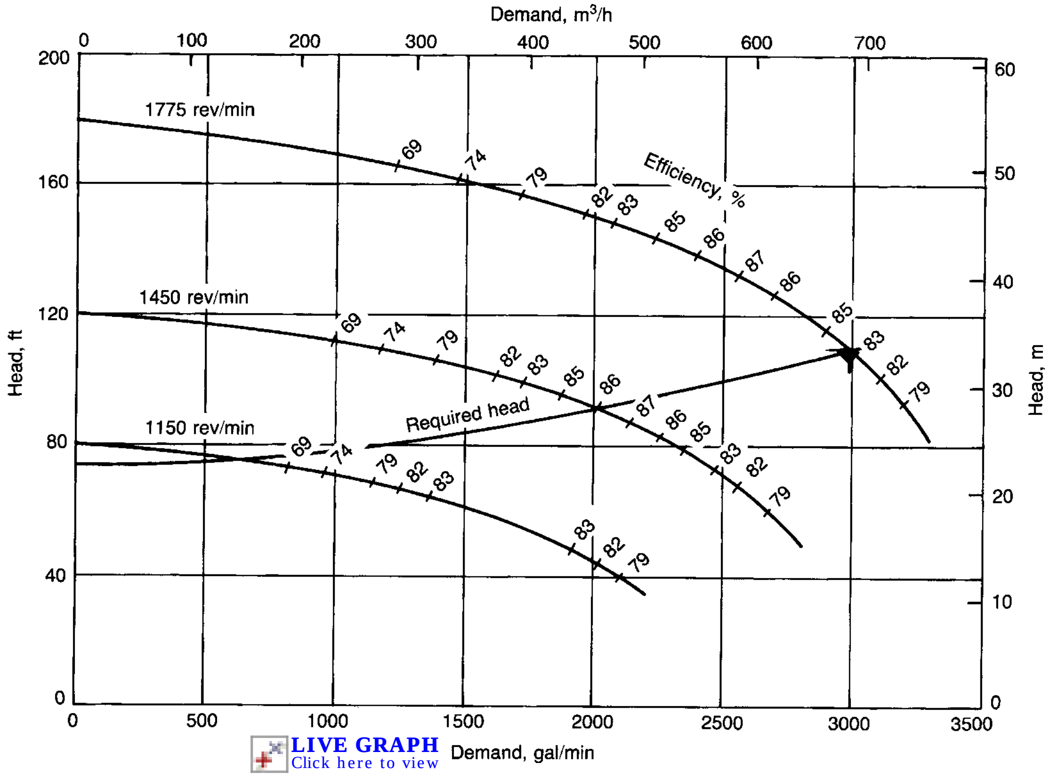


Figure 15-10. Characteristic curves for a booster pump.

different shapes. The system H-Q curve for a booster pumping facility is actually a "required differential head," which represents the differential head that must be added to the suction head to produce the desired constant discharge pressure at all flow rates.

There are four basic types of suction pressure variation.

- The suction pressure decreases with an increase of the pump discharge flow rate due to headlosses in the suction piping system.
- The suction pressure remains constant regardless of flow rate. An example is a pump that takes suction directly from a tank in which the water level is automatically maintained relatively constant by a float valve on the supply inlet.
- The suction pressure varies independently of flow rate. An example is a relatively small pump that takes suction from a large city main in which the pressure varies but is essentially unaffected by flow through the pump. Thus, there are an infinite number of suction head curves.
- The suction pressure decreases with an increase in the pump discharge rate and also varies independently of flow rate. Examples are (1) a booster pump in a city water main in which pressure varies due both to the pump discharge and to other demands, and (2) an in-line transmission booster pump.

Suction Pressure Varies with Flow Rate

The curve of required differential head that the pump must develop is shown by the solid lines in Figure 15-11 to be the desired discharge head minus the curve of suction head. But in actual practice, the booster pumps commonly used do not maintain a constant discharge pressure. Because of the requirement of the control system for a change in signal to the drive, the pump discharge pressure usually has a droop of about 10% from zero to maximum flow rate. The droop is not usually linear, but its actual curve is not important; a linear droop is used here for simplicity. The required differential head curve incorporating the droop is shown by the dashed line in Figure 15-11c, and it is also superimposed on the pump characteristic curves in Figure 15-10, where it is labeled "Required head." Droop can be eliminated with a reset feature in the control system. Elimination of the droop, however, is complex, expensive, and not usually necessary because a 10% droop is not objectionable.

The curve of pump speed versus flow rate shown in Figure 15-12 can be obtained as explained for Figure 15-3. The pump can, of course, operate at any speed between the limits shown by the solid line and at speeds less than 1150 rev/min, as indicated by the dashed line. However, operation at zero flow, or even at low discharge rates, should be avoided as

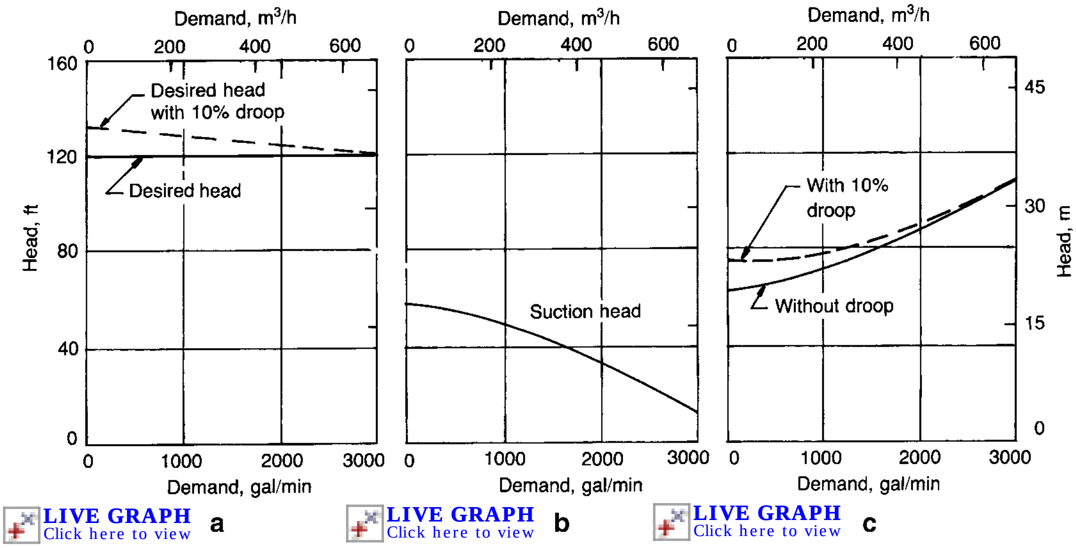


Figure 15-11. Differential head for a suction head varying with flow rate. (a) Discharge head; (b) suction head; (c) required differential head. Note: (a) − (b) = (c).

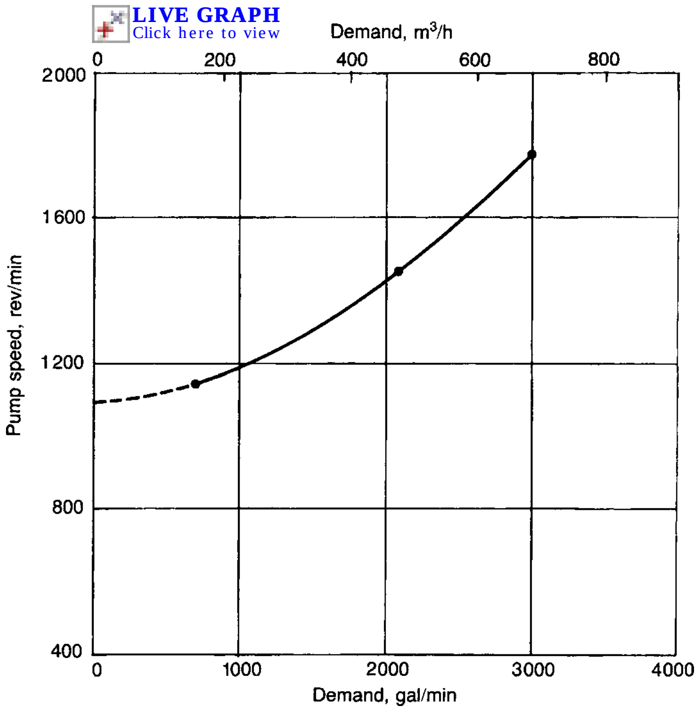


Figure 15-12. Speed versus flow rate for a booster pump.

explained in Section 15-3. Again, the speed-flow rate curve is the most significant depiction of V/S booster pump operation.

Suction Pressure Constant

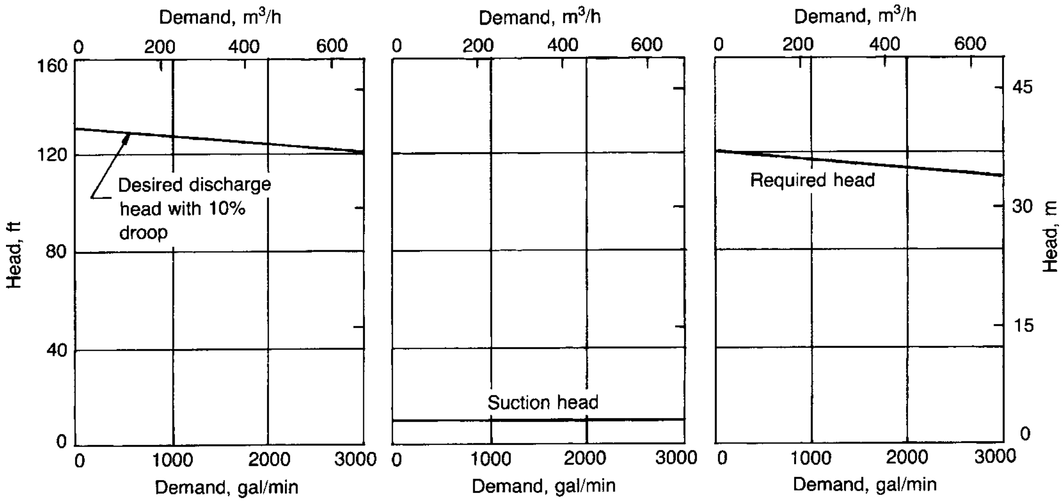
The development of the required differential head curve is shown in Figure 15-13. The superposition of this curve on the pump curves of Figure 15-10 is shown in Figure 15-14. The zero Q speed is found at the intersection of the 1450-rev/min pump curve with the required head curve, but if it were not, it could be found as explained in Section 15-3.

If a pump with a very flat curve is used (as shown in Figure 15-15), the zero Q speed can be calculated to be 1760 rev/min and the pump would always operate within 1% of maximum speed, regardless of flow rate. (All of the curves shown here are published curves for actual pumps offered by well-known manufacturers.) Hence, *there is nothing to be gained by V/S operation of a flat-curve pump with a constant suction head.* A flat-curve pump provides good pressure regulation where the flow rate fluctuates rapidly. The use of steep-curve pumps is sometimes desirable because they permit some adjustment of the discharge pressure without changing the pump impellers. The discharge pressure of a steep-curve V/S pump fluctuates with abrupt changes in demand because the inertia of the rotating elements prevents instantaneous changes of pump speed.

Suction Pressure Varies Independently of Flow Rate

The development of the required differential head curves is shown in Figure 15-16. The minimum suction head requires the maximum required differential head and vice versa. The pump and differential head curves are similar to those shown in Figure 15-14 except that there are an infinite number of required head curves as the suction head changes between the minimum and maximum of Figure 15-16b. The maximum speed pump curve and the required head curve for maximum suction head *must* intersect (which may require a steep-curve pump).

If the pump curve is very flat, the pump speed changes are almost entirely to compensate for suction pressure variations. But regardless of the manner in which the suction head varies, the drive automatically regulates the pump speed to deliver the desired discharge pressure at any flow rate within the capability of the pump.



LIVE GRAPH
Click here to view a

LIVE GRAPH
Click here to view b

LIVE GRAPH
Click here to view c

Figure 15-13. Differential head for a constant suction head. (a) Discharge head; (b) suction head; (c) required differential head. Note: (a) − (b) = (c).

LIVE GRAPH
Click here to view

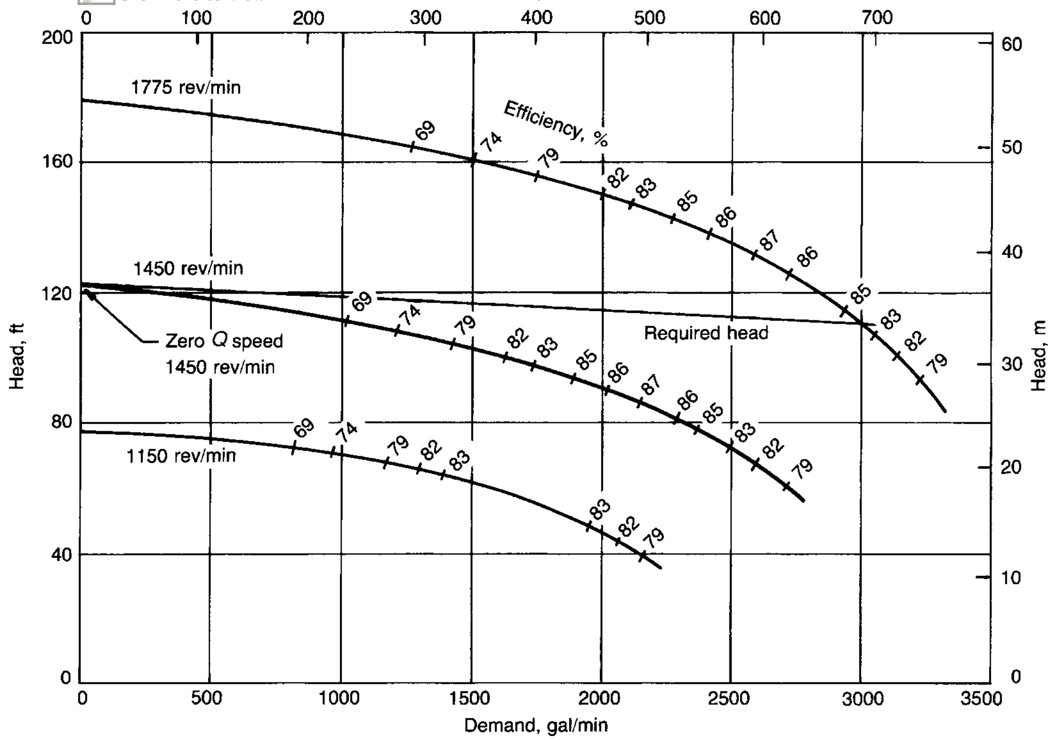


Figure 15-14. Booster pump and required differential head curve for a constant suction head.

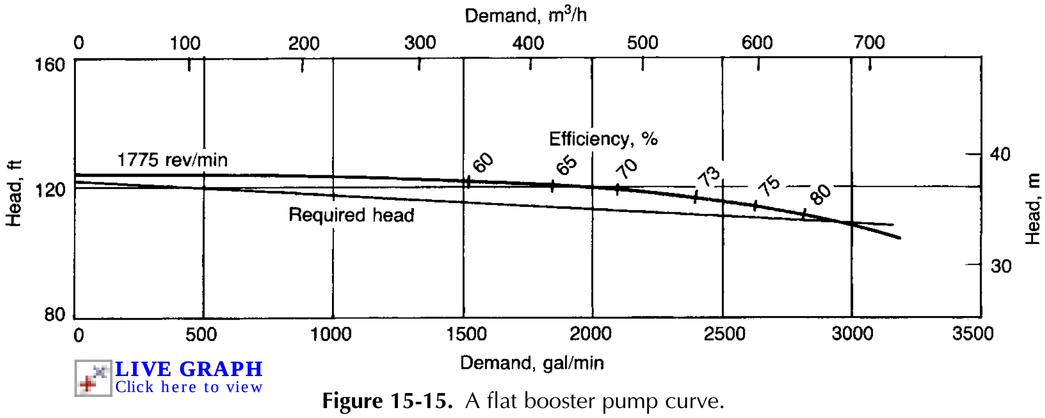


Figure 15-15. A flat booster pump curve.

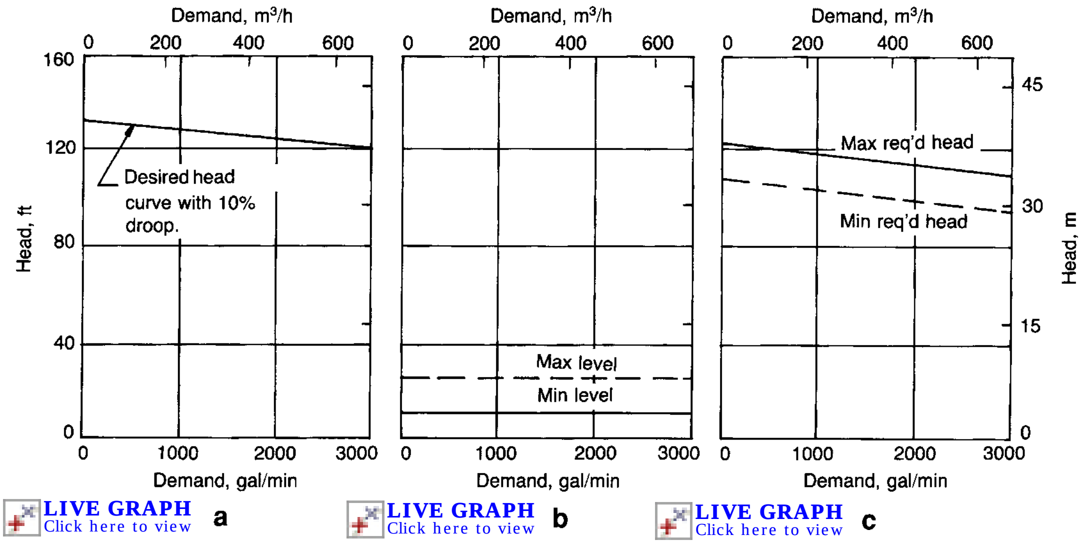


Figure 15-16. Differential head for a suction head that varies independently of flow rate. (a) Discharge head; (b) suction head; (c) required differential head. Note: (a) – (b) = (c).

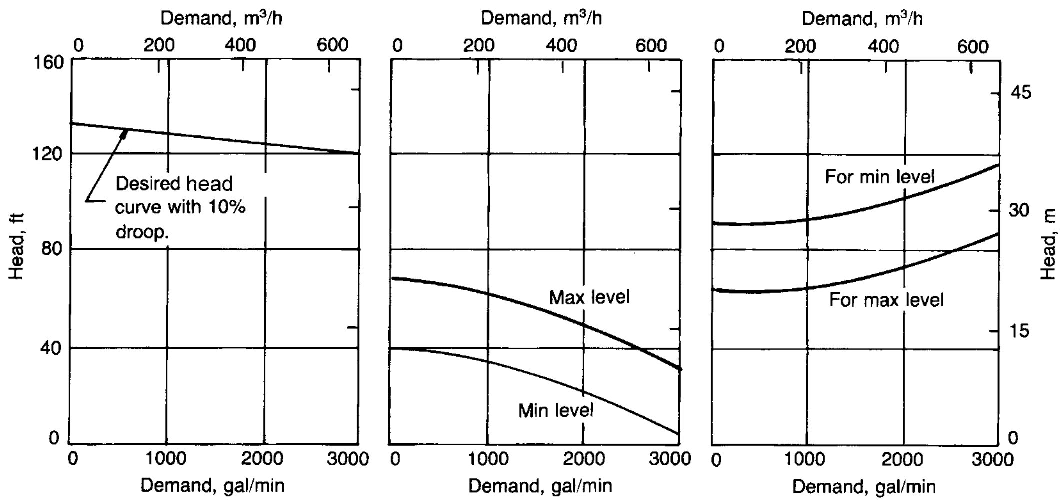
Suction Pressure Varies both Independently and Dependently with Flow Rate

The development of the required differential head curves is shown in Figure 15-17 for a minimum suction head of 12.2 m (40 ft) and a maximum suction head of 21.3 m (70 ft). The differential head is not a parabola with its apex at zero discharge because the droop causes it to be skewed.

The required differential head curves for the extremes of suction head in Figure 15-17c are superimposed on the pump curves of Figure 15-10 and shown in Figure 15-18. The pump operates on an infinite number of system curves between those for minimum and maximum suction head.

15-8. Minimum Flow Rate

A minimum speed feature in a V/S drive *cannot,* by itself, prevent the operation of the pump at low flow rates. Without a bypass, the pump cannot deliver more water than the amount that is being drawn from the distribution system. If the drive prevents operation at speeds lower than a preselected minimum, the pump must operate on its curve for that speed when the demand is less than the flow rate at which the minimum speed curve intersects the required head curve. If the pump with the characteristic curves of Figure 15-18 were prevented from operating at speeds less than 1300 rev/min, the pump would operate on the solid line ABCD



 a b c

Figure 15-17. Differential head for suction pressure varying both dependently and independently of flow rate. (a) Discharge head; (b) suction head; (c) required differential head. Note: (a) – (b) = (c).

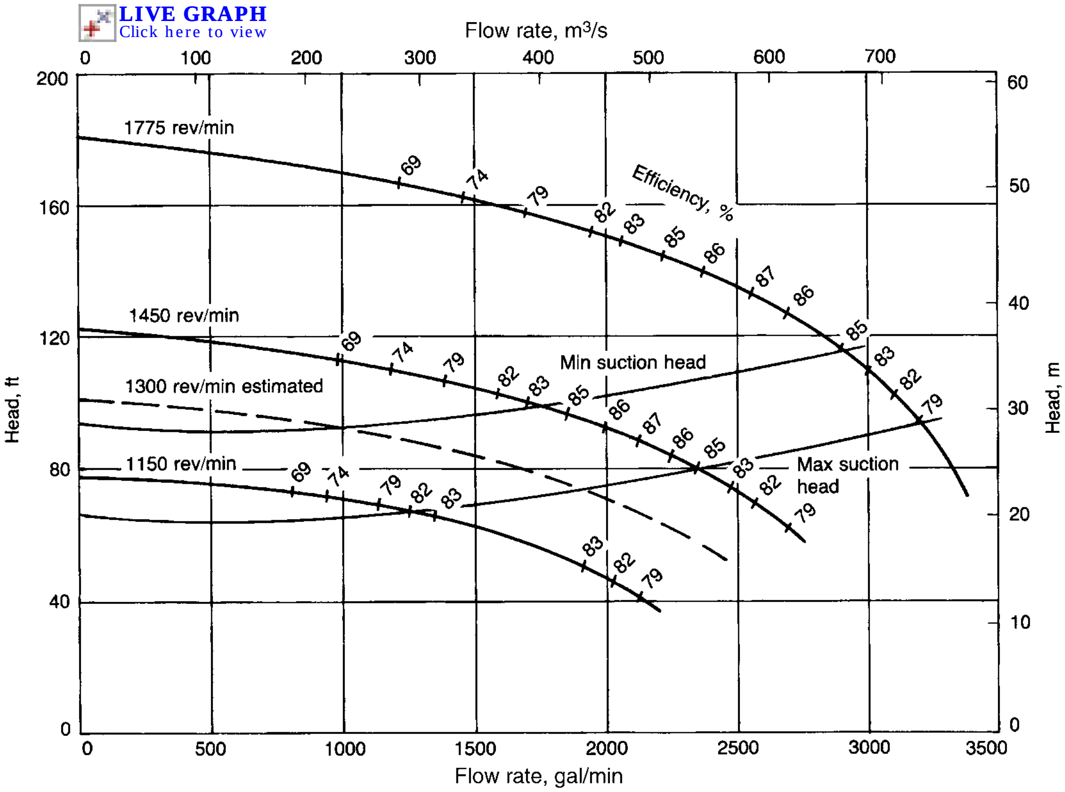


Figure 15-18. Suction varying both dependently and independently of flow rate combined with pump curves.

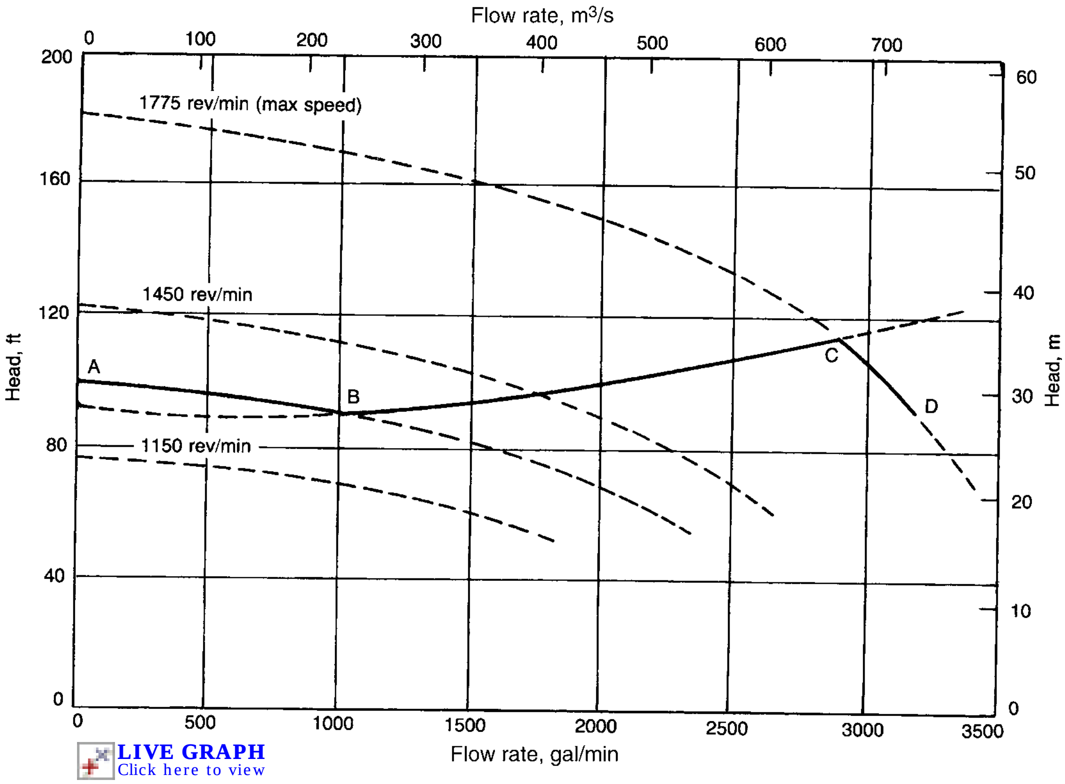


Figure 15-19. Booster pump operation at minimum suction head.

shown in Figure 15-19. For example, at a flow rate of 114 m^3/h (500 gal/min), the pump would develop a head of 29 m (96 ft). If the minimum demand is less than the minimum recommended pump flow rate, the pumps should be protected by one of the three methods discussed below.

Suction Pressure Varies with Flow Rate

Flowmeter and Bypass

Include a flowmeter and a bypass with a modulating valve, as shown schematically in Figure 15-20a. The flowmeter can be any type that has a 4- to 20-mA output signal. Accuracy is not usually important. Because the valve should modulate in response to the flowmeter output signal, a hydraulically operated valve is usually preferred.

The valve actuator is adjusted to maintain the recommended minimum pump demand rate (RMPDR) through the valve when the demand is zero. The valve should be fully closed when the demand exceeds the RMPDR by 25 to 35%. At demands less than the RMPDR, the valve modulates to maintain the flow through the flowmeter equal to or slightly greater than the RMPDR. If two pumps are to operate in parallel, as shown in Figure 15-20b, the signal from the flowmeter should be electronically converted to cause the valve to maintain a minimum combined demand rate of twice the RMPDR when both pumps are operating. This method of operation, however, applies only to load-sharing parallel operation, as explained below.

A bypass around the flowmeter (not shown in Figure 15-20) is desirable so that the meter can be removed for repairs (see Figure 20-12).

The modulating or regulating valves shown in Figure 15-20 would nearly always operate as throttling valves subject to cavitation. The valve selected must be capable of withstanding such severe service for many years. It should be of the type in which metering occurs downstream from the seat, and the cavitation constant should be greater than 1.0. If metering occurs in the seat (as in an ordinary globe valve), the seat would be quickly cut. If the pressure drop is excessive for a single valve, two valves can be installed in series.

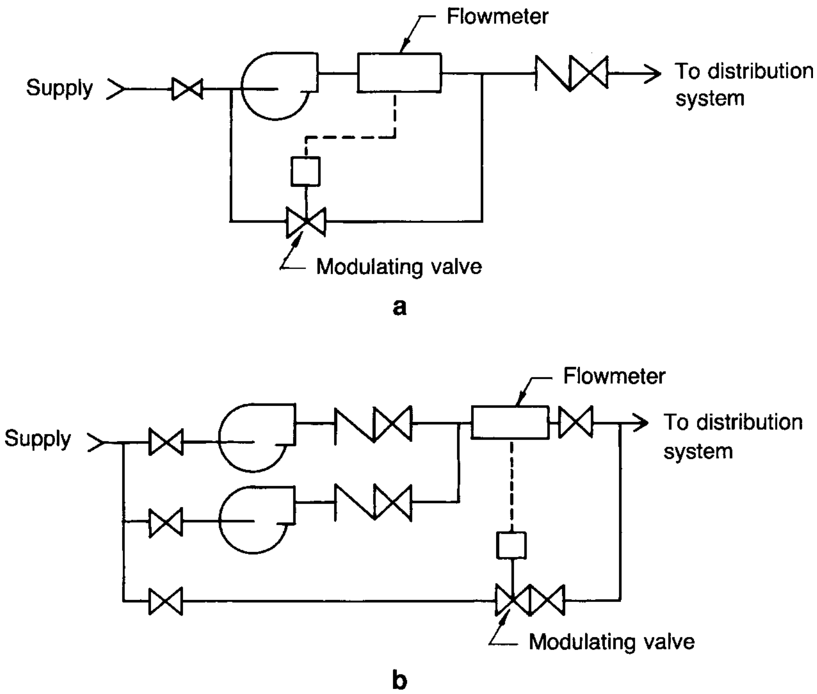


Figure 15-20. Flowmeter and pump bypass. (a) Single pump; (b) pumps in parallel.

Small Pumps

Include a relatively small, flat-curve, constant-speed pump (sometimes called a jockey) to operate during periods of low demand. The V/S pumps do not operate while the small pump is running. Because the head in the pump suction header is high when the flow rate is low, the small pump can be selected for a relatively low head.

The small pump is usually started and stopped by means of pressure switches with suitable time delays to prevent starting and stopping on pressure surges. The switches sense pressure in the station suction header. If the suction pressure variation at low flows is too little for reliable pressure switch operation, it may be necessary to use a flow-measuring device to start and stop the small pump.

Hydropneumatic Tank

A small hydropneumatic tank with an air compressor and an appropriate air/water ratio controller can be used with C/S pumps. It is rarely used with V/S pumps.

15-9. Operations in Booster Pumping

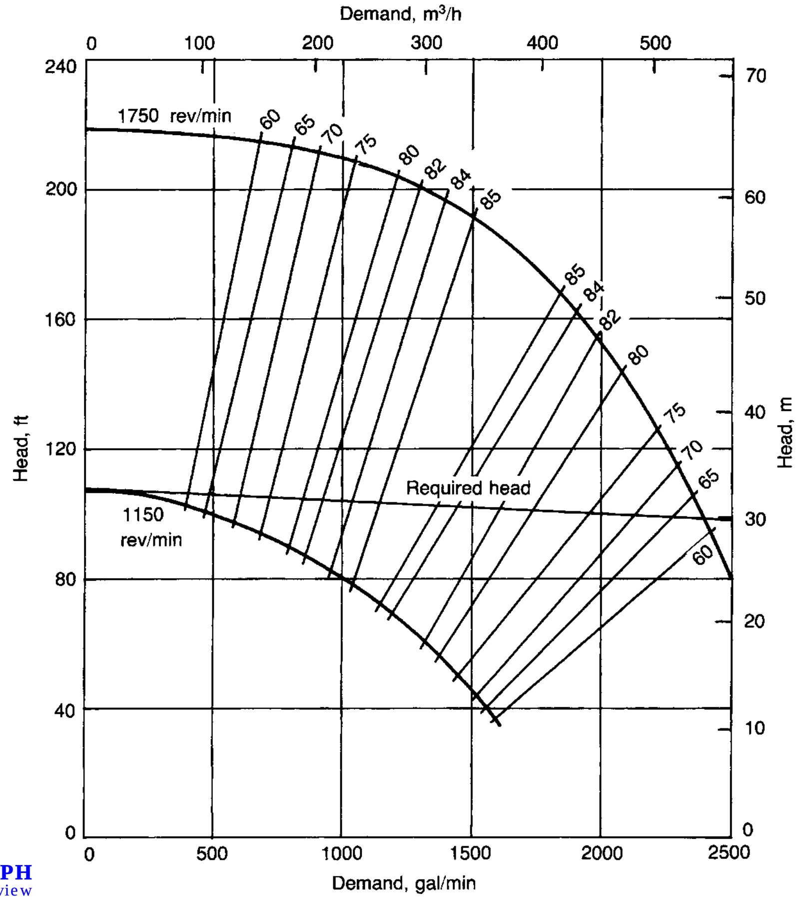
Booster pumps can be operated in parallel in much the same way as described in Section 15-4, except that the difference in power requirement is even greater. Typical, steep characteristic curves of a booster pump are shown in Figure 15-21 with the required head curve superimposed. When two of these pumps are operated in parallel, the total power required in both types of operation is as shown in Figure 15-22.

With a different combination of pump and/or required head curves, the difference in power requirements may be more or less pronounced than in this example, but a situation in which staggered operation would use only a little more power than load-sharing throughout a significant portion of the discharge would be most unusual.

Sequencing

Starting Lag Pumps

A high-quality, mechanical pressure switch that senses pressure in the discharge header could be



LIVE GRAPH
Click here to view

Figure 15-21. Characteristic curves for a booster pump. Efficiency lines are shown straight for convenient (although approximate) interpolation.

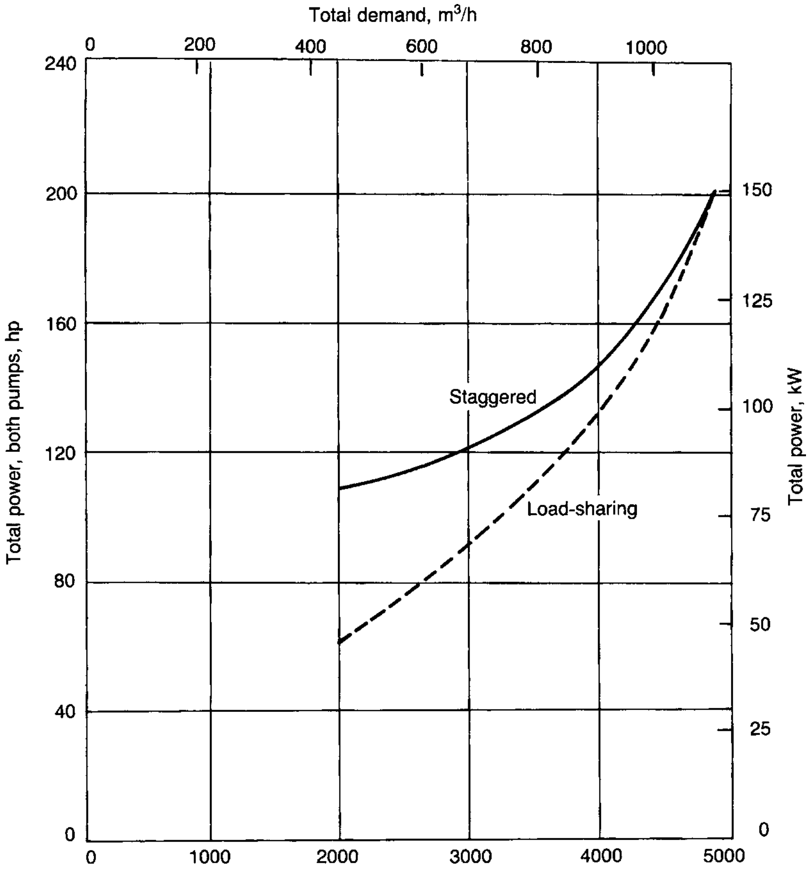
used to signal the second pump to start but only if the pump curve is steep. For starting either (1) flat-curve second pumps or (2) third (and subsequent) flat- or steep-curve pumps, the recommended method is to use signals from a flowmeter and a pressure switch. The lag pumps are normally started either (1) when the header flow rate approaches the maximum flow rate of the on-line pumps, *or* (2) when the pressure in the discharge header is low. Time delays should be included in pressure switch circuits to prevent action on momentary pressure fluctuations.

Pressure switches alone should not be used to activate starting signals for flat-curve pumps or pumps subsequent to the second pump, because the pumps on line may cavitate before the pressure in the discharge header falls enough (about 10% below the desired head) to actuate the pressure switch.

Stopping Lag Pumps

The preferred method of stopping lag pumps is always by means of signals from a flowmeter plus an electronic pressure switch. Each lag pump is stopped when (1) the demand falls to within the combined capacities of the remaining on-line pumps, and (2) the pressure in the discharge header is at the desired value.

A mechanical pressure switch may be used to stop the second pump when the pumps are operated in the staggered mode, but only if the pump curves are steep (as in Figure 15-21). Pressure switches cannot be used to stop lag pumps in a load-sharing-mode operation because the pressure in the discharge header remains relatively constant at all flow rates within the combined capacities of all on-line pumps.



LIVE GRAPH
Click here to view

Figure 15-22. Power consumption in staggered and load-sharing operation.

For pumps operating in a load-sharing mode, the lag pump can be stopped by means of a running period timer. One timer is provided for each step of pumping capacity. A timer is actuated when its associated pump is started, and it runs for a preselected period of time. The associated pump is stopped at the end of the timer cycle, but if the demand still requires operation of the pump, the pump motor is restarted and operates through another timer cycle.

Mechanical pressure switches do not actuate or deactuate at the same pressure on every cycle. The band of actuation/deactuation may be ±5% of the set-point pressure. Pressure changes must be sufficient to reset the switch mechanism. For example, if a pressure switch actuates at 90 on a falling pressure signal, the switch may not deactuate until the pressure rises to 99. This characteristic is called the "differential." The differential span may drift with respect to the set point as the switch linkage wears.

15-10. Simultaneous Operation of V/S and C/S Booster Pumps

Combinations of V/S and C/S pumps are rarely used for water booster pumping. Such combinations could be used if *all* of the following conditions are met.

- The pump curves are steep.
- The capacity relationships are those discussed in Section 15-5. The demand does not change significantly over brief periods of time, so that rapid cycling of the C/S pump(s) is precluded.

Pumping Unit Failure Detection

A pressure switch connected directly between the pump discharge nozzle and the check valve can be used if the check valves do not have outside levers.

Pressure switches are useful and can detect all internal pumping unit failures, but they cannot detect whether flow is occurring. Check valve limit switches are always preferred to pressure switches.

Other Considerations

The special design considerations discussed in Section 15-6 apply equally well to booster pumping.

15-11. Adjustable- and Variable-Speed Drives

The purpose of V/S drives is to regulate the flow rate, and various means for so doing are listed in Table 15-2. The last three types of V/S drives are those most commonly used for water and wastewater pumping. Because some other drives are common in other industries, a limited discussion of a few of them is also included. See Karrasik et al. [1] for discussions of other V/S drives.

A summary of the comparative advantages of the various types of V/S drives is given in Table 15-3. Descriptions and further details are given below.

Direct Engine Drives

Internal combustion engines (discussed in Chapter 14) are easily adapted for V/S pumping. All that is needed is a proportional linkage (mechanical, pneumatic, or electrical) between the sump level or a header pressure monitor and the governor. Note that prolonged operation at low loads causes carbon deposits in a diesel engine exhaust system (see Section 14-22).

Right-angle gears can be used to adapt the horizontally mounted engine to a vertical pump shaft. Nonreversing gears should be used. The connection between the drive shaft and a horizontal pump shaft should be flexible to limit vibration transfer.

Engine drives are ideally suited for the following:

- Pumping stations that are a problem for electrical facilities because of size or location. It may be more economical to use engines than to install long, large transmission power lines.
- In some water hammer situations where the required inertia of moving parts is greater than is economically feasible for electric motor drives.
- Locations where gaseous fuels are available at low cost—near natural gas centers or large wastewater treatment plants with surplus digester gas.

Engine-Generator Drives

If power must be locally generated by an engine, V/S pumping can be had for almost no added expense by controlling the generator speed with an asynchronous engine governor that responds to changes in the controlled variable (such as sump level or discharge pressure) to produce a variable-frequency power source.

Table 15-2. Systems for Regulating Flow Rate or Pressure

Motive unit speed	Pump speed	Regulating system	Notes and examples
Constant	Constant	Throttling valve	Pump may run near zero discharge; for water pumping only.
Constant	Constant	Regulating valve	Excess pump discharge spilled to sump intake; for water pumping only.
Constant	Constant	Variable-pitch propeller	Propeller pumps for pumping very large storm water flows.
Constant	Constant	Intermittent operation and usually multiple pumps in parallel	Storage in wet well or clear well; variable number of duty pumps.
Constant	Variable	Mechanical drive	Oil film cone rollers, automatic gears, vee belt.
Constant	Variable	Hydraulic coupling	Hydrokinetic, hydroviscous.
Constant	Variable	Hydrostatic drive	Hydraulic pump and hydraulic motor with squirrel-cage motor.
Variable	Variable	Slip energy recovery	Wound-rotor motor.
Variable	Variable	Slip drive	Secondary rheostat with wound-rotor motor.
Variable	Variable	Engine drive	Throttle fuel.
Adjustable	Variable	Electronic	Adjustable-frequency drive.
Constant	Variable	Slip drive	Eddy-current coupling.

Table 15-3. Comparison of Variable-Speed Drives

Drive	Advantages	Disadvantages
Direct engine drive	Very reliable when well-maintained.	Engines expensive. Maintenance costs can be either high or moderate (depends on personnel).
Combination engine/ motor drive	Utmost in reliability. Allows peak shaving to avoid electricity cost penalties (time of day, load factor, power demand). Emergency operation without costly switchgear, cabling, and generator. Engine can drive pump at higher speeds than motor if pump is properly selected; greater capacity for storms, fire flow. Only a small engine-generator is needed for lighting, control, and—possibly—ventilation.	Practical, economical power range per engine is 75 to 200 hp. Speed range less than 1200 rpm. Multiple engines are expensive. Dedicated to one pump.
Adjustable-frequency drives (AFDs)	Very popular; nearly all new V/S drives are AF drives (AFDs). Depending on motor and AF converter selected, peak wire to drive shaft efficiency ranges from 81 to 92% or more without a transformer and about 2% less with a transformer; efficiencies at lower speeds reasonably high. Efficiency is misleading; at reduced speeds, calculate power losses based on actual operating conditions; use life-cycle costs (see Example 29-1) for comparison with other drives. No motor starter needed unless full-speed bypass operation is required. Starting current is limited to provide "soft start" and help reduce the required size of standby engine-generator Acceleration and deceleration rates and their duration are adjustable. Careful selection of these parameters can avoid hydraulic surge and water hammer problems. High reliability if provided with adequate protection from harmonics and electrical surges. Electronic components are getting more compact, more efficient, and less expensive, leading to greater application and lower life-cycle costs. Air conditioning requirements are diminishing. Any practical maximum speed obtainable. Can produce frequencies above 60 Hz. Specific speed avoidance circuitry jumps over the drive's natural frequencies that prevent loads from maintaining critical speeds that might cause pump and shaft vibration.	Motors should be carefully selected for use with any AF drive. For new AFD service, motors should be "inverter-duty" (i.e., specifically built with superior insulation and resistance to temperature rise; see Chapter 13). For existing installations, standard motors should not be used with AF drives without detailed investigation. To be satisfactory for AFD service, motors must be built with high-quality insulation and resistance to heating or be suitably derated (see Chapter 13). Generates electrical noise in the form of harmonics, both returning to the power source and out to the drive motor. As a minimum, line side series reactors are recommended to reduce harmonic currents fed back into power supply system. 18-pulse AFDs are recommended for larger drives (50 hp and above) because of the theoretical elimination of the most predominant harmonics (5^{th} through 13^{th}). Where numerous small 6-pulse AFDs are used, an active filter may be required to control harmonics adequately and to meet utility requirements (see also Section 9–11). Susceptible to damage caused by lightning or heavy power surges. Depending on lightning exposure, extensive lightning protection systems may be required. In all situations, add surge arrestors on incoming power. Factory representatives, who may not be readily available in all geographical areas, usually perform adjustment and repair.

Table 15-3. Continued

Drive	Advantages	Disadvantages
	After a power interruption, "catch a spinning load" or "flying start" circuitry can detect direction and speed of a coasting motor and restore it to the correct direction and set-point speed.	For remote locations, specify one spare power module and one spare controller module. Owner can usually install either and send faulty unit to factory for repair. For owners with technicians trained in electronic maintenance, specify a comprehensive set of spare parts. Life expectancy: 8–15 years. Some may shut down frequently with minor power disturbances, so specify automatic "flying restart/catch a spinning load" option. Motor efficiency reduced about 5% by the nonsinusoidal wave power input in older drives. The best new drives reduce efficiency by only as much as about 3%. Higher harmonic contents cause higher heating effect on the motor. Standby generators must be designed to accommodate harmonic load and may need to be oversized to limit temperature rise. Canadian Standards Association and California Safety Order codes may require features not included in standard AFDs that further increase cost.
Electromagnetic eddy-current coupling (a slip drive)	Depending on motor and coupling selected, peak wire to drive shaft efficiency ranges from 83 to 93%; as speed is reduced, efficiencies drop rapidly. Efficiency is misleading; at reduced speeds calculate power losses based on actual operating conditions; use life-cycle costs (see Example 29-1) for comparison with other drives. Uses squirrel-cage induction or synchronous motor Can be driven by any rotating prime mover, including squirrel-cage induction or synchronous motor, or internal combustion motor up to 3600 rev/min. Can be retrofitted onto any existing motor/engine. Creates no harmonics. Insensitive to power quality. Unlimited number of starts since motor need not shut down. Little maintenance: lubricate every 2–3 months, replace brushes, and dress slip rings. Proven life of 20 yr or more. Reliable; relatively low maintenance. Used for many years for variable speed control. Sizes available are 4–2200 kW (5–3000 hp).	Maximum speed about 98% of motor nameplate speed. Slip loss, dissipated as heat to station atmosphere (in air-cooled models), may require increased building ventilation. Application restricted to loads that follow the affinity laws (e.g., centrifugal pumps). May require reduced-voltage starters for starting large motors. Motors may require power-factor correction capacitors. Noisy with motors of 1200 or more rev/min. Replace brushes and dress slip rings every 2–4 years (may require dismantling). Horizontally placed units increase the floor area needed; vertically mounted units may require extra headroom and/or special provisions to resist lateral seismic forces. Becoming obsolete. Fewer suppliers result in higher first cost than competing drives such as AFDs. Rejected slip heat must be accommodated. In air-cooled units, it is rejected to the local air, but can be ducted in some installations. In water-cooled units, the heat can be captured or dissipated, depending on the installation. Air-cooled units can be noisy, due to heat dissipation fin noise.

Continued

Table 15-3. Continued

Drive	Advantages	Disadvantages
	Longest mean time between failures and shortest mean time for repairs in most places. Simple design; most motor rewind shops can repair. Brushes and slip rings are eliminated in the stationary field design; maintenance is reduced to that of a squirrel-cage induction motor.	Larger systems (over 500 hp) require external liquid–liquid or liquid–heat exchanger to remove slip-loss heat. Output speed decreases as load increases. Precise control requires closed-loop feedback circuit. Shaft-mounted units impose an overhung weight on the motor and pump shafts. A check for critical (resonant) frequency within the operating speed range must be performed.
Permanent magnet eddy-current drive (a slip drive)	Same advantages as electro-magnetic eddy-current drive with the following additions/exceptions: Efficiency is up to 10% higher than an electromagnetic eddy-current coupling and comparable with AFDs at speeds above 90%, primarily because of the absence of an external magnetizing field generator. Simple design. No windings. Easy to work on and troubleshoot. Actuation is provided by standard industrial actuators and may be electric, pneumatic, or hydraulic. Process control for pressure, flow, level, etc., can be provided when the actuator is interfaced to a standard industrial set-point controller. These are the only electronic or electrical elements in the system. Little maintenance: lubricate every 2–3 months. Bearings and all wearing parts are designed to provide a 5-yr L-10 life. No brushes to replace. Permanent magnets have a half-life of 20,000 years. Sizes from 5 to 1200 hp (4 to 900 kW). Available in shaft-mounted or foot-mounted configurations, vertical or horizontal. Can be used on higher-voltage motors without modification.	Same disadvantages as electro-magnetic eddy-current drive with the following additions/exceptions: Single source supplier [2]. As the relationship of the distance apart of the opposing magnetic poles and the magnetic forces of attraction is cubed, a cam is necessary on the actuator arm to provide a linear response to a control signal. A closed-loop feedback control is required for accurate response. Because output speed decreases with an increase in load, speed matching of multiple units is virtually unattainable. Noisy with motors of 3600 rev/min. Needs guard to prevent contact with rotating parts.
Fluid coupling (hydrokinetic) drive	Rugged machinery. Basic elements subject to little wear. Moderate cost.	For remote control, the scoop tube is usually driven by a small electric motor, which must have a "dead zone" that results in continuous hunting over a small range in water and wastewater pumping applications and causes occasional failure of scoop tube motor and rapid wear of linkage. External heat exchangers are required. Reduced-voltage starters may be required. Low efficiency, about 5–7% less than eddy-current coupling at all speeds due to higher "windage" losses.

Table 15-3. Continued

Drive	Advantages	Disadvantages
Slip recovery drive	Efficiency comparable to AFD. Can be closely sized to fit application. Reliable. May be more economical than AFD in ratings above 750 kW (1000 hp).	Fewer available manufacturers results in higher first cost than competing drives such as AFDs. Have become obsolete in the lower hp sizes. Solid-state components have caused maintenance problems. Unsuitable for less than 375 kW (500 hp). Requires wound-rotor induction motor. Wound-rotor induction motors are expensive and have been discontinued by many manufacturers. Availability of spare parts a problem in some areas. Poor power factor; requires large bank of capacitors for power factor correction. Repairs must be made by factory representative. Replace motor brushes and dress slip rings every 2–4 years. Difficult to troubleshoot. Life expectancy: 10–15 years.
Variable-ratio belt drive	High efficiency. Good reliability. Excellent for sludge pumping or other applications where speed adjustments are made manually and output speeds are in low ranges.	Ratings limited to 100 kW (125 hp). Most output speed ranges unsuitable for pumping wastewater or water. Unsuitable for continuously fluctuating speed or for frequent starts and stops (causes rapid wear of belt). Chrome plating helps. Belt replacement may require removal and disassembly; check with manufacturer. Cost: low from fractional to about 5.6 kW ($7\frac{1}{2}$ hp); moderate from 7.5 to 19 kW (10 to 25 hp); high from 22 to 100 kW (30 to 125 hp).
Hydrostatic drive	Motor can be submerged indefinitely. Explosion-proof; useful in hazardous atmospheres. Useful for constant-torque sludge pumps. Widely used in industry.	Limited power and output speed ranges. Pump *very* noisy. Overall efficiency of about 80% *at all speeds;* requires larger than normal electric motor. May require oil cooler. High cost.

One or more pump motors can be powered by one generator in load-sharing operation. With a back-up engine-generator set (which is usually required anyway), this system is very reliable.

Combination Drives

If the utmost in reliability is required of the standby power system, consider using direct-drive standby engines for duty and standby pumps instead of an engine-generator set. For horizontal, split-case pumps, the motor is mounted at one end of the pump shaft and the engine is mounted through an automatic clutch at the other end. If a vertical motor drives the pump, a combination right-angle gear and automatic clutch is available to provide a horizontal shaft for an engine and a mount for a vertical motor.

If power fails, the duty engine is started and the motor is allowed to spin freely. An interlock must disconnect the motor from the power source when the engine is started. When electric power is restored, the pump must first slow to well below synchronous speed and the engine must be disconnected before the motor is energized. Energizing a spinning motor might not harm it (unless it is spinning in reverse), but energizing a spinning motor with its residual field remaining from interrupted electrical power by connecting it to another, nonsynchronized power source

could displace the windings, break the shaft, and probably shear the coupling bolts.

Peak Shaving

Significant savings in power costs can sometimes be obtained by using the motor to drive the pump for low and medium flows and by switching to engine drive for high flows. The advantages include:

- A smaller electric motor
- Two or three small- to medium-sized engines and drive gears (plus a very small engine-generator set for controls and lighting) replace a large engine-generator set and expensive electrical transfer switch
- A lower electric power demand
- The engine need not be “exercised” if high flows occur with sufficient frequency
- The engine always operates at or near full load.

The design of a peak shaving system requires an extensive study of diurnal and monthly flow rates. A convenient and simple aid is the use of either arithmetic or log probability paper [3]. Fair *et al* [4] have explained the use of probability paper.

Adjustable-Frequency Drives

The adjustable-frequency drive (AFD) consists of an adjustable-frequency (AF) converter (Figure 15-23) and an ac squirrel-cage induction motor. The AF converter is a solid-state electronic assembly that contains:

Figure 15-23. An adjustable-frequency converter for a 300-hp motor. Courtesy of Robicon Corp.

- A rectifier, which converts incoming ac power to dc power
- A filter to reduce the ripple in the dc power
- An inverter, which changes dc power to ac power at any selected frequency within the design range.

A control signal from external equipment is supplied to cause the inverter output frequency and voltage to vary as a function of some convenient variable, such as pressure for water pumping or wet well liquid level for wastewater pumping. The ac output of the converter powers an ac squirrel-cage induction motor, and the motor speed varies directly and linearly with the frequency of the applied power.

Almost all modern AF converters are pulse-width-modulated (PWM) utilizing insulated gate bipolar transistors (IGBTs) in the output stage, which produce the output waveform shown in Figure 15-24c. Large (over 450 kW or 600 hp at this time) AF converters may use older thyristor designs that produce a stair-step output or pulse-amplitude-modulated (PAM) output (Figure 15-24a). The IGBT-type converter is less expensive than older designs and produces an output waveform that does not heat the motor as much as PAM output. The disadvantages of the IGBT-type are: (1) a very high switching frequency that results in a high rate of change of voltage (*dV/dt*), and (2) high peak voltages applied to the motor windings. Older-type converters were frequently designated as either (1) voltage source, or (2) current source. Voltage-source converters (Figure 15-24a) have been replaced by PWM converters for drives to 450 kW (600 hp). The current-source type (Figure 15-24b) still has adherents, is generally more expensive than voltage source for a given horsepower, and is usually used only for very large drives. As solid-state power technology advances, hybrid drives are now emerging. One manufacturer makes a medium-voltage AF converter using current-source topology with a PWM rectifier front end and a matching inverter (i.e., called an "active front end." This active front end drive is claimed to produce near-sinusoidal output under all load conditions and meets IEEE 519 limitations on harmonic distortion regardless of system impedance.

Current-source converters must be closely matched to their load, but voltage-source converters can operate any size motor, even multiple motors, as long as the power requirement is within their rating. For drives over 600 kW (800 hp), consider the medium-voltage active front end AF converter or the load-commutated converter, which either uses a synchronous motor or a reactive output network to accomplish commutation. Load-commutated converters are particularly suitable for motors over 1500 kW (2000 hp).

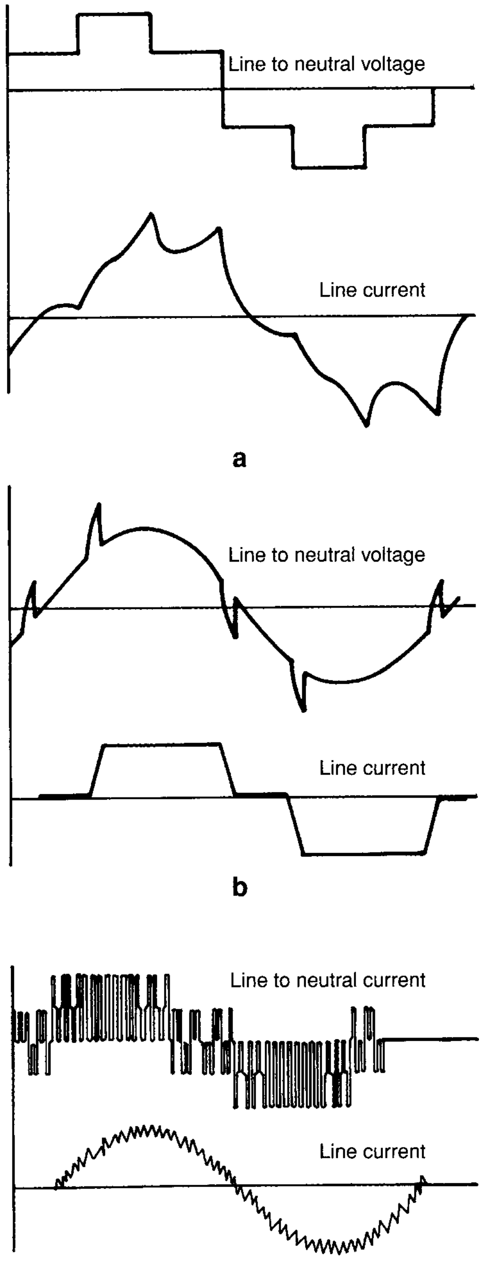


Figure 15-24. Output waveforms from adjustable-frequency converters. (a) Voltage source six-step; (b) current source; (c) pulse-width modulated waveforms. Courtesy of Eaton Electric Drives Divisions.

Harmonics

All AFDs draw 5^{th}, 7^{th}, 11^{th}, and 13^{th} order harmonic currents (300, 420, 660, 780 Hz, etc.) from a 60-Hz

power source. These harmonics distort the voltage supply and cause (1) excessive heating of voltage supply equipment such as transformers, (2) excessive heating or malfunction of other equipment connected to the voltage supply, and (3) interference with communication systems. Voltage distortion must be controlled to prevent difficulty with other plant equipment and the AF drives themselves. Harmonic currents must be controlled to a level acceptable to the power and telephone utilities. Generally, this level of control requires conformance to the requirements of IEEE 519, which limits both the harmonic current load and the amount of voltage distortion permitted (see Section 9-11).

Prospective harmonic distortion produced by AF converters can be readily predicted by power system analysts who use computer programs to simulate the behavior of the entire electrical system. Such an analysis should be performed on any power system where the converter load exceeds 25% of the total load. Power system analysts can prescribe various corrective measures to bring the installation within the requirements of IEEE 519. Typical corrective measures (in order of increasing cost) include:

- Series reactors
- Phase-shifting transformer and six-phase rectification
- Shunt filters
- Active filters.

These corrective measures increase the drive cost and may decrease the overall drive efficiency. Because of the space required for a typical filter, the need for filters *must* be determined early in the design of the building envelope. Harmonics also affect the drive motor. Output harmonics and their effects are discussed in Chapter 13.

Efficiency

The efficiency of a modern AF converter may approach 98% at full load and speed, but the effective efficiency is lower because harmonics increase losses in the motor and the power supply system, and harmonic mitigation schemes absorb energy.

A series reactor reduces drive efficiency by about 1%. A phase-shifting transformer reduces drive efficiency by 2 to 3%. Series reactors and phase-shifting transformers usually remain connected to the power supply even when the drive is not running. Their efficiency loss, though small at no load, can have a significant effect on overall drive efficiency. Shunt filters absorb little power, but like series reactors and phase-shifting transformers, these units are frequently left connected to the power supply when the drive is not running, and they cause a significant energy loss. PWM drives have little effect on motor efficiency (less than 1%), but older drives with step-output waveforms reduce motor efficiency by 2 to 3%. Practically, AF drive system efficiency at full load cannot be quantified, but probably never exceeds 96% for PWM drives and 92% for step-output drives.

Efficiencies of AFDs vary as a function of both load and speed. The published efficiency data for AFDs driving centrifugal pumps are based on the assumptions that (1) the drive is fully loaded at maximum speed, (2) power varies as the cube of pump speed, (3) the motor efficiency at full speed is about 96%, and (4) the drive does not include transformers. Because (1) is usually incorrect, (2) is almost always incorrect, (3) is frequently incorrect, and (4) is often incorrect for wastewater pumping applications, the efficiencies of AFDs are usually significantly less than the published values of efficiency versus speed.

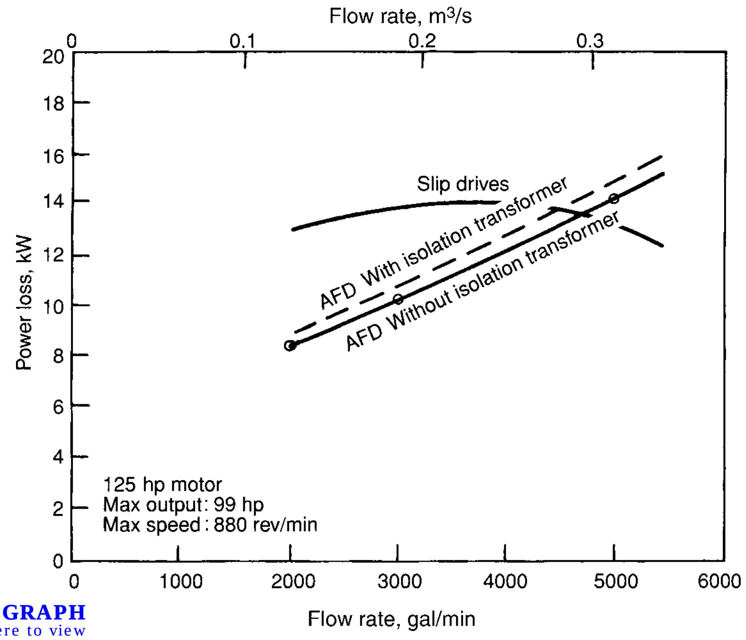
Efficiencies of electric motors fall rapidly at loads below about 25% full load. For this reason, the efficiencies of *any* variable-speed drive that includes an electric motor is very poor at less than 20 to 25% of rated loads.

Note that efficiency is a misleading indication of operating costs. Low efficiency at low discharge is relatively unimportant because the power required is also low, and it is the power losses that are important. Typical calculated power losses in a complete driver system (AF converter and motor) for the 74-kW (99-hp) pump of Figure 15-4 are shown in Figure 15-25, where they can be compared with power losses of typical slip drives such as eddy-current couplings and wound-rotor motor with secondary rheostats.

If dynamometer tests are specified, require that they be made at the loads required by the pump at several typical speeds and with instrumentation that maintains its accuracy in spite of any harmonics generated by the converter. If practical, insist that tests of the motor and the AF converter to be installed in the field be done together. It is even better, although expensive, to test the pump, motor, and AF converter as a unit at the pump manufacturer's test facility.

Operating Costs

The power required by the pump should always be calculated as a function of discharge, head, and pump efficiency and not by application (or, rather, misapplication) of the affinity laws. Affinity laws are too easy to misuse, and they are inaccurate when applied to pump efficiency. Overall (or wire-to-water) effi-



LIVE GRAPH Click here to view

Figure 15-25. Wire to pump shaft power losses for pump and system curves shown in Figure 15-4.

ciency is thus obtained by calculating and plotting the following:

- Water power (kilowatts or horsepower) based on head and flow rate over the expected range of pump speeds. Calculate the actual pump shaft input power (bhp) based on pump efficiencies.
- Motor efficiencies reduced for (1) low speeds, (2) low torques, and (3) nonsine-wave input (about 3%).
- Converter and transformer efficiencies for the actual loads.
- Overall drive efficiency at each of several speeds obtained as the product of pump, transformer, converter, and predicted motor efficiencies.
- Wire power (input power to the motor) obtained by dividing the bhp by the overall drive efficiency.
- Power needs calculated from hours per year of operation at various discharge and pump speeds (see Example 29-1).

Compare various types of drives for each specific application on the basis of the present worth of cost, life and replacement, maintenance, and power (per Example 29-1). The results cannot be predicted reliably if any of these factors is ignored.

Motors

Standard motors should not be used with AF drives without due consideration. For discussion of these considerations, refer to Chapter 13. All drive types impress a nonsinusoidal waveform on the motor. As motor speed decreases, motor cooling provided by shaft-driven fans decreases, but cooling is not the only consideration when applying adjustable-frequency drives. Rapid change in voltage versus time (*dV/dt*) caused by some types of drives has been the source of rapid insulation failure in some motors. Furthermore, stray currents, apparently induced by some variable-frequency drives, have resulted in damage to the bearings and other components. To break the circuit of these stray currents, a thin, dielectric material can be specified for electrically isolating the shaft from the motor frame. Not all problems associated with this technology have been solved, but consensus standards for motors for application to AFDs are evolving.

The claim frequently made by drive manufacturers that standard motors can be used without restriction *is not true*. As it happens, AF converter application to existing motors is frequently successful, because motors are often over-sized for the application, and older motors were frequently designed more conservatively than are modern motors.

Inverter-Duty Motors

Motors for use with AF converters must be specifically designed for such service to fulfill their expected service life. For example, such motors (called "inverter-

duty motors") have stator wires double-dipped in varnish (wires in standard motors are single-dipped); thin, paper-like insulation is added in the slots; and Class H or Class F insulation is used for Class B temperature rise. The bearings are insulated from the motor frame to interrupt stray currents and prevent electrolytic corrosion. An option is grounding the rotor with a brush connected to the frame. Inverter-duty motors are about 15% more expensive than standard motors, but they are more rugged and last much longer. For that reason, it is often wise to specify them for solid-state soft starters or even for across-the-line starters. As always, proper grounding of the motor frame and AF converter enclosure is essential. *Be very conservative in specifying motors for use with AFDs.*

In addition to the usual motor parameters that must be specified when ordering an inverter duty motor, the speed range and load speed/torque characteristic must be specified. The motor manufacturer will then design the motor cooling for the application. Motor manufacturers catalog their motors for speed ranges as great as 10:1 when driving constant torque loads—a much more severe service than driving a centrifugal pump over a typical 2:1 speed range and a load that varies geometrically with speed.

Converters

Step-output converters and PWM converters that do not use IGBT converters are generally satisfactory with existing 1.15 service factor motors driving centrifugal pumps if the motor is not loaded above the nameplate rating (service factor not used). However, both the foregoing AF converters are less efficient and more expensive than modern converters.

IGBT converters switch their output at a very high frequency (Figure 15-24b). They impress a high rate of voltage change (*dV/dt*), and the switching action also overshoots, thereby impressing high peak voltages on the motor. Both characteristics put greater stress on the motor insulation than does a sinusoidal voltage. The motor insulation system must either be designed for this stress, or other preventive measures must be taken. Refer to Chapter 13. Definite-purpose inverter-duty motors are preferred on all new AFD installations. Always locate the AF converter as close to the motor as possible. Long motor cable runs (>45 m or 150 ft) magnify the peak voltages produced by the AF converter. Output reactors are recommended on all AFD installations employing IGBT in the converter.

Operational Problems

Any short interruption of power (sufficient to make a light flicker) can trip an older AFD, after which the motor must be allowed to come to rest before restarting—a requirement annoying in thunderstorms and windstorms, which could cause several interruptions per day. Most modern AFDs, however, are designed to operate through such momentary (less than 0.5 s) power losses, and AFDs can be designed to synchronize with a spinning motor (this feature is called "catch a spinning load" or "flying restart"), but these features must be specified if specifically wanted. They increase the cost of the AFD only slightly, if at all.

Designing and/or specifying the interfacing between the motor and AF converter is not a simple task. One way to promote compatibility is to specify "unit responsibility," in which the pump, motor, converter, and isolating transformer (if any) are all furnished by one supplier (the pump manufacturer) who has full responsibility for the proper operation of the entire pumping unit (see Chapter 16). The pump manufacturer is in no position to warrant the compatibility of the motor and the AF converter, so specify that the entire power package be supplied to the pump manufacturer by the AFD manufacturer, who must ensure the compatibility of the motor and the AF converter.

Because there are torsional pulsations in the output of a motor powered by an AF converter, it is strongly recommended that a torsional analysis be done for any drive of more than 300 kW (400 hp). Vertical motors supplied with variable-speed systems are subject to "reed frequency" vibration. The motor, the pump, and the control manufacturers must coordinate to eliminate this potential problem (see "Unit Responsibility" in Appendix C). Certain speeds may have to be blocked out of the variable-speed range. These speeds must, of course, be passed through during drive acceleration and deceleration, but the drive must not be allowed to linger there. This feature is often standard in new AFDs and is known as critical frequency avoidance. A second problem that can occur if there is an anti-reversing ratchet in the drive shaft is that dwelling at certain speeds can cause an oscillation about the vertical axis, hammer the ratchet, and damage or destroy it.

Adjustable-Frequency Drive Specifications

A summary of specifications and requirements for a reliable, trouble-free AFD system includes:

- *A power system analysis*. A source-voltage range of +10%, −5% from nominal (usually 460 V) is

specified in NEMA standards for AF drives. Most electrical equipment is designed for +10% from nominal. If the plant electrical system cannot meet the tighter tolerance required by AF drives, specify the voltage range that the drive must accept.

- *Voltage disturbances*. Modern drives have microprocessor controllers. These controllers are capable of maintaining drive operation through short power supply voltage sags and automatically restarting the drive if the sag exceeds the hold-up interval. Specify "flying restart."
- *Grounding*. Protect the microprocessors by specifying a ground grid in which each electrode, when isolated, has a maximum ground resistance of 2 ohms—a protection significantly better than that specified in the NEC.
- *Motor insulation and AFD compatibility*. Static power converters impose unusual stress on winding insulation. Early insulation failure will occur if the motor is not designed for this duty. For AFD duty, specify inverter-duty motors. See Chapter 13.
- *Drive tuning*. To ensure optimum performance, the AFD must be tuned to the motor and load characteristics. Some modern drives have a "self-tuning" function that permits the drive to adjust some parameters to motor characteristics, but no drive can analyze load requirements. Specify that the drive be set up by a factory-authorized technician. Do not rely on contractor electricians to set up AF controllers.
- *Voltage-vector control*. Vector control for standard pumping applications is neither necessary nor recommended for pumping stations. For best performance in other applications, however, specify drives with voltage-vector control to regulate both voltage and frequency to produce a sine wave output for optimum motor magnetization and performance, but the added cost is not worthwhile for pump drivers.
- *Isolation*. A contactor to disconnect a drive physically from the motor is not necessary, but consider specifying contactors to isolate shunt filters when the drive is not running. If the drive is equipped with a phase-shifting transformer, specify a line contactor to eliminate the transformer losses when the drive is not running. Input isolation has the benefit of isolating the AFD from line disturbances and lightning damage when the drive is not in use (or on standby).
- *Harmonic currents*. Power factor correction is not required for PWM-type AF drives. Power factor correction is not recommended on the other types of AFDs. If power factor correction is required because other loads on a power system have a lower power factor, filter traps must be added to keep harmonic currents out of the capacitors. A power system analyst can design suitable trapped capacitor schemes for power systems with AF drives.
- *A temperature-controlled environment*. A maximum ambient temperature of 40°C (104°F) is specified in NEMA standards. Unless the space is air conditioned, the temperature requirement may not be met. AF drives generate substantial heat. Specify a realistic ambient temperature requirement. If the space is to be air conditioned (mandatory in many geographic areas), obtain realistic heat dissipation (including heat from transformers) values for the drive from the manufacturer, specify these as maximums, and transmit the value to the air conditioning system designer.
- *Standby generators*. If the drive must be powered by a standby generator, harmonic currents drawn by the drive must be reduced to a minimum or the generator may need to be derated. Heavy harmonic filtration with shunt filters may result in a leading power factor. Use 18-phase rectification, series input reactors, and/or active filters in preference to shunt filters to reduce harmonics. Be sure the generator is capable of handling the harmonic current load and any leading power factor resulting from any filters. Be aware that the generator must be oversized to meet these requirements.
- *Atmosphere*. AFDs require clean, dry, cool air. Hydrogen sulfide in the atmosphere must be limited to 3 ppb or less. Place the air intake on the upwind side of the building, and pass the air through a filter of, for example, activated carbon.
- *Noise*. An annoying hum (sometimes an ear-piercing, high-frequency screech) is generated by AF controllers, and motors are also noisy, so consider sound-absorbing walls and ceilings to reduce the reverberation. IGBT controllers, however, are quiet and emit only a slightly noticeable hum.
- *Operational features*. Modern AFDs offer a wide variety of features. Listed below are the most common ones and are standard on most models:
  - Control. Speed can be adjusted either locally from a keyboard or a potentiometer, or remotely from a computer or control station. The drive can follow a 4–20 ma or 0–10 VDC analog or serial speed reference signal and can switch between them based upon a digital input or serial command. Input selectable speeds (minimum, maximum, etc.), each with its own acceleration and deceleration rates, are included. Local stop/start as well as remote start, with local mode selection are also included.

- Critical Frequency Avoidance. Skipping frequencies with adjustable windows enable the drive to avoid operating at speeds that resonate with the driven equipment.
- Power Loss Ride-Through. Extended power-loss ride-through helps avoid nuisance trips during brownouts or other outages.
- Automatic Restart. The drive can automatically restart following a fault trip. The number of restart attempts and the interval between them are selectable.
- Flying or Spinning Restart. After a power interruption, a spinning restart feature can detect direction and speed of a coasting motor and allows the drive to restore it to the correct direction and then accelerate or decelerate to set-point speed without tripping.
- Communication. Most popular industry-standard serial protocols are available for communicating with a station automation system, including Modbus RTU™, Devicenet™, Ethernet™, and Profibus™.

In conclusion, power system design for AF drives presents problems that have been largely ignored in the past and have resulted in many unsatisfactory installations. Modern AF drives and proper power system design should result in a problem-free variable-speed pumping system with a life-cycle cost lower than that of any other V/S drive.

Standby Generators

The harmonics generated by the converter are fed back to the standby generator and, unless the generator voltage regulators are specifically designed for compatibility with the AFDs, the generator may not operate correctly with the drive (i.e., the regulator may follow the converter pulse firing). The generator manufacturer must match the generator to the specific AFD to be used.

Maintenance

The problems of repairing AFDs are controversial. Some manufacturers claim that diagnostics and plug-in circuit boards make troubleshooting and most repairs relatively easy. However, most circuit boards are expensive, and untrained, nonelectrical personnel should not touch them. Moreover, replacing a thyristor requires a very skilled technician and the proper tools. Modern AF converters are built in a modular fashion, allowing for easier troubleshooting and replacement. A competent, factory-trained maintenance electrician should be able to service AFDs if:

- Training in power electronics is available at start-up and on a periodic basis.
- The proper tools are available.
- A spare of every type of printed circuit board and power electronic component is kept on hand. Many boards, even for different sizes of AFDs, are the same, so the inventory may not have to be large. (Circuit boards usually cost between $750 and $2500 each; the cost of repair is about half as much.) However, power electronic components are different for different sizes of drives. Another reason for keeping spare circuit boards is that some manufacturing firms continue to drop out of the solid-state business, and orphan electronics boards may be very costly to replace and slow in delivery.
- Routine maintenance is low and consists only of keeping the equipment clean and all connections tight.

Other manufacturers state that all but simple problems require a factory-authorized technician, who is sometimes accompanied by an electrical engineer at a charge of about $700 per person-day, portal-to-portal, plus travel time and expenses. Repairs often take two or three days. A worst-case scenario might be a day for travel and diagnosis, a day waiting for parts, and a day for installation and start-up, which could total $4000 or $5000. Troubles might occur yearly, more often, or never. Typically, one might expect no trouble at all for two or three years after start-up, with only occasional problems later. Note, incidentally, that one lightning strike can zap all of the AFDs, so *make sure that the station is protected from lightning* (see Section 8-5).

Some expert consultants with considerable and lengthy experience have had nerve-wracking problems with AFDs, and they approach decisions regarding their use with reservations and caution—especially for municipal projects. They point to the problem of competition with industry in hiring and keeping trained technicians and the difficulty in procuring spare parts for public utilities, and they urge that spares be included in the original bid package. Make sure that no critical items are omitted.

The industry is now more mature than it was 15 years ago, the equipment continues to improve, reliability is better than it was in the past, and new components continue to reduce the price of AFDs. The improvements, however, tend to make electronic equipment obsolete and parts may become unavailable. It seems wise to plan on replacing electronic equipment every 10 to 15 years, and this cost should

be included in the cost analyses of alternatives. The cost of repairs is quite uncertain, but for economic analyses of alternatives, the assigned annual cost should probably lie between 5 and 10% of the fob price of the equipment. There are records of annual costs well above 10% and other records of almost zero.

If C/S drives are to be considered as an alternative to V/S drives, include a comparison of (1) the construction costs of the wet and dry wells, (2) energy usage, and (3) maintenance needs as shown in Example 29-1.

Summary

When evaluating V/S drives, avoid the entrapment of comparing only the efficiencies. Instead, compare power losses in kilowatts at several of the loads and speeds to be imposed. An alternative is to compute the annual energy requirement for various increments of discharge (as shown in Example 29-1), in which the proper power rate schedule is used. Then add maintenance, replacement, and capital expenditure and reduce all of the costs to present worth. Include isolation transformers in the capital cost.

Finally, consider the intangibles of simplicity, flexibility, and reliability together with the owner's situation, such as the number and size of pumps and pumping stations, the size of the public utility, the location (especially with respect to repair facilities), the cost of power, the abilities of the operators, the effects on downstream operations, and the qualifications of the supervisors. The energy savings in one or two small drives would not justify the training and spare parts inventory needed for AFDs. If, on the other hand, there are half a dozen large drives, justification for using AFDs (or any other V/S drive) is likely. The key is to find the overall economics for each situation.

Electromagnetic Eddy-Current Couplings

Eddy-current couplings are nearly obsolete for wastewater pumping. Only two U.S. companies [5,6] now make them. The eddy-current coupling (Figure 15-26) is a unit of approximately the same size as the motor and is placed between the motor and the pump. The input shaft is coupled to the motor, which is usually a squirrel-cage induction motor. The output shaft, coaxial with the input shaft, is coupled to the pump. A small control panel is electrically connected to the eddy-current coupling.

An eddy-current coupling consists of two basic elements: (1) an iron ring (or drum) keyed to the input shaft and rotating with it at C/S, and (2) a multipole electromagnet keyed to the output shaft (see Figure 15-27). The electromagnet pole faces are very close to the inner surface of the iron ring, and the magnets are energized by low-power dc current from the control panel through brushes mounted on the frame and slip rings mounted on the shaft.

When the driving motor is started and the iron ring revolves, the electromagnet is energized through the brushes and slip rings. Electrical currents (eddy currents) in the ring are induced by the field of the electromagnet, and they create magnetic fields. For brevity, it can be considered that the electromagnet is driven by the magnetic fields in the ring. The difference between the rotational speed of the ring and that of the electromagnet is called "slip." The amount of slip depends on the magnitude of electrical current through the electromagnet.

Efficiency

The efficiency of the drive is determined by these factors:

- Motor and coupling friction and windage losses
- Motor core, winding, and stray losses
- Slip loss, which is by far the major loss
- Direct current supplied for coupling's field excitation.

The slip loss is the power lost in speed reduction due to slippage of the electromagnet with respect to the ring. Because of low friction and windage in the coupling, the output torque is only slightly less than the input torque. If friction and windage losses are neglected, the input and output torques are equal. Hence, the slip efficiency (power out/power in) is:

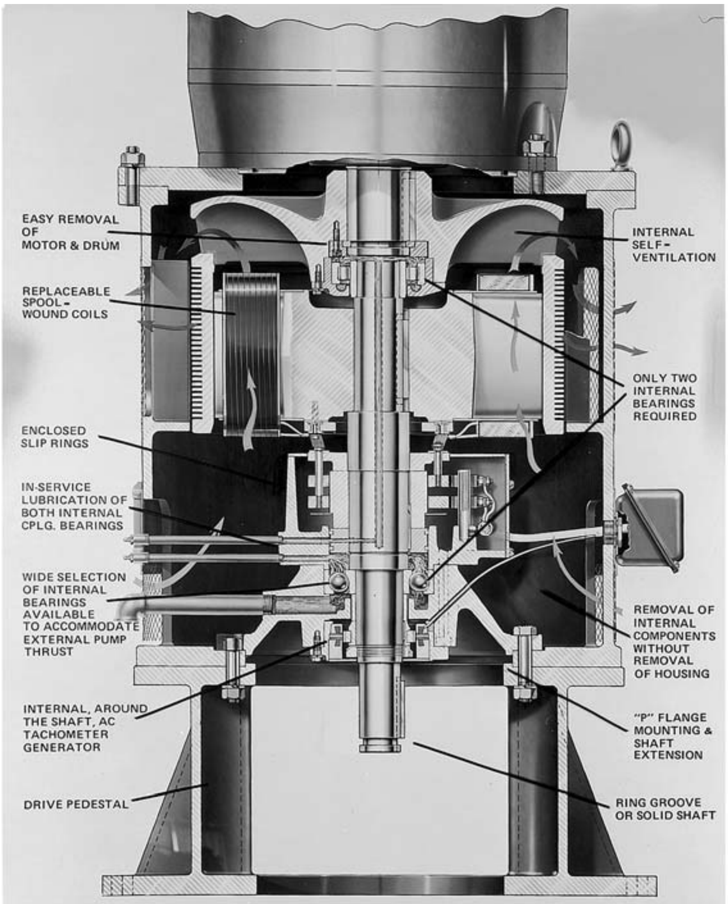
$$e_s = \frac{2\pi n_o T}{2\pi n_i T} = \frac{n_o}{n_i} \tag{15-2}$$

where n_o is output shaft (pump) speed and n_i is input shaft (motor-rated) speed. Thus, the slip efficiency depends only on the ratio of input and output speeds. It is linear and independent of load.

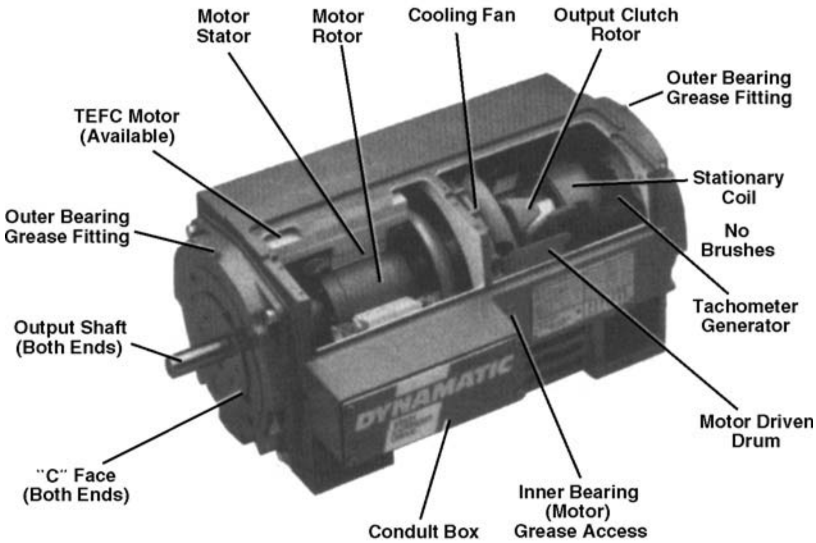
The loss of power due to slip is:

$$P_s = P_p\left(\frac{1}{e_s} - 1\right) \tag{15-3}$$

where P_p is the power output to the pump. So the power lost due to slip depends on the ratio of speeds



a



b

Figure 15-26. Air-cooled, eddy-current pump drives. (a) Vertical mount, salient pole, rotating field (with brushes); (b) horizontal mount, stationary field (without brushes). Courtesy of Dynamatic Corp.

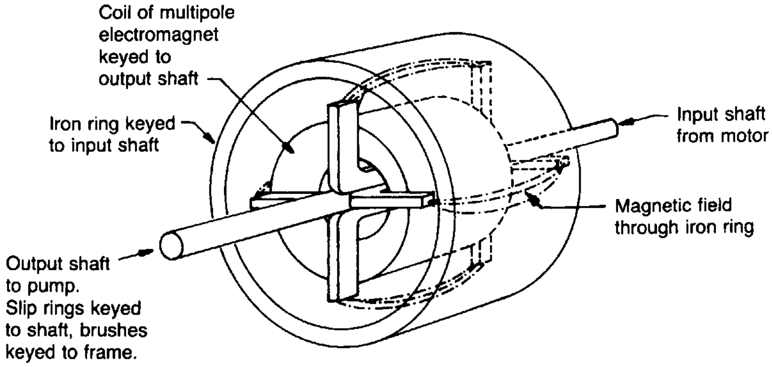


Figure 15-27. The principle of an eddy-current coupling.

and the magnitude of the load at the lower speed. Furthermore, the slip loss is the same for any type of slip drive, including wound-rotor/liquid-rheostat slip drives.

The load for centrifugal pumps may vary between the third and fourth power of the speed over the useful speed ranges, so the power lost may be relatively small even though the efficiency may seem to indicate otherwise. For example, at 75% speed for a load that varies as the fourth power of the speed, the power lost would be:

$$P_s = (0.75)^4\left(\frac{1}{0.75} - 1\right) = 0.105$$

or 10.5% of the power required at 75% maximum speed. Although these losses are not negligible, they should be considered objectively in a life-cycle engineering cost study that includes not only power costs but also the maintenance, reliability, and expected life of the equipment in comparison to other drives.

A typical power loss for an eddy-current coupling driving a pump is shown as a "slip drive" in Figure 15-25.

Stationary Field Eddy-Current Couplings

By altering the design of the unit, the rotating field of an ordinary eddy-current coupling can be converted to a stationary field, which eliminates the brushes and slip rings. Two U.S. companies offer this design. The efficiencies of the new and old designs are almost the same and the prices are comparable, but eliminating the slip rings and brushes reduces the maintenance to an amount comparable with a squirrel-cage induction motor.

Permanent-Magnet Eddy-Current Couplings

The permanent-magnet eddy-current coupling is essentially a compact mechanical version of an eddy-current clutch with an actuator arm that moves two rare-earth magnetic discs closer (for higher speeds) or farther apart. In the electromagnetic version, magnetic field intensity is controlled electrically by adjusting the current to the magnetic coils. In the permanent-magnet version, field intensity is controlled by adjusting the distance between two opposing pole magnets. The strength of the field is approximately inversely proportional to the cube of the distance between the opposing magnetic plates. One plate is stationary; the other is moved by an arm, which is in turn moved by an electro-mechanical actuator. By controlling the position of the arm, the speed of the coupling is affected (i.e., the closer the magnetic plates, the closer the output speed is to the drive motor speed, and vice versa).

Permanent-magnet eddy-current couplings are available in both horizontal (Figure 15-28) and vertical orientations. They are simple, low-maintenance versions of the eddy-current clutch. They have basically the same limitations as the electromagnetic eddy-current clutch but with better efficiency because of the absence of an external magnetizing field generator.

Rheostat-Controlled Wound-Rotor Drives

Rheostat-controlled wound-rotor V/S drives, particularly liquid rheostats, were common several decades ago. Today they are an obsolete technology and no longer used for new pumping stations, although some are still in use in old stations. The second edition of this book contains a description of them.

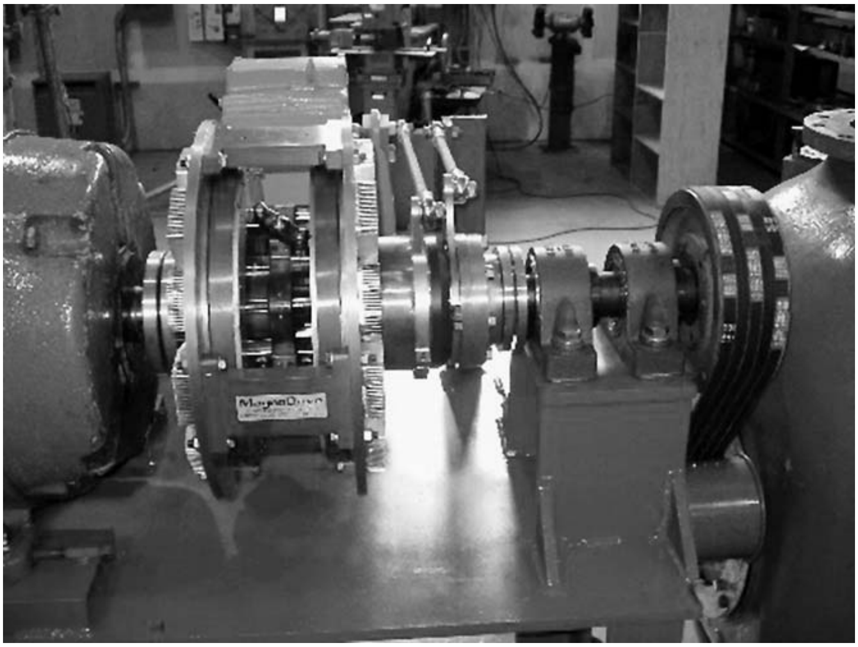


Figure 15-28. Horizontal permanent magnet eddy-current coupling. Courtesy of Magnadrive Corporation.

Fluid Couplings (Hydrokinetic)

The drive in fluid couplings consists of a squirrel-cage induction or synchronous motor and coupling. The coupling, enclosed in a housing, contains these major components:

- An impeller, which is mounted on and keyed to the input shaft.
- A runner, which is mounted on and keyed to the output shaft.
- A scoop tube.
- A casing, which is mounted on and keyed to the input shaft.

A very small, electrically driven pump is mounted on the outside of the housing. The pump delivers oil from a sump at the inside bottom of the housing to the casing. Oil is free to flow from the casing into the impeller and runner.

The impeller and runner have internal blades similar to those in a recessed impeller of a pump. Rotation of the impeller imparts energy to the oil in which it is partially submerged. The oil is "pumped" to the runner and imparts energy to the runner blades, thereby causing the runner and the output shaft to rotate.

The depth to which the impeller and runner blades are submerged in oil directly determines the torque that is transmitted to the runner and the output shaft. The depth of oil in the casing, and thus the submergence of the impeller and runner blades, is adjusted by a "scoop tube" that removes oil from the casing and returns it to the sump. The scoop tube can be positioned from the exterior of the housing through a linkage, and its position determines the oil level in the casing.

Slip-Recovery Drives

A slip-recovery drive consists of a wound-rotor induction motor and a solid-state controller.

The controller is electrically connected to the rotor brushes of the motor, and the effective resistance in the controller is electronically adjustable (see rheostat-controlled wound-rotor drives). Rotor circuit current does not actually pass through resistors, but is rectified, inverted to 60 Hz, and returned to the motor stator terminals. The power losses that occur in resistor-type secondary controllers do not exist in the slip-recovery drive, which is slightly more efficient than the AFD. The controller should include all of

the necessary capacitors for power factor correction and all of the circuit breakers required for line disconnect.

Variable-Ratio Belt Drives

The variable-ratio belt drive consists of a squirrel-cage induction motor and two variable-pitch sheaves with a belt, shafting, bearings, a sheave-pitch-changing mechanism, and a large spring, all of which are enclosed in a housing.

One sheave (input) is either on a shaft directly coupled to the motor shaft or mounted on an extended motor shaft. The second sheave is mounted on the output shaft. Power is transmitted from the input to output sheaves by a heavy-duty vee-belt.

Output shaft speed changes are accomplished by changing the effective diameters of the sheave hubs by changing the spacing between the sheave discs. The discs are inside-tapered so moving one disc (along the shaft) closer to the other disc of the sheave forces the belt to engage the disc at a greater distance from the hub. The position of one disc on the input sheave is changed by an external linkage. The position of the disc of the output sheave is changed by an internal spring to keep belt tension relatively constant regardless of changes in output speed. The discs should be chrome-plated to reduce wear.

Hydrostatic Drives

A hydrostatic drive consists of the following:

- A squirrel-cage induction motor
- An adjustable displacement, multi-cylinder hydraulic pump
- A fixed displacement, multi-cylinder hydraulic motor
- An oil reservoir.

The hydraulic pump is flexibly coupled to the ac motor, and its shaft turns at C/S. The cylinder block (which is cylindrical in shape) is keyed to the shaft and contains the pistons. The entire assembly rotates with the shaft. The piston rods bear and slide on a "swashplate" that does not rotate. The swashplate can be tilted by an external device (through a linkage) to change the effective displacement of the pump.

The hydraulic motor is connected to the hydraulic pump by means of two hoses. The output of the pump is circulated through the motor and back to the pump intake. The shaft of the hydraulic motor is flexibly coupled to the load shaft. The motor construction is similar to the pump, except that its swashplate tilt is fixed. The motor output is connected to the shaft of the wastewater pump. Changing the swashplate angle changes the discharge rate of the hydraulic pump, thus controlling the speed at which the motor turns.

Refer to Table 15-3 for comparative advantages of hydrostatic drives.

Other Drives

Other drives are available but are not described here because they are not widely used for pumping water or wastewater. Obviously, this situation may change, so be alert to new products. Many types of rotating machinery that are quite satisfactory for some specific use, however, are unsuitable for municipal pumping. Be cautious in applying a new product by making a thorough investigation of installations and their maintenance records.

15-12. References

1. Karrasik, I. J., J. P. Messina, P. Cooper, and C. C. Heald, *Pump Handbook,* 3rd ed., McGraw-Hill, New York (2001).
2. MagnaDrive Corp., Seattle, WA.
3. Davis, C. V., and K. E. Sorensen, *Handbook of Applied Hydraulics,* McGraw-Hill, New York (1969), pp. 40–16.
4. Fair, G. M., J. C. Geyer, and D. A. Okun, *Water and Wastewater Engineering, Vol. 1, Water Supply and Wastewater Removal,* John Wiley, New York (1966).
5. Ideal Electric Co., Mansfield, OH.
6. Electric Machinery Co., Inc., Minneapolis, MN.
7. DSI—Dynamatic, Inc., Sturtevant, WI.

Chapter 16

Pump-Driver Specifications

DAVID L. EISENHAUER
THOMAS M. FLEGAL
GARR M. JONES

CONTRIBUTORS
Richard O. Garbus
Philip A. Huff
Melvin P. Landis
Ralph E. Marquiss
Earle C. Smith
Horton Wasserman

Two approaches to writing specifications for pumps and drivers are (1) investing "unit responsibility" for the complete pumping unit (driver, shafting, pump) with a single manufacturer, and (2) separating driver, shafting, and pump so that each can be furnished by a different supplier and assembled at the jobsite by the general contractor. These two approaches are discussed and compared in Section 16-1. The remainder of the chapter is confined to the first approach (unit responsibility) and an explanation of how to apply it to custom-engineered pumps and drivers for a high-lift wastewater pumping station. "Custom engineered" means that most manufacturers must modify their "off-the-shelf" pumps with larger shafts, more rugged bearings, or some special materials to comply with the specifications. Other manufacturers may produce high-quality pumps that need no improvements.

This chapter is keyed to (and is an explanation of) Appendix C, which contains two independent example specification sections based on actual project documents. Section 11050 specifies requirements common to all centrifugal and axial flow pumps, and Section 11304 stipulates detailed requirements applicable to the specific pumps to be provided. Paragraph 11304-1.02B of Appendix C shows how to vest the manufacturer with complete responsibility for properly coordinating sizes, system analyses, and procurement of the complete pump and driver system.

Specifications should be so clear that they cannot be misunderstood (see "Specification Language" in Section 28-1).

16-1. Comparison of Two Approaches to Writing Specifications

Many experts believe that the unit-responsibility approach is superior for public works projects for the following reasons:

- Fixing complete responsibility for the equipment with a single manufacturer.
- Eliminating problems of misfit or mismatch in either size or performance (such as torsional resonance, reed frequency, and critical speed).
- Providing the owner with a single source for spare parts, troubleshooting, and future modifications.

- Ensuring adequate design in subjects for which the consultant's personnel may be lacking in training or experience.

For the common, general-duty application of standard, off-the-shelf pumps and motors of small pumping units—for example, 20 kW (25 hp) or less—the adoption of unit responsibility is less important but, nevertheless, has advantages over separate purchase specifications.

Some engineers prefer the second approach (the separation of specifications for each component of a pumping unit), but responsibility for properly mating all components then falls on the engineer. A list of the added responsibilities the engineer must assume includes the following:

- Coordinating the equipment to obtain correct overall shaft length.
- Coordinating the compatibility of couplings.
- Determining the critical speed for pump, motor, and shafting.
- Analyzing the complete system for torsional resonance (see Chapter 22).
- Coordinating the complete system to avoid dangerous resonance frequencies.
- Selecting the motor starter.
- Matching and coordinating the speed of the pump and driver.
- Ensuring the compatibility over the entire speed range of variable-speed drives.
- Coordinating the field testing of the entire system.

Consulting engineers normally do not have the resources necessary to perform these tasks because they do not manufacture the products. They are in no position to coordinate the multitude of mechanical, structural, electrical, and instrumentation details, and they rarely have the training or contractual relationship to do so.

Some contractors may prefer to purchase equipment components separately to obtain lower prices. In such cases, the owner loses value because of the lack of proper engineering in selecting the various pieces of rotating equipment and their appurtenances. The specification provision requiring unit responsibility is one of the best approaches available for controlling the purchase of individual unmatched (or uncoordinated) components. The owner always benefits from an open bidding process in which more than one manufacturer is allowed to meet the specifications. Specifications, however, should be clearly and properly drafted (consistent with applicable laws) to limit equipment to that most suited to the project. In many (if not most) pumping station projects, more than one manufacturer is capable of providing equipment within the limitations established by the owner's budget and project needs. It is the engineer's responsibility to provide the specification mechanism that will ensure good equipment and good mating of equipment components.

An example of the consequences of a contractor's evasion of the unit-responsibility clause (see Appendix C, Paragraph 11304-1.02B) was found in the construction of Sunset Beach Pumping Station in Steilacoom, Washington. Because the contractor purchased parts from different suppliers, the parts did not fit, completion was delayed by four months, and the contractor suffered significant losses due to penalties plus the cost of new equipment. Permitting such practice has no benefit to either contractor or owner and makes for poor coordination and higher cost. One means to forestall such noncompliance is to require an affidavit signed by both the contractor and an officer of the manufacturer, under the penalty of perjury, that the latter has accepted unit responsibility in accordance with the contract documents.

16-2. Methods for Specifying Quality of Equipment

Several methods for specifying the quality of products are compared in Table 16-1. More complete descriptions are given in Section 28-5 and in the CSI Manual of Practice [1].

16-3. Nonrestrictive Specifications

Appendix C is an example of a combination descriptive, performance, and reference standard approach, to a nonrestrictive specification for pumps and drivers. The specification is a guide for (1) writing a pump specification, and (2) the topics to be included. The format conforms to the CSI three-part section and page formatting guidelines [2].

The example in Appendix C illustrates how both performance and descriptive specification systems are combined in writing a pump specification. Brand names or manufacturers are seldom used in the example. The paragraphs on design requirements and operating conditions state the end results to be achieved and also include the necessary restrictions. Minimum salient characteristics are specified that limit the contractor's choice of equipment but that are sufficiently open to permit competition.

Table 16-1. Methods for Specifying Equipment

Method and description	Advantages	Disadvantages
Performance (functional results specified)	Common in building systems. Concept can be incorporated into any specification.	Burdens contractor with design. Acceptance not ensured. Cost is high.
Proprietary (name or brand and model specified)	Ensures quality desired. Short and easy to prepare.	Bidding noncompetitive. Requires assurance of fair price. Not allowed in public works without extensive justification.
Descriptive or generic (performance, materials, quality, etc., described in detail)	Excellent in principle.	Considerable time to research and write. Often misused. Difficult to evaluate product submitted.
Nonrestrictive (also "or equal") (name two or more products and add "or equal")	Common method for complex products.	Evaluation is subjective, so documentation is required for rejects. Requires at least some performance or descriptive clauses.
Reference standards (reference to a standard specification)	Can be used in conjunction with the above methods. Familiar to contractors. Easy to write.	Limited to simple components or materials. Specifier must understand the entire reference.[a] Ease of use may lead to poor or incomplete specifications.

[a]Not necessarily disadvantageous. Specifiers have a duty (1) to understand fully each reference specification and (2) to include qualifying statements about which material does or does not apply to the equipment specification.

Research Needs

Except for the electric motor specification (which is basically oriented toward certain minimum performance criteria necessary for the project), most of the material requirements in Part 2 of the specification are listed in descriptive form. Preparing such descriptive portions requires researching pumps, pumping systems, and equipment to determine what is appropriate for the intended service and the critical features of materials and equipment required to perform under project conditions. Specification research consists of (1) developing and listing the performance requirements of the equipment; (2) identifying any special needs, such as special metallurgy or resistance to abrasion or corrosion; and (3) requesting preliminary proposals specific to the project requirements from candidate manufacturers. The final specification can then be drafted using the manufacturers' project-specific proposals as references. Descriptive requirements are also based on experience with specifications for similar equipment items as well as on engineering judgment.

Effect of Size and Complexity

The content of any equipment specification varies greatly with the size and complexity of the project as well as with the equipment application and operating requirements. The example specifications are written for wastewater centrifugal pumps of the end-suction, overhung-shaft design. Custom-engineered pumps are specified because, when operating at reduced speed against high discharge heads, there is a potential for damaging conditions caused by excessive vibration and unbalanced radial forces. Specifications for pumps operating at constant speed and low to moderate heads need far less detail and restrictive requirements, so they can be written around most manufacturers' standard pump designs. The example is not intended to exclude standard pumps if they comply with the specification. Instead, the special nature of the intended application is recognized in a specification that is designed to alert the manufacturer to these requirements.

In the following paragraphs, portions of the specifications in Appendix C are discussed and informa-

tion is provided to assist in the preparation of a specification that will obtain the pumping unit required for a particular design or installation. Where proprietary specifications are permitted, the specification can be shortened considerably. But first read the warnings given in ''Proprietary Specifications'' in Section 28-5.

16-4. Operating Conditions

Many pump manufacturers will guarantee only one operating point for a given pump. Nevertheless, it is necessary that the pump manufacturer be aware of all operating conditions critical to pump performance and operation that are expected to occur over the life of the pump—particularly for variable-speed pumping installations. A method for presenting several operating conditions is given in Paragraph 11304-1.01E of Appendix C. In that example, a listing of anticipated operating conditions is followed by detailed notes that explain the significance of each condition and also specify criteria delineating acceptable pump operating characteristics. Note in particular the use of ''Allowable'' and ''Preferred'' Operating Regions as defined in ANSI/HI 9.6.3 to describe acceptable performance.

16-5. Mass Elastic Systems and Critical Speeds

All pumping units must operate at speeds sufficiently removed from critical speeds that could cause destructive vibration and fatigue failure. 15050-1.04C of Appendix C shows provisions designed to assure that. For larger pumps operating at variable speed, a formal analysis of critical speeds and the complete mass elastic system is generally warranted and should be considered, particularly for custom-engineered units. The results guide the equipment manufacturer in avoiding combinations of rotating elements that may be subject to fatigue failure. Paragraph 15050-1.05B of Appendix C lists recommended criteria for determining whether a pumping unit should be subjected to such an analysis, requires the analysis to be performed by an experienced, registered design professional and specifies the methodology to be employed (also see Section 22–11).

16-6. Pump Testing

The true test of pump performance is satisfactory operation in the intended installation, and it may be desirable or necessary to test a pump in the factory or in the field (or both) to provide assurance of the pump's capacity to meet project requirements. Consider, however, that the methodology used to determine the required flow rate is the result of projections based on unit demand factors or total consumption figures developed from historical data. These data, in turn, are based on flow records (which are usually inaccurate) and on population estimates, which in themselves are never more accurate than $\pm 10\%$. Headloss figures are based on estimations of pipe wall roughness and data for turbulent losses through fittings, valves, and special conditions (such as dividing or uniting flows) in which each loss is within the ''shadow'' of another and losses may easily double by swirling. Some of these data were developed from model studies (1) without a reasonable allowance for the effects of scale; (2) were performed many years ago, some under questionable assumptions; and (3) with metering and measuring devices less accurate than the apparent precision of the hydraulic computations in which they become an integral part. On the other hand, the range of static heads can be precisely determined, a range of pipe friction losses can be confined to an acceptable band with confidence, and the inaccurate losses in fittings are a small part of the total losses and, hence, of minor effect. In summary, pump performance requirements should be viewed objectively—not with the assumed precision many engineers assign to them.

Ensuring that the pump supplied will meet the required performance specifications requires a guarantee from the manufacturer, which, in turn, may require a factory test. Whether the test should be made at all (with the designer and client relying on previous tests of similarly sized impellers), whether a specific pump test should be witnessed, and whether factory tests must be followed with field tests depend on the cost–benefit ratio of testing and the need for positive assurance of performance. One way to approximate the cost–benefit ratio is to speculate about the effect of efficiency. Suppose, for example, the efficiency of a pump is 85% for a flow rate of $0.38\,m^3/s$ (6000 gal/min) at a head of 15 m (50 ft) and the efficiency of the motor is 95%. For a life of 20 yr at 8% interest with power at 6¢/kW · h, the present worth of a 3% change in efficiency is nearly $11,000 (see the interest formulas in Section 29-4). A factory test of a pump of this size might cost the owner $2500 (depending on the configuration of the unit). A witnessed test might cost nearly double plus the cost for the consulting engineer's on-site representative. Either a factory test or a witnessed factory test would establish not only the efficiency but also the head and flow rate under test conditions and, to some degree, vibration and noise.

Test Requirements

At the outset, the designer should identify two or more manufacturers with the capability of conducting full-scale tests. Mid-sized equipment (up to 500 hp, 480 V, and 15,000 gal/min capacity) can be tested by most manufacturers. At least three manufacturers can test equipment up to 1500 hp and 4160 V. For larger equipment, it may be necessary to utilize an independent hydraulic laboratory. Despite these limitations and difficulties, some experienced engineers believe the cost of properly performed tests of each pump is money well spent. Consider, however, the following means of keeping costs at a minimum.

- Discuss the proposed tests with at least two manufacturers. Determine the probable cost of the tests and any viable modifications.
- Consider using a factory-calibrated motor or dynamometer. However, failure to use the actual production motor on large units leads to controversy if subsequent wire-to-water power field tests vary significantly from factory test results.
- Test at constant speed. But for variable-speed pumps, select four or five well-spaced operating points from minimum to maximum flow rate.
- Consider testing only one of a group of identical units, but be aware that (1) the manufacturer may not guarantee any pump not tested, (2) the efficiency at the design point may vary as much as 3% and there may be significant variation in shut-off head and performance curve characteristics due to machining tolerance or errors, and (3) some experts insist on testing each pump in the group.
- Test only those pumps for which a cost–benefit ratio is reasonable. If the costs are too great, it may be more cost-effective to oversize the pump and driver.
- Consider whether nonwitnessed pump tests are sufficient. Much depends on the manufacturer's reputation and standard methodology in testing. One benefit is that it affords the owner the ability to assess the pump's potential for producing excessive noise.

Factory Pump Tests

If factory tests are made, the following are essential:

- Require the pump manufacturer (or the general contractor) to guarantee performance at the specified conditions of head, capacity, and efficiency. But note that unless the contract is a direct pre-purchase one, the owner has no direct contractual relationship with the manufacturer, so obligate the general contractor, who in turn obligates the manufacturer.
- Require that the testing and the test setup conform to ANSI/HI 1.6 together with any special requirements [e.g., testing a submersible pump with its discharge elbow and a 1.5-m (5-ft) length of discharge pipe].
- Require that both test logs and curves be certified by an officer of the manufacturer's corporation. (Note that submittal of test logs is usually a special requirement.)

Performance requirements for the pumping station may not be satisfied if the field conditions cannot be reasonably met in the factory. For example, it may be impractical to test a long, deep well pump in a vertical position without reducing the length to a few bowls. Very large pumps may exceed the capabilities of the factory test site. Note also that ANSI/HI 1.6 standard tolerances are (depending on capacity and head) as much as 8% of specified capacity or head as the default. (Note: The standard is currently being rewritten so consult the Hydraulic Institute for updates.) Be careful to ensure that these tolerances cannot overload the driver.

Both factory and field tests of pumps are discussed at length by Dicmas [3]. Groundwater pumping tests are extensively described by Walton [4].

Witnessed Pump Tests

Owner-witnessed factory tests cost considerably more than factory tests because (1) the tests are made twice—once to make sure the pump will meet specifications and again when the owner's representative is present—and (2) the owner must pay for the engineer's attendance and travel. Still, witnessed factory tests are valuable because they tell the manufacturer that the owner is serious, which encourages honest performance in test setups. Anecdotes abound about the discovery of faults by a witness, but none, however, has been personally experienced by the authors of this chapter.

The benefit to the owner of witnessed tests depends on many factors, including the size of the pump, the projected hours of service, the cost of power, the reputation of the manufacturer, and the engineer's familiarity with the manufacturer's test methodology. In general, it is wise to confer with the owner to assess the potential benefits before specifying that tests be witnessed.

The following is a checklist for the engineer who is required to witness a pump test:

- Be thoroughly familiar with project specifications, any referenced standards, and the manufacturer's proposed test setup prior to arriving for the test. Be familiar with the test procedures and setups in ANSI/HI 1.6 and in Dicmas [3].
- Require submission and approval of the test setup prior to beginning the test.
- Develop a checklist based on project specifications and referenced test standards prior to arriving for the test.
- Be completely familiar with the test data compilation procedure prior to the test date.
- Be sure to arrive early enough the night before to be fully rested and alert.
- Be prepared to stay longer than scheduled if a test fails.
- Upon arrival, check the test setup against the agreed-upon testing procedure.
- Obtain photocopies of the calibration records for all test gauges, meters, motors, and dynamometers.
- Check the calibration curves against the serial number of each gauge, meter, motor, and dynamometer.
- Go over the test procedure with the manufacturer's test engineer to be certain all concerns have been addressed.
- Note the time testing begins for each pump.
- Make a note of the serial number of each pump. (On bowl-type pumps, the serial number is on the bowl.)
- Check all zero points or readings before starting the test.
- Once testing begins, observe the data recorded by the test technician, read gauges, manometers, and other instruments at the same time that the technician reads them, but do not record any data, only confirm. Be certain gauges and manometers are stable at the time readings are taken.
- *Subjectively* assess pump vibration by feeling each pump case. A pump usually vibrates more on the test stand than in the pumping station because the pumps are not mounted on a proper foundation and are not driven by the field drivers. On the other hand, resonance or equipment problems may occur in the field.
- Listen for any unusual noises (which may be difficult because of ambient noise at the test site). Use a mechanic's stethoscope to listen for unusual or excessive case noises.
- Note the time that testing is finished for each pump.
- Sign each test log and obtain photocopies of all test data as soon as they are developed.
- Spot-check the calculated results by reducing a few data points yourself.
- Do *not* approve or sign developed performance curves until you have independently checked four or more data points.
- Be prepared for long periods of inactivity if several pumps are to be tested. It takes awhile to disassemble one test setup and assemble the next. Take your briefcase, work on the report, study this textbook, or tour the plant. This period of inactivity is an excellent opportunity for an experienced engineer to educate an intern who can then be the witness for other tests.

When the test is completed, the witness should prepare a report for the owner that describes the test, conditions, observations, and results. Appendices to the report should include copies of the test logs, generated curves, and any other information pertinent to the test procedure or results.

Variable-Speed Pumps

Many specifications require that the intended field drive equipment be used in the tests of variable-speed pumps, which doubles or triples the cost of the tests. The perceived benefits include: (1) the evaluation of the pump's ability to perform at specified conditions, (2) the determination of wire-to-water power efficiency, (3) the determination of the compatibility of all components, and (4) the discovery of objectionable vibration. The setup in a manufacturer's test bed, however, is never the same as in the final installation. Although efficiency and compatibility can indeed be assessed, harmonics in AFDs are a function of the entire electrical system, including the supply, which is different in the field. Observations of vibration under various operating conditions are qualitative at best because the vibrational characteristics of the supporting structure are missing in the field. Factory tests of a complete pumping assembly are particularly expensive if special test stands must be fabricated—for example, for motors mounted on a floor above the pumps. Shipping motors, controllers, and other parts of the driving units to the pump manufacturer as well as the assembly, disassembly, electrical work, coordination, and possible delays increase the costs for testing complete pumping units. In summary, engineers should have a clear idea of what is to be accomplished, exactly how the tests should be run, what added costs are incurred, and the benefits that accrue from such tests before specifying factory tests of complete pumping assemblies.

Example Specifications

Example specifications for hydrostatic and performance testing are given in Paragraphs 11050-1.06C and 1.07B of Appendix C. The testing procedures in paragraph 11050–1.07 are particularly desirable for custom-built pumping units.

Field Pump Tests

Field pump tests are of two general types: (1) *field operational pump tests* in which accuracy of flow and head are measured to within an error band (or uncertainty) of ±5 to 7%, and (2) *pump acceptance tests* in which final acceptance may be based on satisfactory field performance with reward/penalty clauses written into the contract for failure of the pump to perform within specified limits. Such a specification requires expensive tests that may have limited practical value because manufacturers cannot be responsible for factors outside their control, such as improper sump design with intake problems, insufficient NPSHA, system characteristics different from those submitted, and inadequate supporting structure. A good factory test is a great help in identifying and resolving such problems.

Field Operational Tests

Comprehensive field operational tests are important, and many engineers require such tests of every installed pump as a check in which the entire pumping unit (including instruments, controls, motor, pump, and valves) is observed for operational integrity and function. Testing flow rates and pressures at allowable errors (or uncertainties) of 5 to 7% is not difficult, time-consuming, or expensive. The methods given in Section 3-9 for measuring the roughness of pipe apply equally well to the testing of pump flow rates. (Note, however, that a pump manufacturer's guarantee should apply only to the factory test.) Such tests are valuable for discovering trouble. For example, in a two-hour field trial, a single-stage vertical turbine pump delivered 30% less flow than expected. When the pump was disassembled, the impeller was found to have slipped down the shaft. Repairs restored the full discharge capacity. Without the test, the inefficiency would not have been discovered. A field test is, however, as much a test of the hydraulic design of the pumping station and of the technicians' skill as it is a test of the pump and driver. For example, vortices profoundly affect the pump performance, and these are, in turn, a function of the sump design and the suction intake velocity. Confer with the manufacturer to make sure the sump in the field test is acceptable. The field operational test is also valuable as the basis for the intelligent operation of the facility.

Before beginning field tests, observe the sump surface for vortex formation or signs of disturbance that would lead to rotation. Vortices may be intermittent on 15- to 30-s intervals, so allot sufficient time for the observation. Even with the use of clear water, subsurface vortices are not visible, but if they are suspected, dye (which is used in model studies) can also be used in field tests, although the water in the system must be changed when discolored. An alternative is to use colored plastic chips of polyethylene (which has a specific gravity of 0.92 to 0.96). Chips $25 \times 25 \times 1.6$ mm ($1 \times 1 \times 1/16$ in.) are about the right size. They can be put into a small sack, which can be positioned at the suspected origin of a subsurface vortex and then broken to release the chips. (See various directories [5, 6] for suppliers of pigmented polyethylene chips.) The chips slowly float to the surface and can be netted. If vortices are detected, model studies are definitely required.

Flow can be measured with the pumping station flowmeter after calibration in situ by volumetric or tracer methods (see Section 3-9). If no permanent flowmeter is installed, a temporary flowmeter, such as a strap-on magmeter, orifice, pitot tube, or elbow meter, can be installed and also calibrated in situ. Alternatively, tracer and/or volumetric methods can be used for metering the flow throughout the test.

Because one purpose of an operational test is to provide the basis for facility operation, it is a *change* in the operational status—not absolute values—that is important. Consequently, the pressure gauges to be used by the operating personnel should be used for the test. The gauges should be calibrated and installed in accordance with ANSI/HI 1.6. The discharge pressure gauge should be located at least three pipe diameters downstream from the pump discharge nozzle with no intervening flow disturbances such as those caused by valves or fittings. Electrical instruments in pumping stations are not adequate to obtain overall (wire-to-water power) efficiency. If the efficiency must be checked, special meters must be temporarily installed.

When running the tests, try to obtain two or three separated test points of head versus flow rate on both sides of the normal pump operating points.

If a special lateral and torsional analysis has been required as a part of the equipment selection process, consider requiring a field torsiograph test to confirm the design.

Field Pump Acceptance Tests

Only with extraordinary care and special instrumentation can the errors in field testing be reduced almost to the same order of accuracy obtainable in factory tests (about 1%). The principal disadvantages of acceptance by precision field testing are the cost and the difficulty of rejecting a unit that fails the test. Claims (and, perhaps, lawsuits) will abound in regard to (1) the accuracy of the field test, (2) the validity of the design point, and (3) the reasonableness of the flow rate requirements. Delays due to repairs, adjustments, or replacements defeat the project schedule and can force the engineer to a compromise adverse to the client. A comprehensive, witnessed factory test is not foolproof, but it can usually help to avoid such confrontations.

Consider, on the other hand, a factory test of a large, long, multistage deep well turbine. Some manufacturers might be able to test only a short column length consisting of, perhaps, only a single bowl (or four to five bowls at most). It might be necessary to operate at reduced speed because of power supply limitations. In the field, however, full-speed operation can be performed to evaluate (1) the suction well conditions, (2) the oscillation of the full assembly in its vertical configuration, (3) the actual hydraulic losses in the full column length, (4) the mechanical friction losses in additional shafting and intermediate bearings, (5) the true power requirements, and (6) the overall efficiency. The accuracy of such a field test might be superior to a factory test, provided that (7) the technicians are knowledgeable and skillful, (8) the pumping station was designed with field testing in mind (isolation valves and drain pipe from force main to sump for recirculating the flow), (9) pressure gauge taps are properly located, and (10) calibrated, sensitive instruments are used. But note that (11) the calibration of flow meters to an accuracy comparable to a factory flow meter (repeatability of, for example 0.5%) requires very great care, considerable skill, and meters that are installed in strict accordance with the manufacturer's recommendations, (12) pressure gauges and meters must be 5 or, better, 10 diameters downstream from disturbances, (13) the pressure taps should be located at quadrant points connected to a common collector, and (14) the pressure gauges should have a face at least 200 mm (8 in.) in diameter and must be readable and accurate to 0.25%. The other instrumentation must be of similar precision to approach the accuracy of the factory test (see Dicmas [3] and ANSI/HI 1.6. Hence, specify factory tests where proper instrumentation is available, followed by field operational tests unless circumstances are so unusual that such a procedure is inadequate.

16-7. Shipping Major Pumping Units

Damage to major equipment items during shipment is of concern on most construction work, particularly with custom-made units. To protect against potential damage from sudden impact, it is often desirable to specify shipping requirements and restrictions. Recording accelerometers are available on a rental basis from most railroad and trucking firms at a slight additional cost. The presence of the device on a shipment usually provides some incentive on the part of the common carrier to exercise more than usual care. A 3.0 *g* value (of acceleration) is a commonly accepted criterion for indicating potential damage.

On a recent project, three 125-horsepower motors were shipped to a jobsite with recording accelerometers as specified. All accelerometers recorded values at least as great as the peak chart value and many times the specified maximum tolerable acceleration. The contractor was required under the terms of the contract to disassemble the motors and allow inspection by the owner's representative. This inspection revealed bent shafts, damaged bearings, and other defects—thereby proving the worth of the specification.

16-8. Submittals

If the product is specified by performance criteria or is to be used in a complex system, or if a variety of contractor options are permitted, submittals should be specified as in Paragraph 11050-1.08 of Appendix C. Submittals are also required for equipment or materials that have long delivery times to reduce the chances of delays due to the rejection of the products after delivery.

Submittals listed in the specifications are designed to provide sufficient information to the engineer regarding the design to indicate that the contractor has properly interpreted the drawings and specifications. Because the submittal requirements are specific and limited, the contractor does not have to guess about the amount of information required and the design engineer does not waste time reviewing irrelevant or unnecessary documents. The submittal and the specification together should give a description sufficient for the engineer to determine whether the contractor's planned equipment or material will conform to design requirements. Specify only the infor-

mation needed to make that determination. The submittal material requested in the pump and driver specification, Paragraph 11050-1.08 of Appendix C, conforms to the above criteria.

16-9. Product Data

Equipment specifications also require that other information and product data be provided to meet a wide variety of owner needs. Much of the information goes into files as part of the project records, some is used by the engineer or construction manager in inspecting the contractor's work, and other data form the basis of the O&M manual. There must, however, be sufficient detail for the engineer or construction manager to determine the acceptability of the equipment. Examples of information to be provided with a pumping unit are listed in Paragraph 11050-2.06 of Appendix C.

16-10. Seals

Mechanical seals are often preferred over conventional packing because mechanical seals reduce maintenance. But mechanical seals are not only costly but are subject to failure on occasion, and the failure may be sudden. If packing fails, the pump can usually be kept running by temporary adjustments until it is convenient to shut it down. If a mechanical seal fails, the pump usually must be shut down at once. Some types of mechanical seals, however, fail gradually with ample warning.

There are a variety of proprietary seal designs, and no industrywide standard exists as a convenient specification reference. Because seals are a continuing maintenance concern and interchangeability is desirable, it is recommended that, if possible, one manufacturer be required to supply all mechanical seals.

16-11. Pump Shafts

The extensive specification for the pump shaft (Paragraph 11050-1.05C in Appendix C) is intended to require specific attention by the manufacturer to the need for heavy-duty shafts for applications where high discharge heads and operation at variable speed may result in substantial radial thrust loads. The methodology specified as the basis for the calculations is presented in ANSI/HI 1.3. The maximum allowable shaft deflection of 0.037 mm (1.5 mils) at the face of the shaft seal (Paragraph 11050-1.04D.4 in Appendix C) is considered a reasonable objective.

16-12. Pump Shaft Bearings

For pump installations where radial thrust is a problem, bearings must be sized and rated to withstand successfully the worst loads applicable to the specified continuous operating conditions. To ensure compliance with the load requirement, Paragraph 11050-1.09D.5 of Appendix C requires the submission of documentation to support the selection of bearing sizes.

As a guide, bearing design lives (based on the ABMA L-10 rating method) for continuously operated pumps typically range between 40,000 and 60,000 hours if good reliability is expected. If the operation of the equipment has to be extremely reliable or if the equipment is quite large, design lives of 100,000 to 200,000 hours are not unreasonable. Check with equipment suppliers where extreme reliability is required to ascertain that the specified design life can be provided.

16-13. Vertical Drive Shafts

An intermediate bearing is required and specified for the vertical drive shafts in Paragraph 11304-2.03H of Appendix C. If intermediate bearings are required, the design should provide both a rigid support for the bearing and convenient access for maintenance. The support can often consist of a reinforced concrete walkway that serves as lateral support for the wall (see Section 25-10).

16-14. Electric Motors

Standard electric motors less than 150 kW (200 hp) are normally covered by a separate specification. However, information such as voltage, power, speed, and enclosure type, plus all motor modifications including over-temperature devices and space heaters, should be specified in the specific equipment section.

Over-temperature or thermal protection for motors is often recommended for pumping units with variable-speed controllers to protect the motor against sustained overload conditions. Over-temperature sensing devices may be located either in the

motor starter or in the motor winding. Sensing elements specified in the motor windings provide a more accurate and faster indication of winding temperature and are more reliable than a device in the motor starter.

Space heaters (to reduce condensation in the motor enclosure), thermal protection, and other motor modifications should be specified only when the value of the motor warrants the cost of modification. If the motor has epoxy-coated windings, space heaters are not usually required, but if the windings are unprotected, space heaters are normally specified whenever the motor is located (1) outside (especially if, as a standby unit, it sits idle for extended periods), (2) where humidity is high for long periods, or (3) below grade in a damp atmosphere.

16-15. Optimum Efficiency

An analysis of capital and operating costs (including the cost of energy) shows that the present worth of wire-to-water energy efficiency can have a profound effect on selecting the most advantageous bid. A purchase specification (for a pumping unit) that includes an efficiency evaluation is given in Paragraph 11304-4.01 of Appendix C. The specification is simplistic, but more complete energy cost factors (such as maximum power demand, power factor, peak and off-peak rates, seasonal changes in rates, and other power-cost-modifying concerns) can be added, as well as expected inflation and power escalation costs. The calculation of such costs can be imposed on the bidder if applicable formulas and data are given.

Although high efficiency is desirable, it is less important than reliability—a characteristic that is difficult to quantify. Service records of similar pumping units are sometimes used as a basis for judgment.

16-16. References

1. *CSI Document MP-PRM2004, The Project Resource Manual - CSI Manual of Practice;* Module 5, *Construction Documents,* The Construction Specifications Institute, Alexandria, VA (2004).
2. *CSI Documents MP-2-2, SectionFormat* (1997); *MP-2-3, PageFormat* (1999); The Construction Specifications Institute, Alexandria, VA.
3. Dicmas, J. L., *Vertical Turbine, Mixed Flow, and Propeller Pumps,* McGraw-Hill, New York (1987).
4. Walton, W. C., *Groundwater Pumping Tests-Design & Analyses,* Lewis Publ., Chelsea, MI (1987).
5. *Modern Plastics Encyclopedia,* McGraw-Hill, New York (updated annually).
6. *Plastic Directory,* Cahner's Publishing Company, division of Reed Publishing, Newton, MA (updated annually), with a subscription to *Plastic World.*

Chapter 17

System Design for Wastewater Pumping

GARR M. JONES

CONTRIBUTORS
Zbigniew Czarnota
Gary S. Dodson
James C. Dowel
William R. Kirkpatrick
William H. Richardson
Michael G. Thalhamer

The thought process involved in, and a formalized approach to, the design of wastewater pumping stations are described in this chapter. Although several aspects are unique to wastewater pumping, the process is similar for all types of engineering design. Regardless of the size of the project, engineers should school themselves to follow the procedure described here, which is equally applicable to the very smallest as well as the largest projects. The only difference is in the formalities used to ensure that nothing is omitted from the thought process. Novices are encouraged to follow the process formally without regard to project size. Experienced engineers may wish to take some shortcuts in the formalized process—especially for small projects.

17-1. Organization and Control of the Process

The development of a well-organized plan to complete the work is fundamental to the design process. Last-minute changes in project direction (often leading to an unsuccessful project) can be avoided by first thinking through and formalizing the individual steps required to complete the assignment. Because there is no substitute for an orderly planning process, set up a procedure for managing the documentation of project information. The documentation plan should provide for the filing and identification of the following:

- Project management plan
- Quality assurance plan
- Correspondence to and from:
  - The owner
  - Equipment manufacturers
  - Information sources
  - Consultants
  - Regulatory authorities
  - Others affected by or affecting the project
- Calculations
- Notes of meetings
- Reference documents such as utility maps and "as-built" drawings
- Reports from consultants
- Reports/recommendations to the owner
- Design memoranda.

Several software programs are available to assist in organizing and locating project documentation. Favor those that can be used to alert all project participants to changes or additions, but restrict entries only to those authorized to make such changes. Document control and prompt distribution of information to all members of the project team are basic requirements for a successful project. Recognize that, in future weeks or years, others will need to refer to the project record for information. Under the best of circumstances, the need will occur because the project has been eminently successful and others may wish to discover the secrets of success. Under the worst of circumstances, the records will be needed because contractor or client claims or litigation have made it necessary to determine how the project went awry. Either way, if the task was performed properly, there should be no fear of exposure.

Calculations should always be accompanied by an explanation of their significance because, otherwise, their meaning will make interpretation in future years almost impossible. Indeed, a good set of calculations usually involves more text than numbers. Engineers come and go, and the one who starts a job may not be the one to finish—another reason to make calculations perfectly understandable by any likely reader. Some organizations ban the use of spreadsheet calculations or restrict such tools to authorized programs that have been subjected to rigorous quality checks because of the difficulty of checking this type of calculation tool. Others have standardized on Mathcad®, a proprietary software program that has numerous advantages in its presentation that facilitate error detection and checking.

Another important element is the quality assurance procedures. Protocols should be established for:

- Calculation format
- Checking the calculations
- Reviewing specifications and drawings
- Cross-checking between disciplines
- Peer review
- Final coordination reviews
- Final signing and sealing of contract documents.

A properly conceived and implemented quality assurance plan is the cornerstone of a successful project. If quality assurance is not considered and documented as an important facet, the result may be, at best, an inordinate number of costly change orders or, at worst, a disastrous failure.

Project events usually fall into three stages during the design process: (1) preliminary engineering, (2) detailed layout, and (3) detailed design. The general sequence of activities at each stage of design development is described in the following sections.

17-2. Preliminary Engineering

The purpose of preliminary engineering is to gather the information necessary to perform the design, to develop the basic concepts, and to obtain acceptance from all project participants. If the preliminary engineering has been properly executed, all of the information required to complete the detailed design will be within arm's reach. Additional information and details on preliminary engineering are contained in Sections 25-1 through 25-4.

Need for Pumping Stations

Even small pumping stations represent a substantial investment over the life of the facility—particularly true with wastewater pumping stations where structure size, excavation depths, and environmental considerations generally result in comparatively greater costs than those of water pumping stations of an equivalent capacity. Therefore, deeper sewers, tunneling or jacking through hills, alternate pipeline routes, and other strategies should be considered on a present-worth basis with generous allowances for the cost of operation and maintenance (O&M) of the pumping station before deciding whether to pump.

Site Selection

Frequently, an engineer has little choice in site selection for wastewater pumping stations because of sewer system hydraulics, political considerations, and available land. Some aspects of site selection could have profound implications with respect to the cost of the project. These include subsurface conditions, aesthetic considerations, and the route of the force main.

The concerns and considerations associated with evaluating and selecting pumping station sites are discussed in Chapter 25 and are briefly summarized in the following subsections.

Subsurface Conditions

The characteristics of the underlying soils at a site often require some specific method of construction. For example, a caisson may be required because of soft soils and high groundwater; thus, the configuration of the pumping station is virtually dictated by the caisson and, in turn, by the subsurface conditions. This situation is illustrated in the Duwamish Pumping Station (Example 17-1), where a caisson was used.

A caisson must be open at the bottom and top for excavating the material inside, so a cylinder (designed as a ring girder to resist wall loads) is more efficient than a box-type structure in which the necessary internal diaphragms or struts are obstructions. Once the structure has been sunk to the planned elevation, a tremie seal can be cast at the bottom. After curing, the structure can be dewatered and internal walls and slabs can be constructed. While this example is perhaps somewhat unusual in the United States, wastewater pumping stations are, by their nature, often deep structures. The location and thickness of floors and walls and the configuration of the structure are often determined by the external wall loads and buoyancy caused by both hydrostatic and passive soil pressures. Therefore, after a site has been selected, a comprehensive geotechnical investigation should have the highest priority.

Aesthetic Considerations

Most property owners agree that wastewater facilities are necessary and beneficial to the public good, but only when located "anywhere but in my backyard." As the sensitivity of sites to environmental conditions increases, provisions for architectural treatment, odor and noise control, landscaping, and similar concerns grow more costly. Pressure is often exerted to eliminate any visible presence of a pumping station, thereby requiring electrical and instrumentation equipment to be located in rooms below ground. Such an arrangement should be avoided (because of the potential for flooding) by selecting another site, if possible, or by appealing to reason. It usually requires many weeks to restore a station to operation after vital electrical and instrumentation equipment has been immersed in wastewater.

Force Main Route

While not specifically a site issue, the selection of the force main route may have an impact on station cost and may require consideration of a different location for the pumping station. The identification and evaluation of hydraulic transients is considered in Chapters 6 and 7. Control of transients in wastewater systems is more difficult than in water systems because of the concern over the reliability of air and vacuum release valves (although the advent of Vent-O-Mat® valves provides reasonably reliable air and vacuum relief) and surge control valves when applied to a fluid containing grease, grit, and rags. Still, the relocation of the station or the force main, or both, to avoid knees, intermediate high points, and plateaus that could result in column separation is a high priority. Additionally, force main length should be short to reduce dynamic headlosses, the production of malodorous, toxic, and corrosive gases, and cost.

Required Capacity

Because numerous texts [1, 2, 3] deal with the methodology for developing information on wastewater flows, the subject is not addressed here. It is sufficient to emphasize that an engineer must have reasonably well-developed projections of initial minimum, average, and peak flows before embarking on a design. Although predictions of future needs are often difficult, severe operational problems are likely to occur if initial loads are either under- or overestimated. The flow data must be competently developed and completely realistic. This information is essential when addressing issues such as:

- Initial and final number and size of pumps and motors
- Initial size of the force main and the size of a future parallel force main, if any
- Schedule for a staged installation of pumps, motors, and force mains.

Alternatives include: (1) installing minimum impeller sizes now and larger ones later, (2) installing two or three pumps now with provisions for more to be added later (see Example 12-2), (3) using small pumps initially with piping designed for larger pumps to be substituted later, and (4) installing a small force main initially with provisions for a larger one to be laid in parallel with the first and to be used alone in intermediate years (whereas both would be used at full development).

Wastewater Characteristics

In addition to quantity requirements, the characteristics of the wastewater must be understood. There is no such thing as "standard wastewater." If the wastewater system already exists, look for the following:

- Unusual quantities or size of gross solids. (Perhaps mechanical grinding or a screen will be required to protect the pumps, but try to pump the solids without screening if possible, because both screening and grinding equipment are notoriously maintenance-intensive.)
- Unusual grit quantities or sizes. (Alternate metallurgy, such as Ni-Hard or, in a worst-case situation, an upstream grit removal system may be required.)

- Corrosive constituents. (Special metallurgy or chemical conditioning systems may be necessary.)
- The potential for toxic, explosive, or flammable materials. (Monitoring systems and scrubbers for ventilation exhaust may be needed.)
- Septic wastewater and large quantities of hydrogen sulfide. (Special coatings in the wet well and variable-speed pumping to reduce residence time in the sump may be advisable.)

The above information is needed in the equipment selection process, in the station layout, and in other aspects of design.

Mode of Operation

The mode of operation of the pumping station—constant-speed (C/S) versus variable-speed (V/S), the size of the pumping equipment, and the type of pump driver—should be settled as soon as required capacities have been selected. Issues to be considered in making such decisions include the following:

- *Impact on receiving facilities:* Constant-speed pumps may discharge slugs into a treatment plant and, thus, upset unit processes.
- *Cost:* Compared with V/S pumping equipment, C/S equipment is simpler to design and maintain, and the first cost of the entire station is usually slightly less, but the energy used is usually greater. On a life-cycle cost basis, there is often no significant difference (see Example 29-1).
- *Production of odors:* Large wet wells and turbulence from slug discharges of anaerobic wastewater may generate and release copious quantities of malodorous compounds.
- *Pump capacity:* If too large, V/S pumps may be forced to operate in regions of unfavorable radial thrust, which risks the possibility of damage and reduces efficiency (see Sections 10-6 and 15-3). Start-up of large-horsepower electric motors for C/S pumping may induce unacceptable voltage fluctuations in the local electrical distribution system, although soft-start systems reduce the fluctuations greatly. AF converters and eddy-current couplers limit voltage fluctuations in V/S pumping to low values.

Preliminary Hydraulic Profile

A preliminary hydraulic profile should now be developed. It may not be necessary at this stage to refine the calculations, because they need be only accurate enough to ensure that equipment sizing and the layout for the preliminary system evaluations are valid for the final design. The following are some practical rules and shortcut hints for constructing a preliminary system head-capacity (H-Q) curve (hydraulic profile):

- In pumping stations (where pumps discharge through isolation and check valves to a common header), use 1.5 m (5 ft) as the headloss from the pump inlet to the end of the header.
- Use street maps or even U.S. Geological Survey quadrangle maps both for force main length and location and for a preliminary estimate of static lift.
- Construct a system H-Q curve envelope using a Hazen–Williams C of 110 to 120 at a low wet well level to estimate the maximum total dynamic head (TDH) and a C of 130 to 140 at a high wet well level to estimate the minimum TDH. Add 5% for fitting losses if the force main is 250 mm (10 in.) in diameter or less or if the force main is less than 1000 ft long. Note that the above values are suggested unless a fitting-by-fitting takeoff is made, in which case calculate the turbulence losses in fittings and pipe friction separately (see Section 26-2).

The profile should be constructed for the pumps, the force main, the receiving sewer or structure, and the influent sewer(s) to provide a clear description of the operating conditions at initial and projected future conditions that include the extremes of maximum and minimum flows. The pumping station must work in harmony with its upstream and downstream hydraulic elements. The potential for damaging hydraulic transients should be included (see Chapters 6 and 7). Some shortcut methods for computing the effects of hydraulic transients are sufficiently conservative (especially for wastewater systems) to permit proceeding with reasonable confidence (see Parmakian [4]). Shortcut analyses are often sufficient to determine if conventional check valves can be used or if powered pump stop-and-check valves must be used to control head rise at the pumps upon power failure. Where doubt exists, a detailed computer analysis should be made in the final design stage.

Preliminary Equipment Selection

The selection of pumps is discussed in Chapter 12. However, consideration should also be given to the type of driver when making a preliminary selection of

pumping equipment. Information on electric motors and internal combustion engines is contained in Chapters 13 and 14. In some circumstances, a combination driver (see Section 15-11) may be appropriate.

Select pumps that are true nonclog designs and pass solids at least 25 mm ($2\frac{1}{2}$ in.) in diameter. It is better to avoid the need for influent screening or grit removal whenever possible because the best place for these labor- and maintenance-intensive operations is at the treatment plant. Grit or screenings stored at a pumping station almost always produce prodigious odors.

Once preliminary equipment selections have been made from catalog data, contact the candidate equipment manufacturers' representatives and describe (1) the preliminary operating requirements and criteria, (2) the preliminary equipment selection, and (3) any decisions regarding driver selection. Request written confirmation and any suggested alternatives.

Power Supply

After the preliminary selections of the main pumping equipment have been made, estimates of total station power requirements can be developed. The project electrical engineer should produce a preliminary estimate of station service requirements and discuss the availability of power for the project with the local electric utility. Service characteristics, reliability, and special concerns such as limitations on starting loads should be obtained at this time (refer to Chapter 9 for more information).

Owner Preferences

By now, a preliminary concept for the station is taking shape. It is time to discuss the project with the owner and, particularly, the owner's operating staff. The purpose of the discussion is to develop an understanding of the preferences and internal requirements for the following:

- Types of equipment
- Standards for reliability (such as more than one standby pump and back-up power supply)
- Station amenities (such as restrooms, offices, and shop)
- General arrangement
- Access to equipment (such as provisions for cranes, monorails, and vehicles)
- Controls (such as location of switches and type of variable-speed systems—see Chapters 9, 15, and 21)
- Monitoring systems (such as types of indicating lights, vibration monitors, and overload detectors—see Chapter 20)
- Telemetry (see Chapter 20)
- Aesthetics (see Chapter 25)
- Odor-control system (see Chapter 23).

Regulatory Agency Requirements

Regulatory agency requirements affect the equipment selection and the design of a variety of station features. Regulatory agencies to be contacted may include:

- State and local water pollution control agencies
- U.S. EPA
- State health department, if different from the above state agency
- Building permit agency
- Planning commission
- U.S. Army Corps of Engineers
- Fire marshal
- Water utility
- Flood control district.

Public hearings, environmental impact statements, or use permits may be required. The subjects to be discussed (and resolved) with the regulatory agencies include:

- Power supply reliability and the need for on-site standby power
- Potable water supply and backflow prevention
- Flood protection
- Requirements for station reliability
- Ventilation standards, including odor control
- Fire protection
- Architectural standards
- Building permit requirements
- Local design codes
- Access and egress. Local setback requirements, and on-site dedicated parking for O&M crews.

Station Utilities

The next step is to conceptualize the design of the station utility systems. Equipment need not be selected at this time, but the basic approach for all of the station needs should at least be resolved. Issues to be considered include the following:

- Source of water for pump seals and domestic and housekeeping purposes (local utility or on-site well)
- Ventilation system (heating, cooling, odor control)
- Compressed air (type, capacity, quality standards)
- Drainage (destination and connector limitations)
- Internal drainage (destination and method of control)
- Standby power.

17-3. Detailed Layout

If the preliminary engineering stage (which is now complete) is done properly, the project engineer has all of the information and tools to put the project together. What remains are the details that will make it work successfully. A generalized description of the sequence of considerations for a typical wastewater pumping station project is presented in the following subsections. Circumstances unique to an individual project may, however, require a different approach. Careful and thorough preliminary engineering work can provide the guidance for the specific sequence from this point forward.

Piping and Instrumentation Diagrams (P&IDs)

First and foremost is the production of piping and instrument diagrams (P&IDs), which are the most useful type of schematics. Examples of P&IDs for pumping stations are illustrated in Figures 17-1 to 17-3 as well as in Figures 18-14, 21-5, and 21-6. The P&IDs are used to:

- Define the nature and control of each system
- Show piping sizes
- Locate piping devices, such as valves, pressure sensors, and meters
- Show control methods and strategies
- Specify the rated conditions and power requirements for all equipment.

The project leader, electrical engineer, and instrumentation engineer working together should develop the P&IDs. They are first sketched in pencil, then drafted when most details are worked out. Because many changes will occur as the project develops and because a P&ID drawn manually is initially so often large and unwieldy, it is advisable to use a CADD (computer-aided design and drafting) system for P&ID production and updating. Properly executed, P&IDs are useful for design in the following ways:

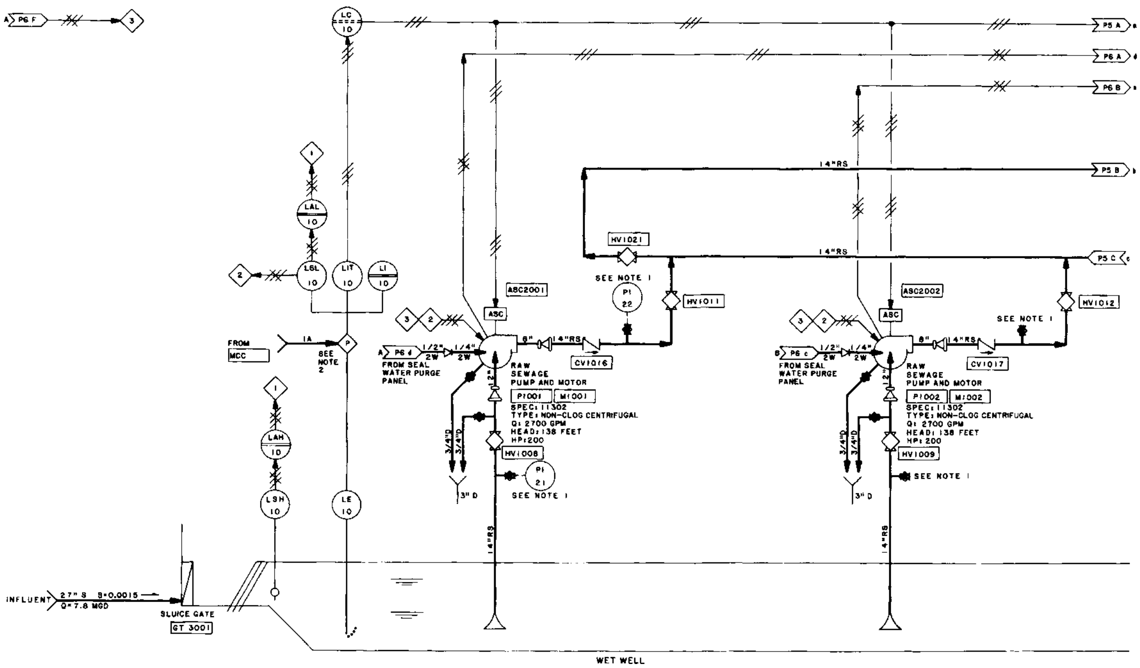


Figure 17-1. P&ID of main pumping units 1 and 2. Wastewater Pumping Station, Steilacoom, WA. Courtesy of Brown and Caldwell.

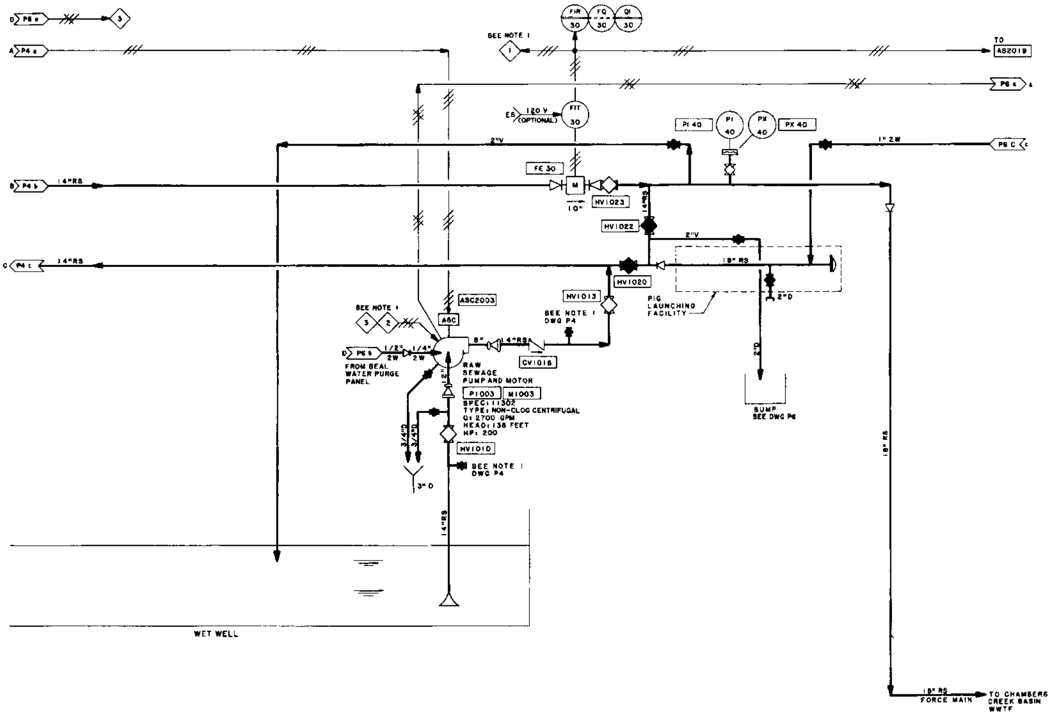


Figure 17-2. P&ID for main pumping unit 3 and metering system. Wastewater Pumping Station, Steilacoom, WA. Courtesy of Brown and Caldwell.

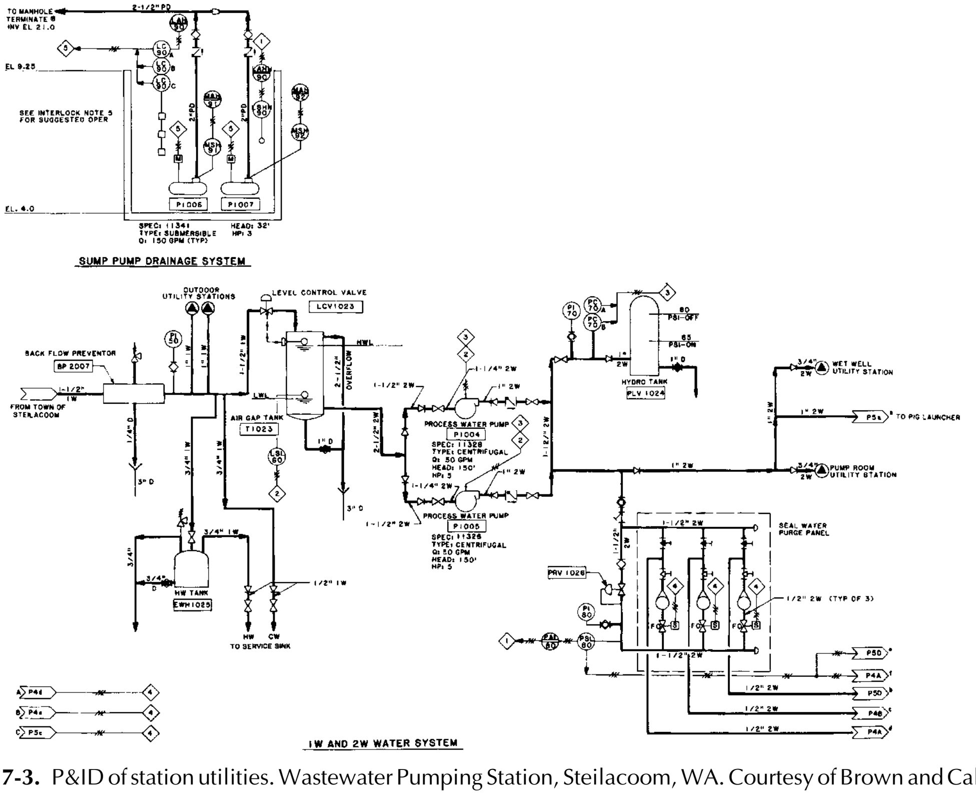


Figure 17-3. P&ID of station utilities. Wastewater Pumping Station, Steilacoom, WA. Courtesy of Brown and Caldwell.

- As the source of project information for producing detailed drawings and specifications
- For control of project information and changes
- For cross-checking and final document coordination
- As a means of communication between project disciplines.

P&IDs are also useful (1) as control documents during construction, (2) as aids in personnel training, and (3) for the operation of the facility.

Structural Considerations

At this stage in the process, recommendations from the project geotechnical engineer should be available to the project structural engineer. It is now time to make important decisions regarding construction methods. A detailed station layout is not necessary at this time, but a rough idea of station dimensions and loads is necessary.

Later in the detailed layout stage, the structural engineer should again be consulted about the thickness and placement of walls and slabs, although structural design should not commence until the next stage.

Sump Design

Extreme caution should be exercised in the sump design, particularly as it relates to inlet conditions. A poor design will very likely result in an inability to develop the intended capacity of the pumps, contribute to or cause severe damage to the pumping equipment, and result in shortening the useful life of the station. A comprehensive discussion of sump and pump intake design is located in Sections 12-3 and 12-5 to 12-8. The authors recommend improved self-cleaning wet wells for all wastewater pumping stations.

Details of some successful sump designs for large, moderate-size, and small pumping stations are shown in Sections 17-5, 17-6, and 17-7, respectively, but improved wet well designs as exemplified in Figures 12-3, 12-16, 12-23, and 12-31 should be used instead. The method of construction may influence the sump design, for example, as in the Duwamish Pumping Station where a caisson was used to construct the station substructure. A good sump and inlet design must have the following:

- Minimal turbulence and no influence from the incoming sewer, such as a cascade that might entrain air in the liquid and, therefore, into the pumps and force main. Avoid cascades preferably with an approach pipe (see Section 12-7) or, at worst, a drop pipe or a drop manhole. The entrance velocity into the sump should not significantly exceed 1.1 m/s (3.5 ft/s).
- Conservative values of NPSHA (see Figure 10-13 and text in Section 10-4) obtained by a suitable elevation of the pump relative to the LWL plus suction pipe velocities limited to about 2.4 to 3.4 m/s (8 to 11 ft/s) for keeping headlosses low.
- Suppression of vortices by (a) good geometry, (b) sufficient pump intake submergence as determined by Equation 12-1 (see also Dicmas [5]), and (c) the addition of cones under water pump intakes or flow splitters under wastewater pump intakes and fillets in corners to eliminate or reduce them. (See Figures 12-3, 12-14 to 12-16, and 12-19 to 12-21 for design details.)
- Adequate pump intake velocities for removing solids and, at the same time, for obtaining a good hydraulic environment for the pumps. See Table 12-1, but be aware that the author and editors consider 1.5 m/s (5 ft/s) to be a near-maximum velocity and not a target. All of the pumping stations in Sections 17-5 and 17-6 were designed for a maximum of 1.1 m/s (3.5 ft/s) intake velocity, and they have performed very well.

Pipeline Orientation

The relative positions (vertical as well as horizontal) of the incoming sewer or sewers and discharge pipelines can influence the station layout. However, avoid undue influence in this regard. The principal cost savings in the project are realized by making the pump room and sump layout as efficient as possible, by saving excavation and structure costs, and by the smaller cost of ventilation and lighting. Thus, for the price of a manhole or bend or two, significant cost savings in the pumping station can often be realized. Efficient pump room layout is a key factor in owner satisfaction.

Pump Room Layout

The efficiency of the pump room layout, as suggested previously, has a great effect on the overall cost of the project. Used in this context, *efficient* or *efficiency* means conserving room or structure plan dimensions without sacrificing optimum machine performance or personnel access for operation and maintenance of the equipment. The following are some guiding considerations:

- Use the room walls to support heavy valves and piping.
- Avoid piping configurations that impede access to pumps, valving, and other equipment (see Figure 12-33).
- Locate seal water valving and appurtenances, such as solenoids and pressure-regulating valves, rotameters, and isolating valves, on a wall adjacent to the pump and supply the seal water to the pump in copper or stainless-steel piping encased in the pump room floor slab. This arrangement avoids clutter around the pump and improves maintenance access.
- Never connect a wastewater pump discharge to a manifold or header from underneath. To do so invites the plugging of check valves and piping. Make the connection at the side of the manifold as shown in Figures 12-36 and 12-38. To avoid plugging check valves and the piping above them, locate check valves in horizontal pipes—never in a vertical pipe.
- For simple lift stations where the final discharge is not fully submerged, use a flap valve in a structure at the receiving sewer, which thereby avoids the space-consuming isolation and check valves (see Figure 12-37).
- Provide adequate clearance, such as 0.9 to 1.1 m (36 to 42 in.), from the outside edge of all piping flanges or other projections—not just the edge-to-edge of pump bases or pipe shells—on at least three sides of each pump and any discharge valving. Space is needed for a crew, not just a single worker.
- Provide enough room to remove bolts from the thrust harnesses (if used) of sleeve couplings and to slide the coupling off the joint.
- Provide a quick, unobstructed exit for people working around the pumps.
- Don't forget ventilation. Be sure that air can be moved through the room efficiently to prevent the accumulation of odors and moisture condensation and to scavenge gases from the space adequately. Try to exhaust air from a location near the drainage sump where the odorous gases tend to accumulate. Locate the fresh-air intake away from the exhaust and on the prevailing upwind side of the structure.

Some of these principles are illustrated in Figures 12-35 through 12-39 for different pump room layouts. Details of good piping layout are also shown in Figures 26-7 and 26-8.

Superstructure and Interior Spaces

The pump room layout should be developed with due consideration for other portions of the station. All spaces must work in harmony and must properly accommodate the function of all systems. Considerations in pumping station layout include the following:

- Access to and from all spaces for both personnel and equipment.
- Intermediate floors in deeper pumping stations can provide internal diaphragms to resist heavy wall loads and are excellent locations for utility systems such as water and compressed air equipment.
- Ventilation of all spaces. Now is the time to begin the preliminary sizing of ductwork and fans. If these features are left to the last, there will not be enough room and you may have to start all over.
- Location of utilities and support equipment. Some practical rules follow:
  - Water pumps, hydropneumatic tanks, and similar equipment can be located in the pump room and other spaces below grade.
  - Avoid locating fans in the pump room or in other areas below grade that are more prone to flooding.
  - Locate utility stations for housekeeping as close as possible to entrances to rooms below grade.
  - Do not provide washdown water services to engine rooms and electrical equipment (except motor) rooms.
- With the project electrical engineer's assistance, locate the motor control center and station switchgear in a roomy, dry area in the station superstructure. This equipment requires a source of clean, uncontaminated air and good air circulation. Cooling may be necessary. Be sure to be generous when estimating space requirements for electrical equipment. Add, say, 50 or even 100% to the space estimated by the electrical engineer. As the project progresses toward completion, new electrical loads will be added and initial load estimates will probably be too low. Also, plan for major conduit and cable tray runs to allow adequate provision in slabs or in passageways.

Architectural Treatment

Now that the station configuration and dimensions are roughly defined, consult the project architect about exterior treatment for the superstructure, site access, landscaping, and so on. These discussions should lead to final decisions on roof treatment, siding materials, and such other features as noise (see Chapter 22) and odor (see Chapter 23) suppression (see also Section 25-3 and Figure 25-1). The architect must bear in mind the clearances required for equipment operation and removal and maintenance access.

Preliminary Sketches

With the foregoing completed, the engineer and architect can now produce preliminary sketches showing the location and arrangement for all station features. These should be produced to scale, although not in great detail. The purpose of this step is to provide a basis for discussion with the various discipline leaders. After review, the preliminary sketches are revised for use in the design report.

Design Report

Using the information and materials developed thus far, a draft design report is prepared. Included in the report are (1) the purpose of the project, (2) the design criteria, (3) illustrations of the proposed and alternative arrangements, (4) features of the pumping station, and (5) rough estimates of capital and operating costs. The intent is to produce a concise basis-of-design document for all project participants.

The draft design report is submitted to the owner as well as to the funding and appropriate regulatory agencies for comment. After receiving their comments, the engineer should meet with interested parties to discuss the comments and to seek agreement on revisions, if any. The design report is then revised and issued as the guidance document for final design.

17-4. Detailed Design

Once detailed design is authorized, the production of final construction documents can begin. By the time the work in this stage commences, the site has been selected, the property presumably has been acquired, the force main route is assured and rights of way (both temporary and permanent) are being secured, required permits are being obtained, and all legal impediments including public hearings are being resolved. But the project is not ready for drafting just yet. Before the final drawings can be prepared, the following steps must be completed.

Refine Hydraulics

Sufficient information is now available to refine the preliminary hydraulic calculations completed earlier. Calculate NPSH curves for all of the critical conditions in which altitude, wastewater temperature, and atmospheric pressure are considered. Produce accurate minimum and maximum system H-Q curves for new and old piping. Make or authorize a rigorous hydraulic transient analysis if the shortcut methods used earlier indicate one is necessary. Be sure to develop neat, legible calculations for all of the anticipated operating conditions.

Finalize Equipment Selection

Once final performance requirements are known, the draft specifications describing the operating conditions, the performance requirements, the required construction features (such as materials), and so on should be produced for the major pieces of equipment. Copies of these should be furnished to representatives of the candidate manufacturers for their comments and suggestions. Equipment manufacturers who should receive draft specifications include:

- Pump manufacturers
- Motor manufacturers
- Engine manufacturers (for engine drives and standby generators)
- Makers of other important equipment packages.

In addition, final system calculations and equipment selection and control strategies should be developed for all peripheral and utility systems.

Revise P&IDs

Next, the P&IDs are revised to their final form, in which all monitoring and control elements, rated conditions for equipment, and power requirements are shown. The revised P&IDs can now be used as control documents for completing the construction documents.

Detailed Layout Sketches

Before preparing the final documents, the layout sketches are revised to show the final dimensions, including maintenance space requirements, code clearances, and final equipment outlines. By this stage, no more than 30 to 40% of the total design project effort should be expended.

Final Production

At this point, production of the final documents can begin. If the foregoing process has been followed with

no significant revisions, the project should be ready for final production. All that remains is to follow the original plan and to coordinate the efforts of the various disciplines.

From now on, the project leader acts as a coordinator who makes certain that each member of the team performs according to plan at the proper moment and that all members are aware of project events. The quality assurance plan should be brought into full effect. Communication and cross-checking between disciplines is of utmost importance. Final specifications (see Chapters 16 and 28) should be developed at the same time that final drawing production begins. As the project reaches the 75% completion stage, interdisciplinary cross-reviews for both drawings and specifications should be well under way (see Sections 9-12 and 13-15, Appendix E, and, by all means, Chapters 24 and 27).

Final Cross-Check

If the project has followed an orderly and carefully thought-out plan, the final cross-check should not reveal any significant discrepancies. Regardless of project history, however, there is no substitute for an independent final detailed review by an experienced senior-level engineer.

The concept is simple. A cross-check consists of checking drawings against specifications and discipline against discipline to discover inconsistencies. The purpose is not to second-guess the design decisions; those are considered given. Rather, it is to uncover any areas where one discipline is not properly coordinated within itself or where interdisciplinary communications have not been adequate. As a rule, at least an hour per drawing is required if coordination has been reasonably good. Well-coordinated documents require about half as much time, and poorly coordinated documents may require as much as five times as much effort plus commensurate efforts for document corrections.

17-5. Examples of Large Lift Stations

A great deal can be learned by studying the following examples of pumping stations selected to illustrate different problems and approaches to their solutions. All work well, meet their design objectives, and are neat, clean, and—even those in service for 15 to 25 years—have the appearance of new stations.

All of the pumping stations in Sections 17-5 and 17-6 were designed to minimize deep excavation by using variable-speed pumping systems. The ventilation systems meet the recommendations in Section 23-2, in which air is powered in near the ceiling and powered out from the lowest level to produce a slight vacuum in the wet well and ensure a safe, corrosion-free environment for personnel and equipment. Ease of access by operators and maintenance workers was obtained without unduly increasing the size of the structure by following the precepts embodied in Figures 12-36 and 12-38.

Examples 17-1 and 17-2 have draft tube pump inlets somewhat similar to the U.S. Army Corps of Engineers Type 10 intake (Figure 9.8.3 in ANSI/HI 9.8-1998). Ordinary isolating valves (eccentric plug valves in smaller systems) become massive and costly for suction piping larger than 400 mm (16 in.) in diameter. Those big, expensive valves can be replaced by inexpensive sluice (or slide) gates. The configuration allows the pumps to be placed close to the floor of the pump room. In comparison with ordinary suction piping layout (as in Figures 12-38 and 12-39), the pump support is far more rigid, and susceptibility to vibration and earthquake forces is greatly reduced. Access to machinery is improved by the comparatively low mounting.

Example 17-1
Duwamish Pumping Station

The substructure of Duwamish Pumping Station (see Figures 17-4, 17-5, and 17-6) was constructed as a caisson because of the poor quality of soils underlying the site and concern that conventional open-cut construction, no matter how carefully accomplished, would result in the settlement of nearby structures. Concrete for the substructure, 24.4 m (80 ft) in diameter and approximately 15.2 m (50 ft) deep, was placed in three lifts with the structure built at existing grade. After the concrete had cured, dunnage (cribbing) under the cutting shoe was removed, and the caisson was sunk into position within two days by excavating from within the structure. When the tremie seal was poured, the caisson was less than 25 mm (1 in.) below the planned elevation and only 0.25 degrees out of plumb.

Each pump in this simple lift station (which was built in 1968) discharges through approximately 21 m (70 ft) of dedicated piping to individual flap valves in a structure at the receiving sewer. Expensive and space-consuming large-diameter isolating valves and check valves are thereby avoided. Each of the three 900-mm (36-in.) pumps is rated $2.2\,m^3/s$ (50 Mgal/d) at a total head of 5.33 m (17.5 ft). The V/S pumps, which maintain normal levels in the upstream sewers, are driven by wound-rotor motors operating through a liquid rheostat for speed adjustment. The maximum speed of the pumps at the rated performance requirement is approximately 300 rev/min. To reduce the cost of both structure and equipment, the pumps were specified with 1200 rev/min vertical motors to be mounted on specially designed parallel shaft gear reducers. An integral backstop prevents reverse rotation on pump shut-down. Each pump has a draft-tube-type inlet conduit fitted with a vortex suppressor. Each pump inlet can be isolated by means of a fabricated steel slide gate placed using a motorized hoist on a monorail. A hydraulically operated sluice gate, which is tripped by power failure or by float switches in the wet well and pump room, protects the station against flooding.

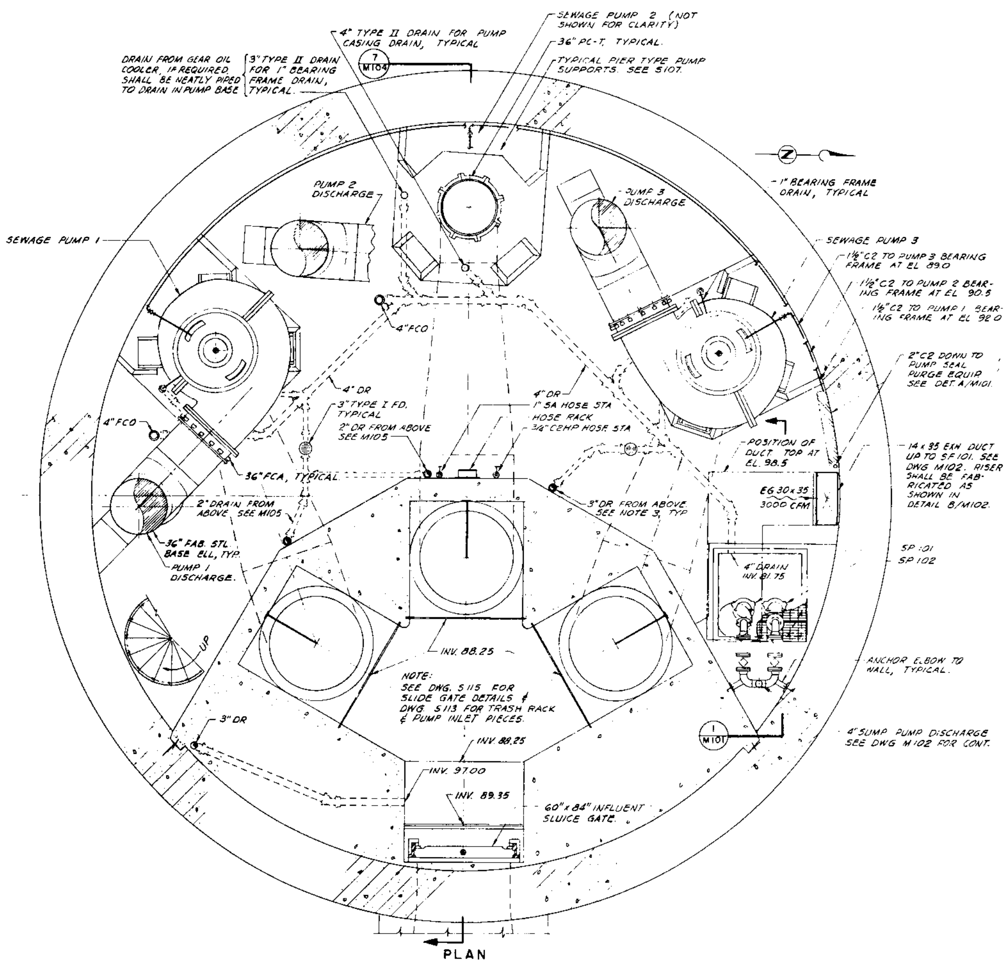


Figure 17-4. Pump room plan. Duwamish Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

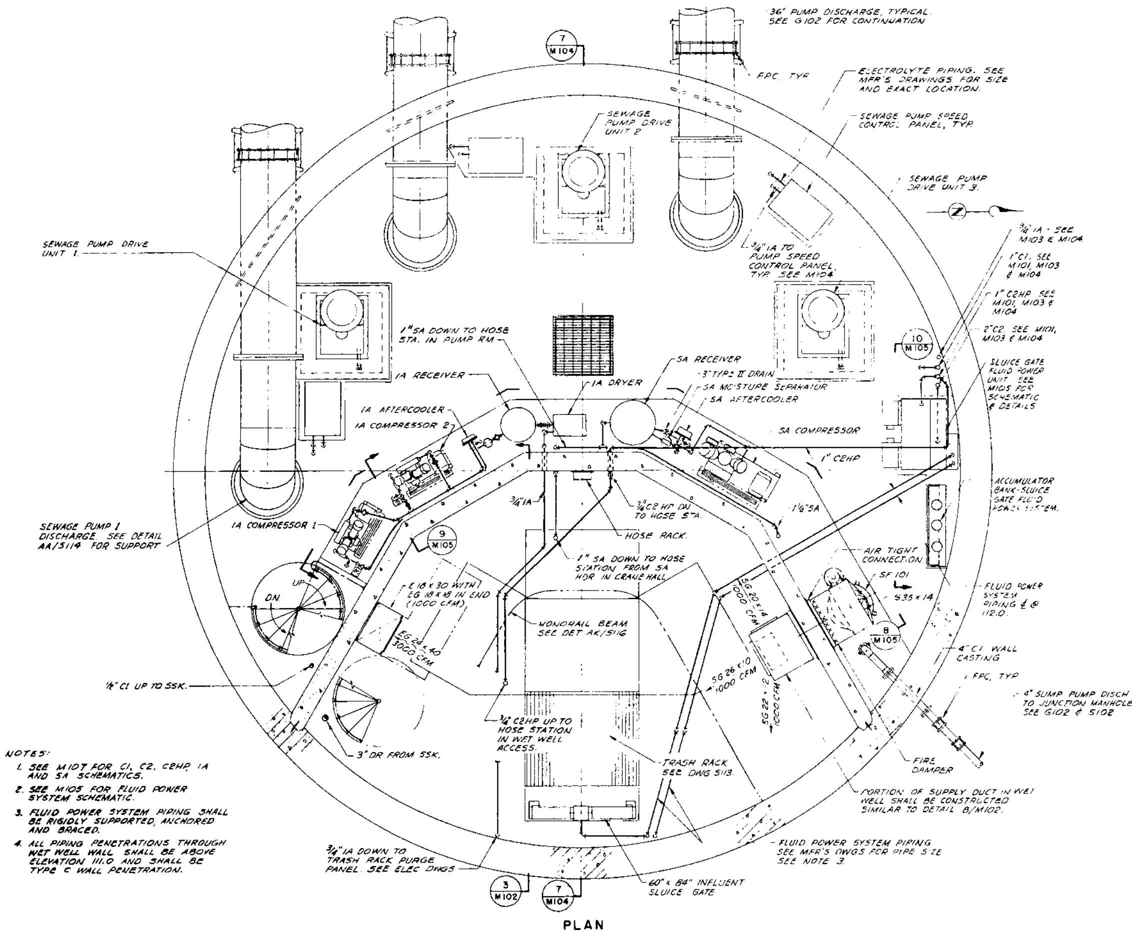


Figure 17-5. Equipment room plan. Duwamish Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

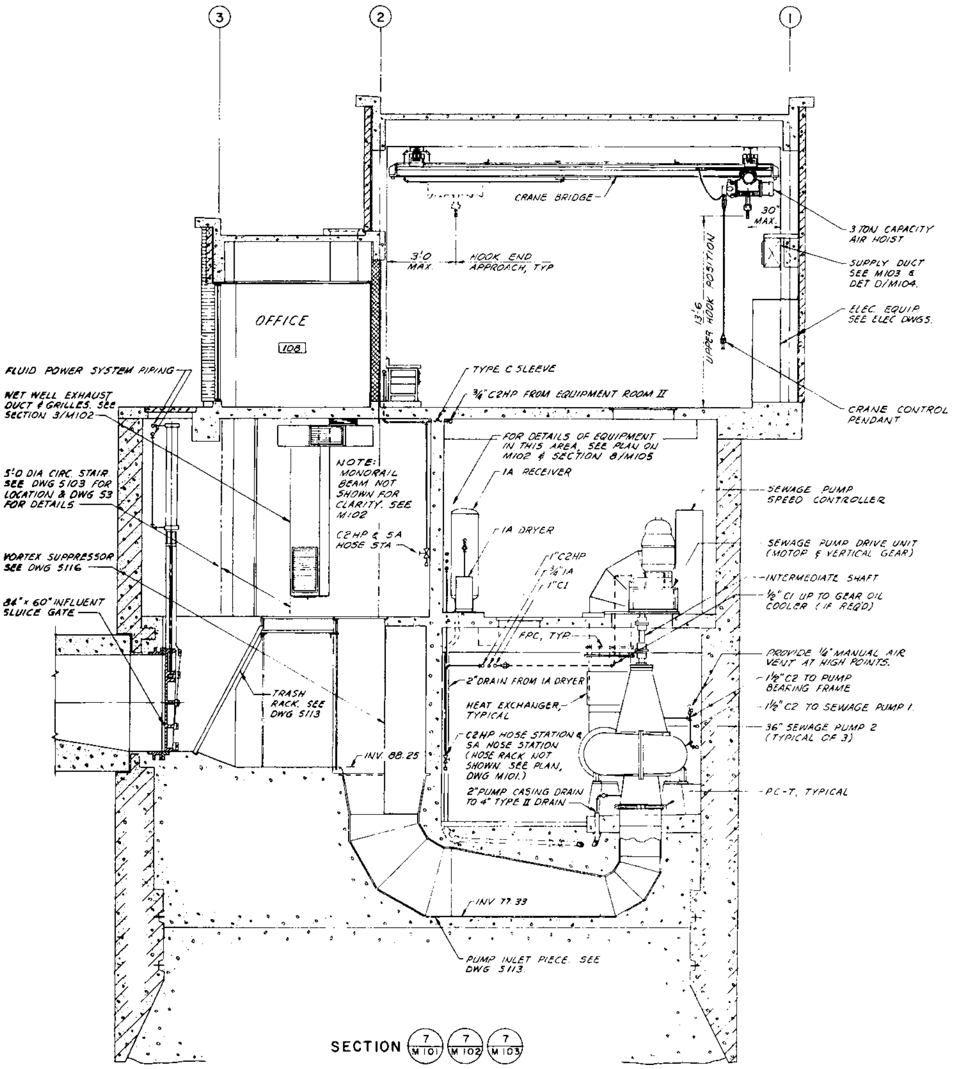


Figure 17-6. Typical section. Duwamish Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

Example 17-2
Interbay Pumping Station

The Interbay Pumping Station (shown in Figures 17-7 through 17-9) was constructed in 1966 on a site in an ancient saltwater embayment. The underlying soils were quite soft, and the groundwater table reflected fluctuations in tidal level in an adjacent harbor. The site was very constrained, with important structures (some of them quite fragile) nearby. Because soils at the base of the structure lacked sufficient supporting strength, piles were favored over a caisson. The construction procedure was to (1) drive a sheet pile cofferdam, (2) excavate the site, (3) drive the pile foundation with the cofferdam flooded, (4) pour a tremie seal at an elevation below the planned foundation slab, (5) dewater the cofferdam after the tremie concrete had cured, (6) cut off the piles, and (7) construct the foundation slab. General dewatering at the site, which would have compressed the local soft soils and severely damaged many nearby structures and surface improvements, was thereby avoided.

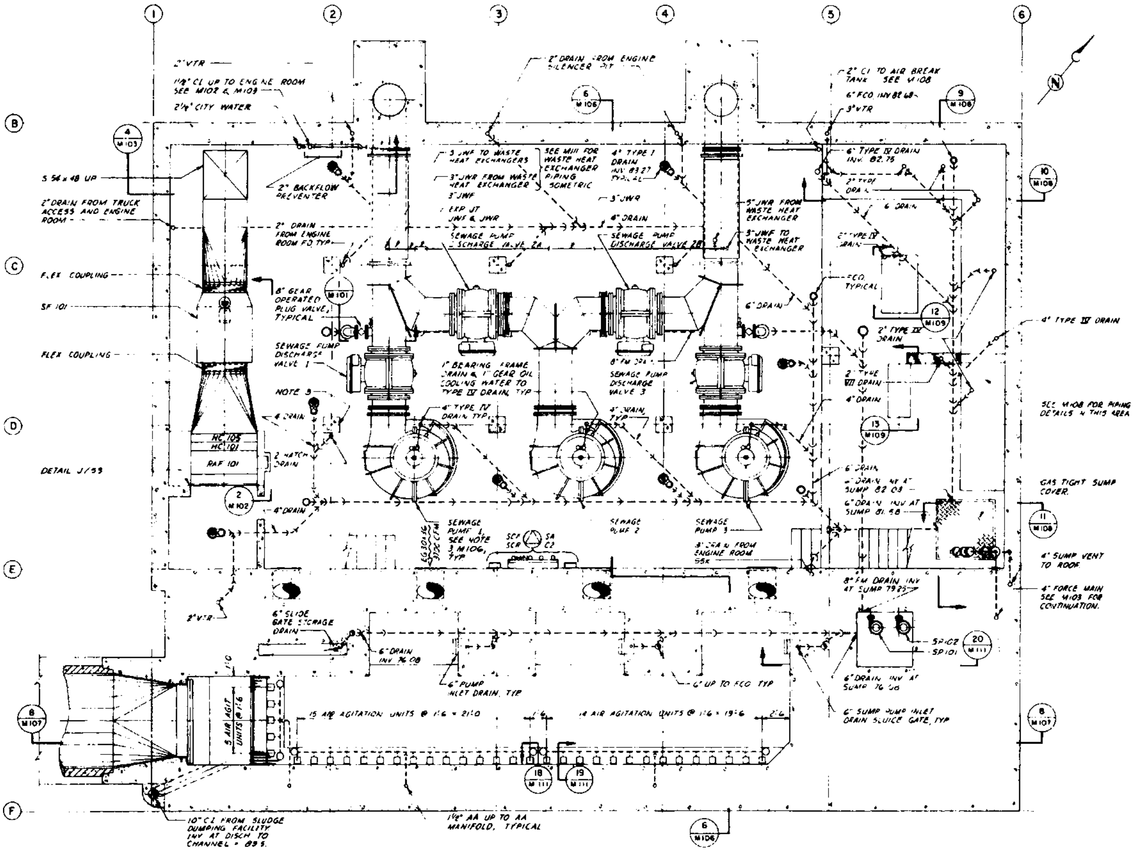


Figure 17-7. Pump room plan. Interbay Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

Each of the three 900-mm (36-in.) pumps is rated to deliver 2.6 m^3/s (60 Mgal/d) against 12 m (39 ft) of total head at a maximum pump speed of 360 rev/min. The engines are rated for 370 kW (500 hp) at 820 rev/min. The twin 1200-mm (48-in.) force mains are each approximately 1100 m (3600 ft) long. Almost all of the static head in the system had to be gained at the pumping station itself because of construction difficulties associated with the elevation of nearby bridge pier foundations. On the basis of a transient analysis, column separation was likely to occur in the force mains at the pumping station on loss of power at virtually any flow—even when all reasonable measures were taken to avoid it. Mitigating means include (1) engine drives with dual fuel sources to provide a higher degree of reliability, and (2) check valves installed on a branch of the force main at the probable point of column separation. These check valves act like vacuum-relief valves (see Figure 7-2c) to admit a column of air that cushions the rejoining of the two columns of water. Hydraulically operated, slow-closing pump stop-and-check valves on the pump discharges permit relief of pressure rise when the pipeline repressurizes on the return wave.

To maintain normal depth in the influent sewer, the pumps operate at variable speed through control of the engine throttle setting. Engine fuel is natural gas with a liquid propane reserve supply. The propane is vaporized at the engine using the heat available in the engine's jacket water. A hydraulically operated sluice gate, triggered by float switches in both the wet well and pump room, protects the station against flooding.

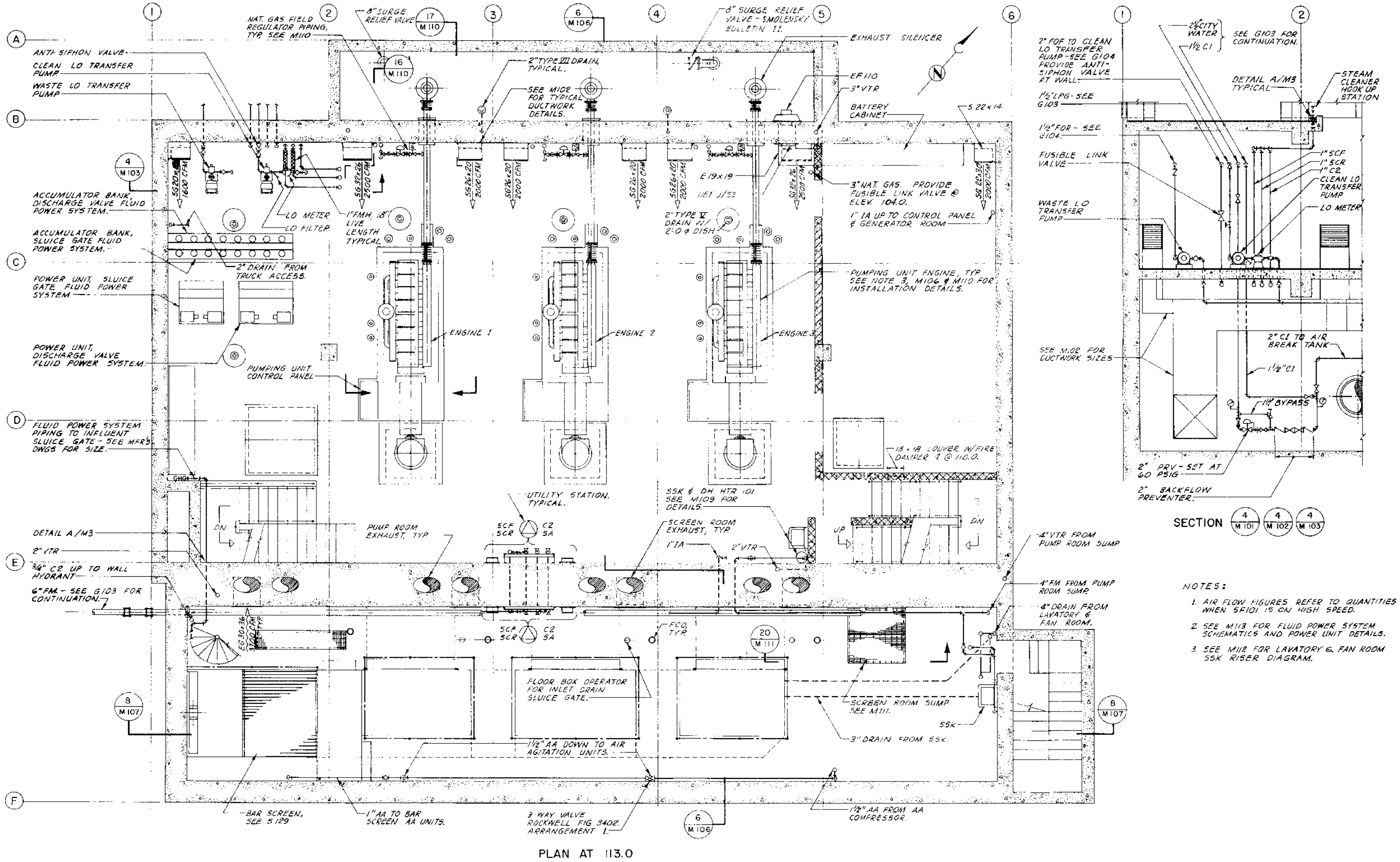


Figure 17-8. Engine room plan. Interbay Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

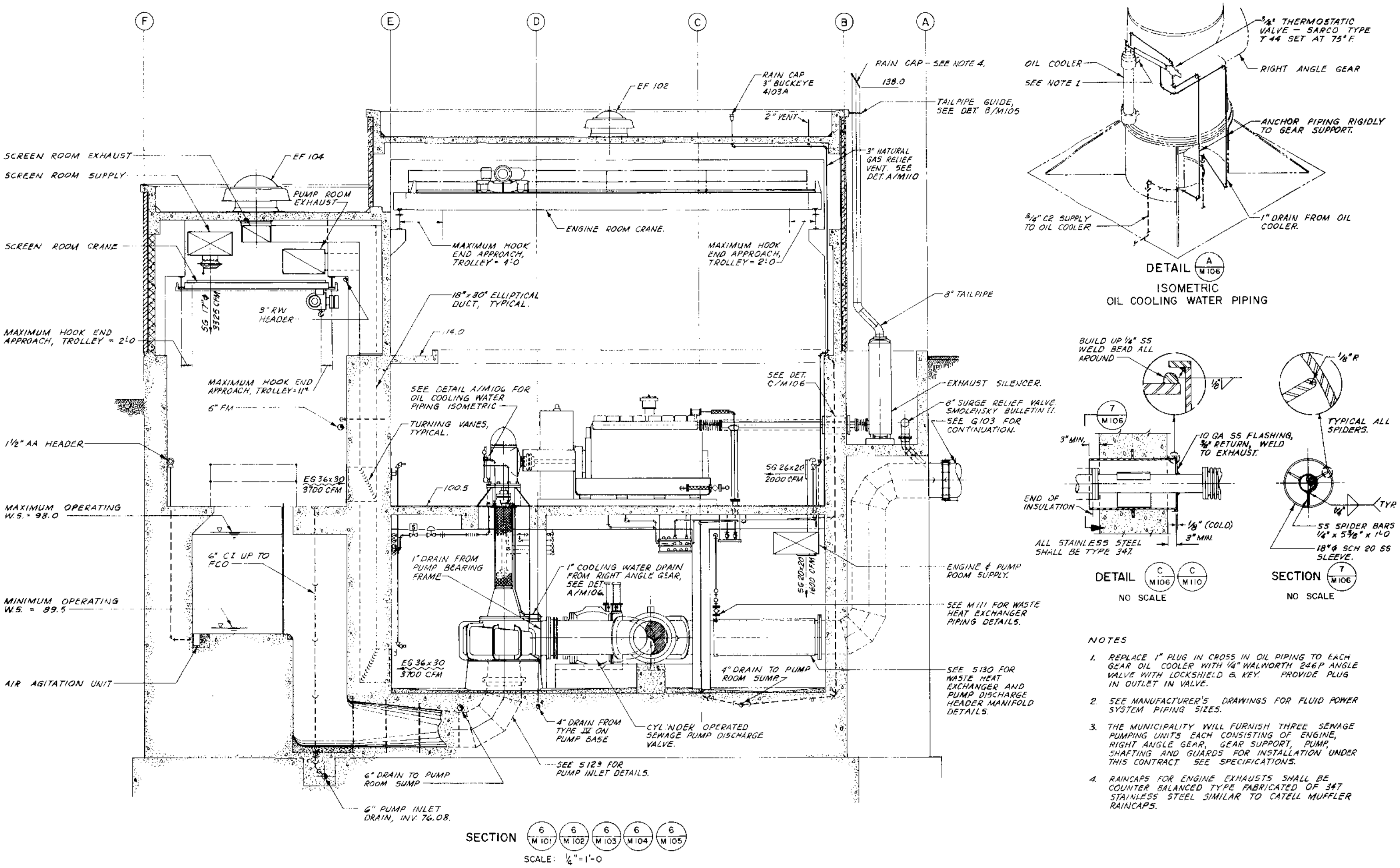


Figure 17-9. Typical section. Interbay Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

In 1998, some modifications were made to the wet well to avoid some surface vortices, probably caused by a bend in the influent conduit close to the wet well. Note that it takes at least five (eight is much better) pipe diameters of length to counter effects caused by bends or other flow-disturbing geometries. However, another pumping station with "chimneys" leading to draft tubes developed strong surface vortices that persisted through the draft tubes. The vortices were eliminated by adding a submerged, inclined "roof" over the chimney to interrupt the vortices. Model tests showed that a grating (inclined to allow solids to slide off) would also eliminate the vortices, but it made the draft tube inaccessible. Today, the size of the pumps would make model tests mandatory, and the tests would make the problems apparent.

Alternatives to FSIs

Formed suction intakes as illustrated in Examples 17-1 and 17-2 are often necessary for large pumps when the size and cost of large fittings and valves become excessive. There are alternatives, however, for large pumps that both solve the intake piping issue and eliminate the dry well altogether. At least one manufacturer of large column type solids-handling pumps has designs for units of this type as large as 4400 L/s (100 Mgal/d), and one manufacturer of submersible wastewater pumps has designs rated for 2800 L/s (65 Mgal/d). Heads at these capacities are generally limited. The major advantages of both types is the elimination of the dry well and expensive intake pipe fittings, but both types have disadvantages, and some of the more serious ones are listed below.

Column Type Solids-Handling Pumps

- The design must allow access for heavy mobile cranes to remove these pumps for maintenance or repairs. Room must be set aside (1) to allow access for a suitable flatbed truck-trailer and (2) to lay the pump down for disassembly or to lay disassembled sections on the ground, because sometimes it is necessary to disassemble the pump in sections as it is lifted from the wet well.
- Estimate the expected frequency and the cost per incident of pump removal with the aid of the manufacturer and users of these pumps. Evaluate the present worth of the costs and compare it with the cost of maintaining dry pit pumps plus the cost of the dry pit to determine which type of station is advantageous.
- Suction bells at the end of long columns move sideways as much as 75 mm (3 in.) with the hydraulic forces acting on the column.

Submersible Pumps

- Large submersible pumps are heavy–as much as 6 tons or more. Either (1) allow access by a large mobile cane and a flatbed truck or trailer suitable for transporting the machine or (2) include a suitable, heavy duty fully motorized crane for lifting the pump from the wet well and placing it on a flatbed truck or trailer.
- Power cables for high horsepower submersible pumps can be heavy and difficult to mainpulate. Handling them is not the easy task described in Section 25-6. Motorized cable reels are often necessary to ease the burden of removal and resetting the equipment, especially for 480-V cables. Cables for 4160-V are smaller and might be handled by the "elevator cable" method, but some utilities may object to the higher voltage.
- At least two submersible pump manufacturers have flange fitting details that prevent significant leakage when the pump is in operation. Leakage from pumps lacking an effective seal has been reported to be as much as 15% of the pump's flow rate, and 5% leakage is not uncommon. Leakage increases with head. Guard against excessive leakage (or at least be aware of it) by requiring pump tests to include both pump and discharge fitting where the test unit exactly duplicates the field installation. Leakage can be qualitatively observed when the water level is only a few inches above the jont. Dye injected into the pump suction or a short piece of yarn attached to a probe helps.

17-6. Examples of Medium-Size Lift Stations

As in Section 17-5, all of the pumping stations described in this section work well, meet the design

objectives, and are neat, clean, and virtually odorless—even after 15 to 40 years of service. All were designed to minimize the size and depth of the wet wells by incorporating V/S pumps. All have ventilation systems that meet the recommendations in Chapter 23. Air is blown in near the ceiling and removed by a powered exhaust system from the lowest level to produce a slight vacuum (about 6 mm of water column) in the wet well to ensure a safe and corrosion-free environment for both equipment and personnel. Note that ease of access, adequate room for maintenance and repairs, and a minimum size of footprint were obtained by following the equipment layout shown in Figures 12-34b, 12-37, and 12-39.

Examples 17-3 and 17-4 have trench-type wet wells designed before the advent of the ogee ramp, so they are cleaned by water coursing along the trench at only subcritical velocities. Consequently, not all sludge is removed, and there is an average residual depth of about 50 mm (2 in.) of hard sludge remaining after cleaning—a depth that could be reduced somewhat by modifying the operating procedure. (All scum, incidentally, is removed.) Nevertheless, the cleaning is effective in suppressing odors.

The Kirkland Pumping Station appeared to be one of the most typical or representative of the Seattle-area stations with the original trench-type wet well design. Thus it became the prototype for the model studies described in Chapter 12.

Example 17-3
Kirkland Pumping Station

The Kirkland Pumping Station (shown in Figures 12-8, 17-10, and 17-11) was constructed in 1965 in the downtown area of a small city on the outskirts of a large metropolitan area. The station replaced a wastewater treatment plant as a part of a regionalization project. Because the incoming sewer would be only a few feet below the existing grade, horizontal pumps were selected over vertical units to keep the station profile as low as possible. By chance, the owner had two pumps (surplus from an abandoned station) equipped with two-speed motors that could satisfy project requirements. The existing impellers were replaced to improve solids-passing capability. The second (lower) motor speed, however, was too low and, therefore, was not used. Instead, the pumps were converted to V/S operation by means of eddy-current couplings. A duplicate third pump was powered with a new single-speed motor to complete the main pumping units for the new station.

To avoid any outward indication of the station's purpose, the large transformer and high-voltage switchgear were enclosed within the station's superstructure. Each pump delivers 0.11 m^3/s (2.5 Mgal/d) against a rated head of 57.6 m (189 ft) at 1750 rev/min. The suction bells are 400 mm (16 in.) in outside diameter. At maximum flow for one pump, the intake entrance velocity based on the OD of the bell is 0.85 m/s (2.78 ft/s) and the suction pipe velocity is 2.1 m/s (6.8 ft/s).

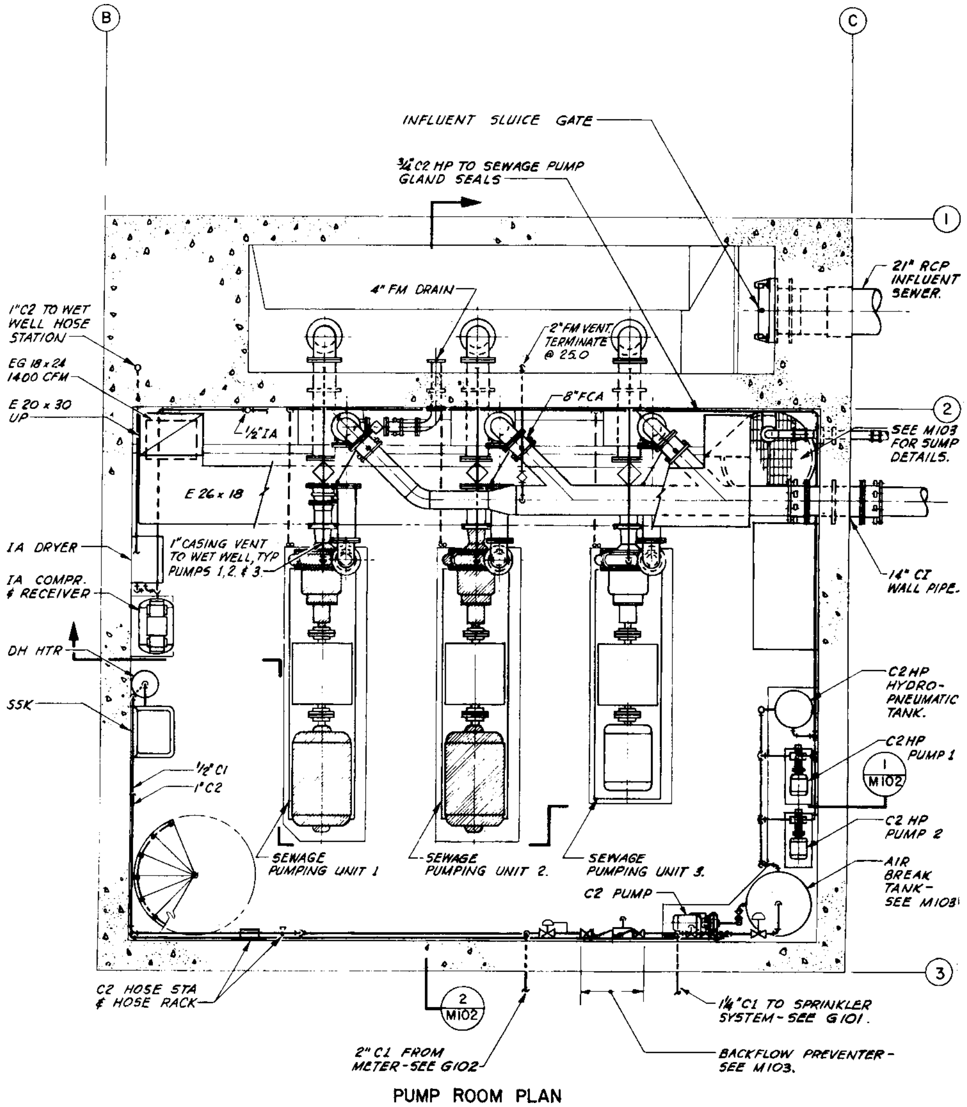


Figure 17-10. Pump room plan. Kirkland Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

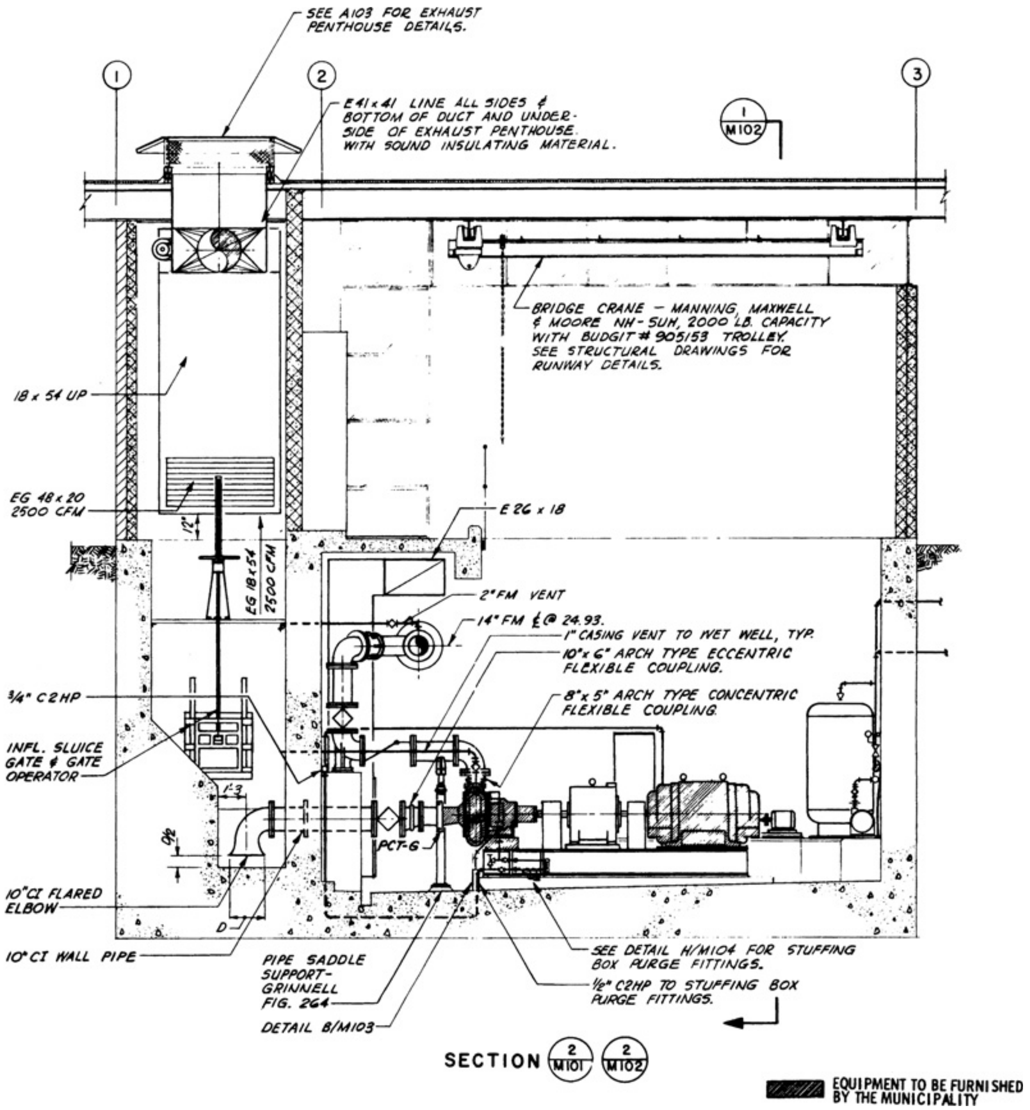


Figure 17-11. Section. Kirkland Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

Critique of Kirkland Pumping Station

The wet well is covered by a light grating that must be removed section by section for washing grease off the walls. Gratings were omitted in some of the other Seattle-area pumping stations in favor of a walkway beside the wet well for easy access for hosing walls—a feature much preferred by the operators.

The walls of the wet well were coated with coal-tar epoxy. After 30 years of service, the coating began to peel. Lining with PVC is much more expensive, but linings last far longer. Of course, it is also expensive to renew coatings in an operating wet well.

Observation of the cleaning procedure at Kirkland and the water splashing from the inlet pipe onto the trench floor, losing energy that could be used to create supercritical velocities along the trench, prompted Sanks [6] to reflect that a curved ramp from inlet to floor would make cleaning more effective. From the model tests described in Chapter 12,

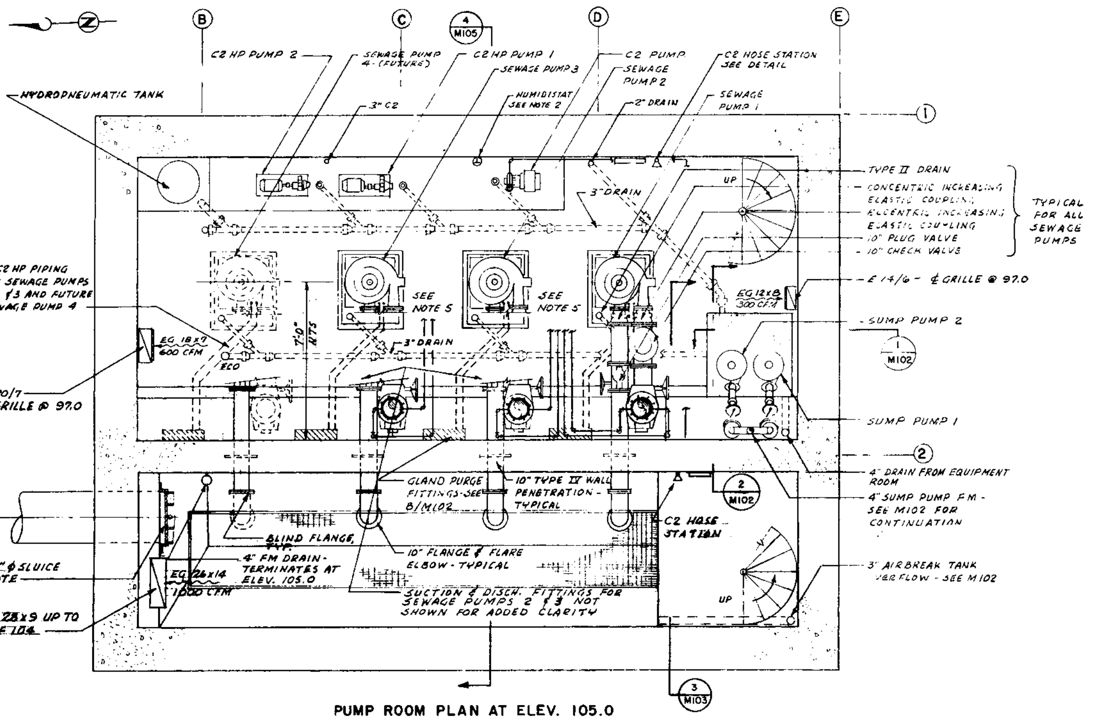
the cleaning was 50 to 100 times more effective as measured by the time required to remove all solids. Regardless of whether time is the means used to rate cleaning effectiveness, the curved ramp is a tremendous improvement.

The Kirkland Pumping Station serves as an outstanding example of the importance of good maintenance. Aside from some areas in the wet well where the coating has peeled, the station has the appearance of one built only a year or so ago.

Example 17-4
North Mercer Island Pumping Station

The North Mercer Island Pumping Station (Figures 17-12 through 17-14), built in 1969, is exceptionally deep (13.7 m or 45 ft). It is located in a quiet residential neighborhood where aesthetics (architectural treatment and control of odor, sound, and light emissions) all were of concern. The station had to have the appearance of a modern ranch house to match nearby structures. The typical cyclone fence, so common to most municipal wastewater pumping stations, had to be avoided at all costs. A small stream, which coursed through the site, had to be preserved. The sylvan character of the site, if anything, had to be enhanced by the proposed improvements.

Each of the pumps is rated 1.1 m^3/s (2.6 Mgal/d) against a total head of 43 m (140 ft). The force main route, the only economically feasible selection, featured a plateau at approximately the middle half of its 760-m (2500-ft) length. This plateau was at an elevation that corresponded roughly to two-thirds of the total static head. From a transient analysis, it was determined that column separation was likely to occur upon power failure at maximum pumping rates. The solution was to specify the pumps to be driven by wound-rotor, specially designed motors incorporating flywheels that have rotating moments of inertia sufficient to prevent column separation under the worst conditions. The motor speeds are controlled by a liquid rheostat and adjusted to maintain normal depth-of-flow conditions in the upstream sewer.



PUMP ROOM PLAN AT ELEV. 105.0

Figure 17-12. Pump room plan. North Mercer Island Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

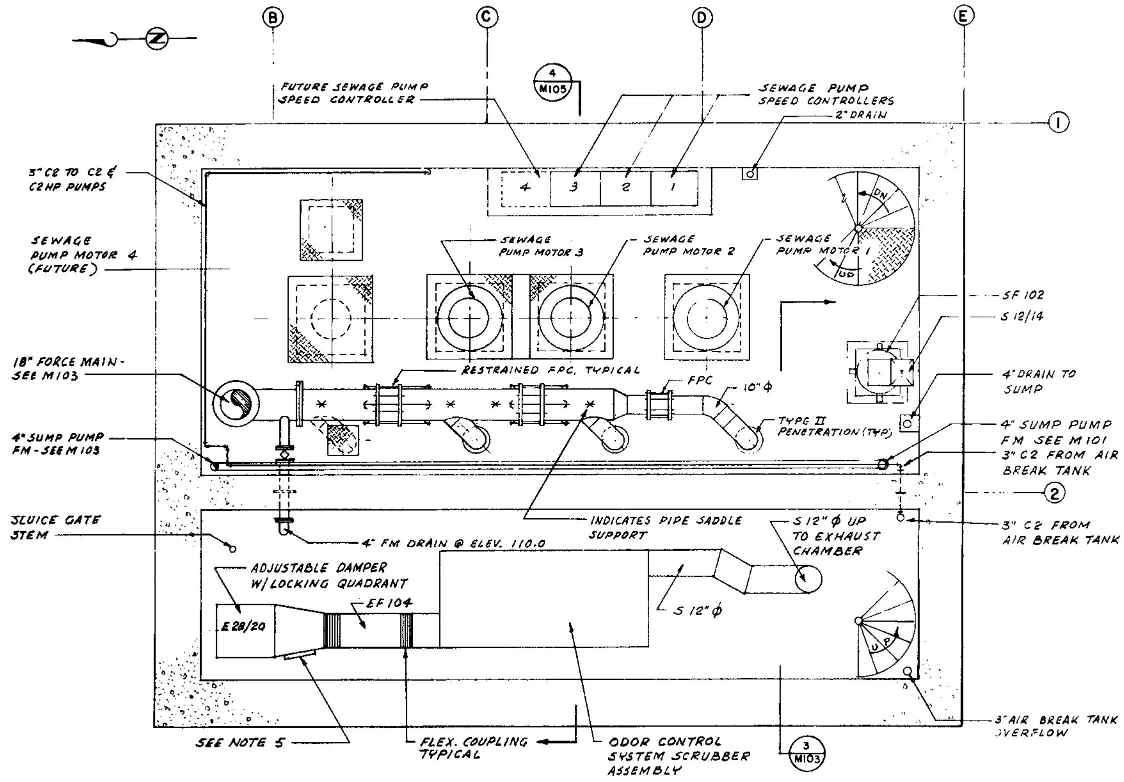


Figure 17-13. Motor room and wet well plan. North Mercer Island Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

The noise suppression measures include the following:

- The pump drive motors were specially designed for quiet operation.
- The pump drive motors are located in a below-grade room with all access openings fitted with hatches or doors.
- All walls above grade are of double-wall construction with a sound-absorbing filler between them.
- Double glazing (with nonparallel panes) was used for all windows. The panes of synthetic, bulletproof glass are resilient to absorb energy.
- The heavy transformer enclosure is located two-thirds of a wavelength from the transformer shell. The walls are high enough to focus a 60-Hz hum upward.
- A special sound-absorbing design was used for air inlet louvers.
- The design criteria included extra-heavy construction for all substructure and superstructure floors to prevent sound transmission to adjacent soils.
- All ventilation systems discharge upward with special enclosures to focus sound upward.

All above-grade lighting systems were designed to mimic residential lighting except when maintenance is required. Odor suppression measures included variable-speed pumping to maintain clean upstream sewers and a self-cleaning wet well. Originally, a dry scrubber with pellets of potassium permanganate and alumina was installed on the ventilation exhaust. However, material costs were very high, charging the reactors with pellets was labor-intensive, and—worst of all—odor removal was poor. The system was replaced with a mist scrubber that removes 95% of the H_2S. *[Ed. Note: A passerby would smell nothing, would hear nothing but the brook and the wind in the trees, and would assume the structure to be an attractive residence with an attached double garage and a wide driveway.]*

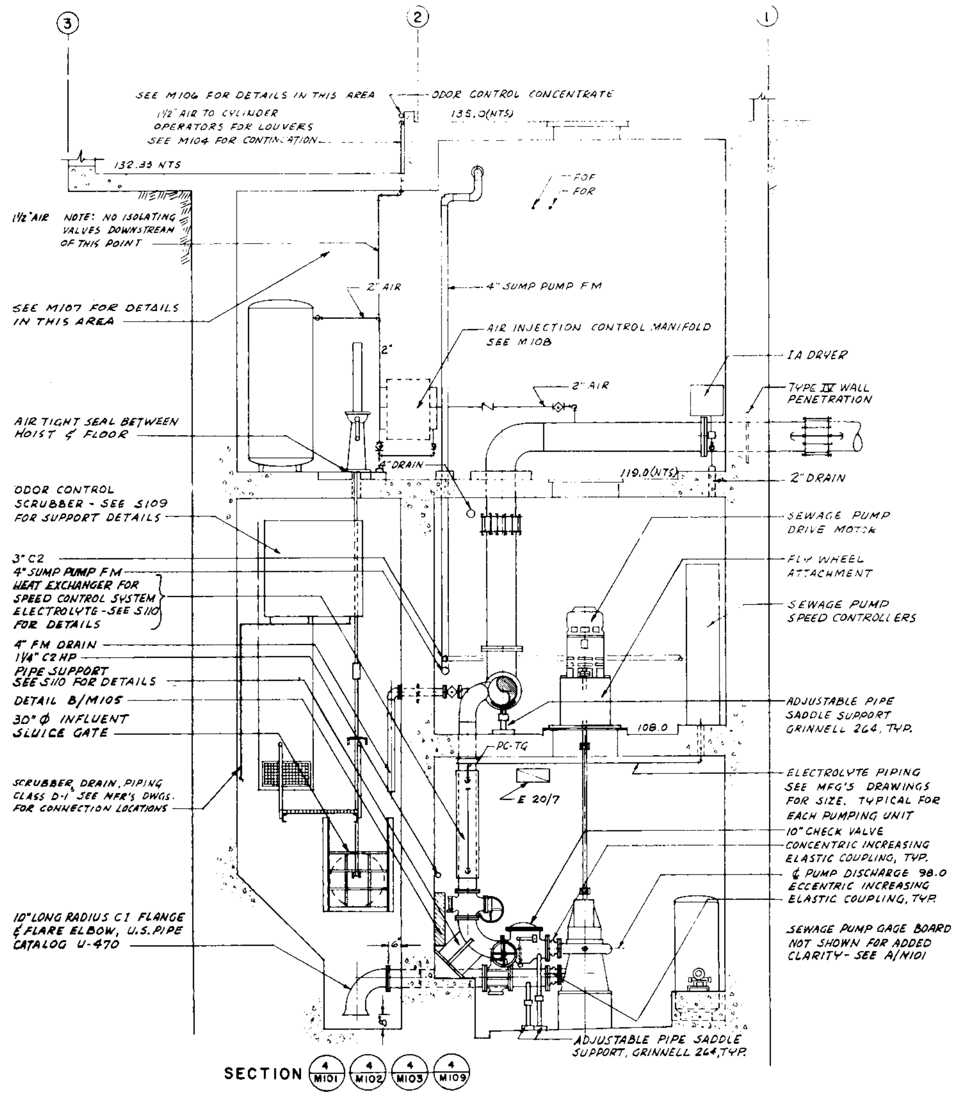


Figure 17-14. Typical section. North Mercer Island Pumping Station, King County Wastewater Treatment Division. Courtesy of Brown and Caldwell.

17-7. Examples of Small Lift Stations

Small pumping stations usually feature C/S duplex pumps in round wet wells. Some older designs have flat bottoms with either small fillets or none. Cleaning was given no consideration, and such wet wells are, typically, repositories of noxious sludge and scum. The facilities described herein were, on the other hand, designed to be either self-cleaning or cleaned with a minimum of effort. Consequently, all are successful, all meet their design objectives, and all are easy to clean and are kept clean. Passersby detect no odor whatsoever, and even when the wet well covers are opened, there is only the faint smell of fresh domestic wastewater. All of these pumping stations in Examples 17-5, 17-6, and 17-7 have been described by Sanks, Jones, and Sweeney [6].

A conical-bottom wet well is shown in Example 17-5. A flat-bottom wet well cleaned by violent hydraulic mixing is shown in Example 17-6. A conical bottom well for dry pit pumps is depicted in Example 17-7.

Example 17-5
Vallby Pumping Station

Vallby Pumping Station is located in Sweden, not far from Stockholm. It has been so successful that a great many—perhaps most—of the small pumping stations built since are copies. It was designed by an experienced operator—not an engineer—who wanted a facility that could easily be kept clean and would require the absolute minimum of attention and time for maintenance and operation.

The hopper bottom shown in Figure 17-15 is made of 18-8 stainless-steel (ss) plate 12 mm (½ in.) thick and bent so that its top fits the round, vertical concrete pipe and its bottom conforms to a rectangle of minimum size with rounded corners. The sides are inclined at least 60 degrees. Special discharge elbows were made and welded to a heavy plate for bolting to the hopper side. The space between the hopper bottom and the side of the round pipe is filled with concrete. The sump was perfectly clean at the time of inspection, and a demonstration of pump-down to the lowest achievable water level (the top of the pump volute) showed that cleaning would be very effective indeed.

The unique features include the following:

- By placing most of the pump discharge elbow outside of the ss "cone," the flat floor can be made very small, so that virtually all sludge beneath a pump is ejected when the pump is turned on.
- A curb supports: (1) the guide rods, (2) the conduit through which the pump power cable and the level indicator cord reach the control box, and (3) the water lance and keeps it within easy reach.
- Guide rails are stainless-steel telescoping tubes that are raised out of the water except when a pump is to be reinstalled.
- Instead of floats for monitoring water level and controlling the pumps, a piezo-electric pressure cell is placed within an open 100-mm (4-in.) PVC pipe with its lower end beside the volute of the pump. These cells are reliable, long-lived, and excellent for sophisticated systems because they can, unlike floats, provide input throughout the liquid level range to a PLC for activating the pumps for both normal operation and for pump-down. In the long run, they are probably less expensive than floats. (Some floats must be replaced yearly, although there are some that contain micro-switches instead of mercury switches and are supplied with more resistant electrical cables, which can last for many years.)
- A fresh water supply for washing is equipped with a quick-connect and a valve contained in a large pipe with a padlocked cover.
- Inside the wet well, there is a wash-down water lance that slides in a collar supported by a universal joint that permits freedom of direction. The water lance is equipped with a quick-connect at the top and a nozzle at the bottom. A short hose with mating quick-connects on each end is carried in the operator's truck. The system makes wash-down not only quick and easy, but the jet strikes the pump or hopper bottom so far away that splashing does not reach the operator.
- Under the hatch there are two hinged grates consisting of heavy rods spaced at about 150 mm (6 in.) each way to cover the opening and prevent large objects from falling in. They give workers a feeling of safety. Only one must be lifted to remove a pump. The curb is a support for the 150-mm (6-in.) pipe that carries the power cable from the control box to the sump. It takes only seconds to unplug the power cable and pull it out when removing a pump.
- Operating water levels can be easily changed at the control panel. A pushbutton is provided for making a motor run backward to clear a clogged pump.

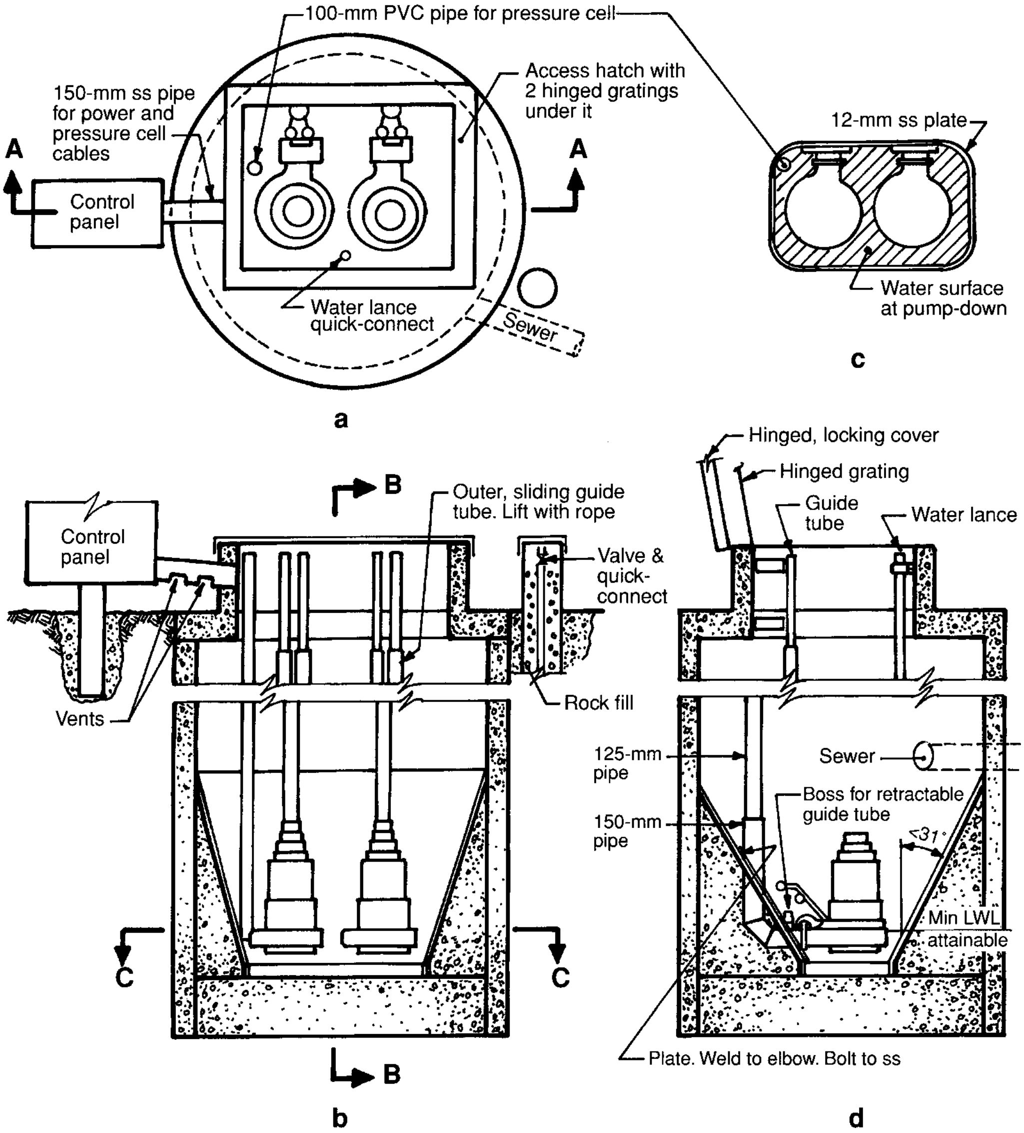


Figure 17-15. Schematic diagram of Vallby Pumping Station. (a) Plan; (b) Section A-A; (c) Section C-C; (d) Section B-B.

Critique of Example 17-5

If the waterfall into the wet well were avoided by means of an approach pipe laid at a 2% grade and discharging at low water level, this wet well would conform to all the concepts for self-cleaning wet wells described in Chapter 12. At small flows, the jet from the inlet strikes the sloping stainless-steel plate, so the generation of bubbles is reduced.

The thickness of the stainless-steel hopper-bottom insert (12 mm or 0.5 in.) seems excessive and unnecessarily costly. The plate needs to be only stiff enough to allow concrete to be placed between it and the concrete pipe wall. Perhaps a thinner plate with interior stiffeners or a molded plastic shell would be less expensive and just as satisfactory.

The floor under the pumps is so small that significant amounts of sludge cannot accumulate. But where scum is a problem, the controls could be programmed for automatic pump-down at some suitable interval, say, twice per week. On the other hand, some engineers feel that any time a wet well is

pumped down and the pump loses prime, an operator should be at hand for troubleshooting.

It is important to keep the area between the pump volutes and the side of the hopper bottom as small as possible. Otherwise, the vortex that forms at pump-down will not suck all the scum into the intake. Of course, it is not necessary to remove all the scum (more will accumulate anyway), but if cleaning is required to leave a spotless water surface, the smallest permissible water surface area at lowest water level is prerequisite.

Other Swedish Pumping Stations

A growing number of Swedish pumping stations are similar to Vallby Station not only in design but also operationally. One, intended to serve a future subdivision, was connected to only a few houses, so the detention time was very long, and the wet well produced overwhelming odors. The steep-walled sump is advantageous, because detention time and odors can be reduced by setting the HWL close to the LWL to increase the frequency of pump starts and keep the wastewater at least a little fresher. Adding a small flow of fresh water would also help, as would feeding iron chloride to sequester the sulfide ion. A small, obviously inexpensive hydro-pneumatic tank was installed in one station for controlling water hammer. Many such tanks have been in use for many years and are said to be quite satisfactory and devoid of excessive maintenance problems.

Example 17-6
Clyde Pumping Station

The pumping station at Clyde in Contra Costa County, California, is shown in Figure 17-16. The two 7.5-kW (10-hp) C/S pumps are each capable of discharging 26 L/s (410 gal/min) of wastewater at a TDH of 11.1 m (36.5 ft) at a minimum efficiency of 62%.

One of the main functions of the unique 76-mm (3-in.) piping system is to backflush the pumps to remove rags. Another function is to drain the short force main, and a third function is to promote the vigorous mixing of scum, sludge, and grit so the "homogenized" mix can be discharged to the force main and thus clean the sump. Smaller pipes might plug. The four eccentric plug valves allow complete flexibility of operation. Either pump can be used to unclog the other, and either can supply the mixing water while the other one pumps the mixture into the force main. Alternatively, the force main itself can be tapped for about 15% of its flow for recirculation as mixing water, while either (or both) pump(s) discharge the mixture.

The 76-mm (3-in.) pipe to the sump discharges just above LWL. At the time of inspection in 1993 (a few months after the station was put into service), the sump was remarkably clean with no material floating on the surface. There was no odor.

Although a little manual labor is required to manipulate the valves, the station can be cleaned with less effort than, say, Kirkland Pumping Station. So it is fair to describe it as a "self-cleaning" facility.

Critique of Example 17-6

In comparison with the Vallby pumping station, the gentle bottom slopes allow more active storage volume, and the simpler bottom with its plane surfaces is less expensive to construct. On the other hand, four extra plug valves are required in a valve vault that must be enlarged to accommodate them. The 76-mm (3-in.) piping system allows maximum versatility in mixing. If a less expensive system is wanted, the system could be reduced to one valve at the 102-mm (4-in.) force main. Flexibility would suffer, but there would be little loss of effectiveness. Discharging the 76-mm (3-in.) piping at or above the LWL drives both scum and bubbles into the pool, whereas discharge at a lower elevation would be just as effective for sludge and perhaps a little less effective for scum, but it would avoid the bubbles.

One advantage of the Clyde piping system is that one pump can be used to backflush the other to remove a blockage.

Unlike the Vallby pumping station, which ejects sludge whenever a pump is turned on and which can be programmed to eject scum automatically, an operator must visit the Clyde pumping station to clean it—no real disadvantage if the policy is always to have an operator present at pump-down.

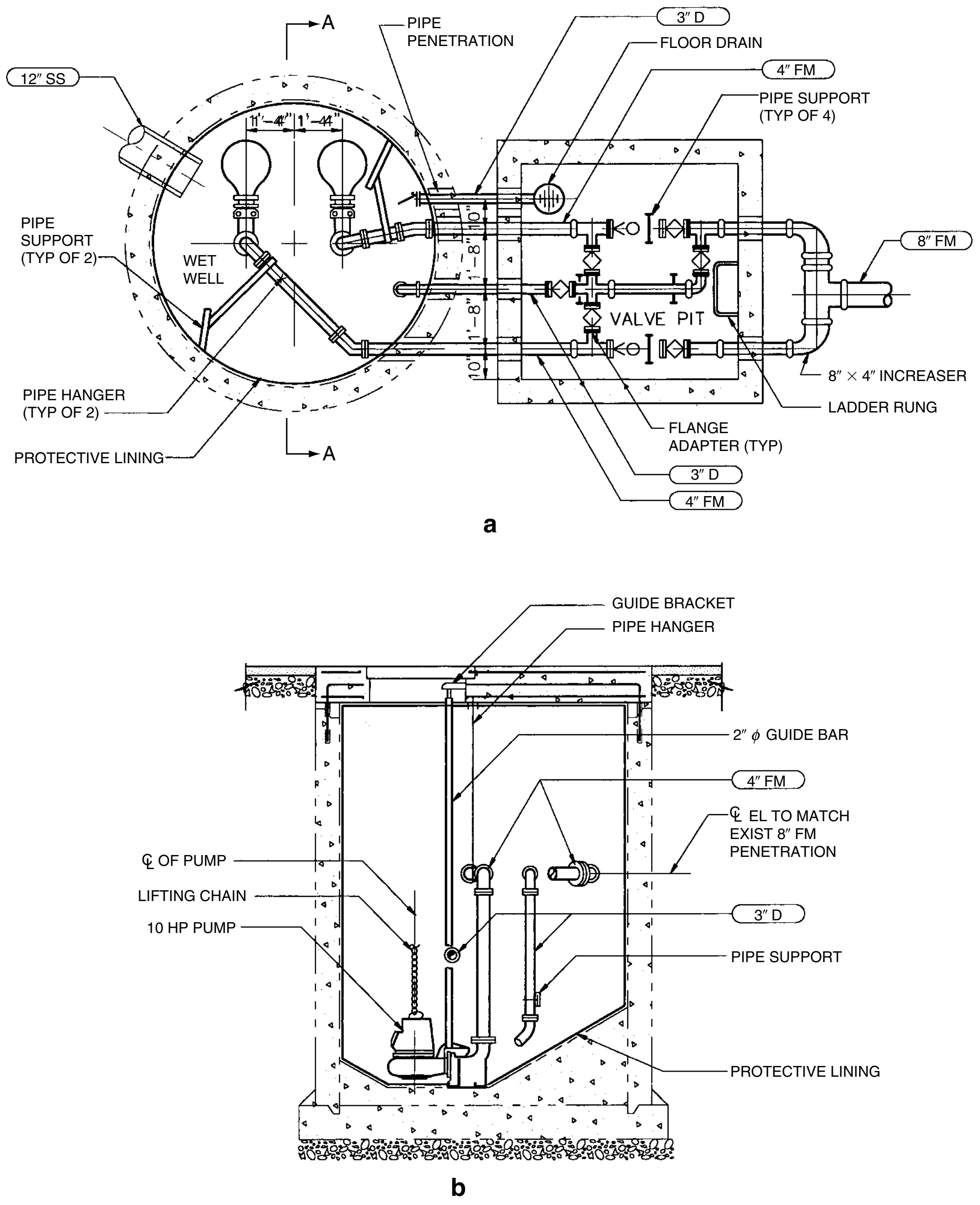


Figure 17-16. Clyde Pumping Station. (a) Plan; (b) Section A. Courtesy of G. S. Dodson & Associates, Consulting Engineers.

At Clyde, the operator must climb down into the pump vault to operate the valves. An improvement would be to install a removable grating about 0.6 m (2 ft) below grade and extend the valve stems through the grating to just below the hatch. The operator could then manipulate the valves without entering the vault. This improvement was incorporated into the California Maritime Academy Pumping Station at Vallejo, California, in 1996.

Example 17-7
Black Diamond Pumping Station

Plans for the wet well for the lift station at Black Diamond, Washington, are shown in Figure 17-17. The 400-mm (16-in.) approach (influent) pipe slopes at a 2% gradient for 61 m (200 ft) but is horizontal for 10 pipe diameters before entering the wet well. There are two self-priming pumps of 63 L/s (230 m^3/h or 1000 gal/min) each housed in a nearby building and each supplied by a 250-mm (10-in.) suction pipe in the wet well. Each suction pipe ends in a bell of 400 mm (16 in.) diameter, so the entrance velocity is only 0.49 m/s (1.6 ft/s).

The layout of the pump room (not shown) allows for the utmost flexibility of operation—a requirement of the owner. A diesel gen-set furnishes enough power to operate the station in the event of a power outage.

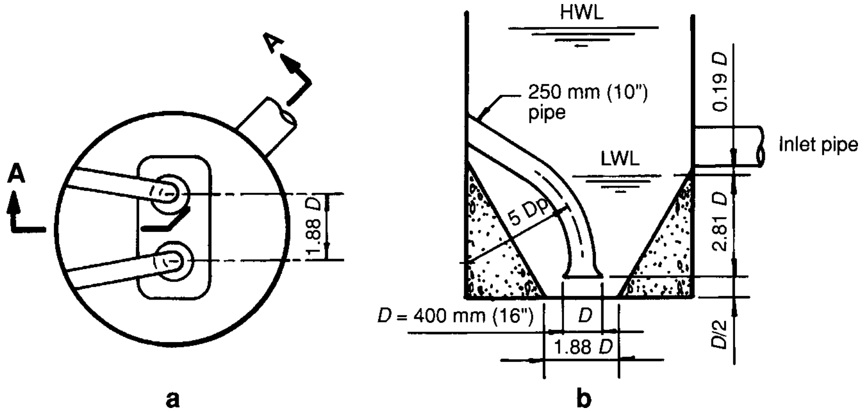


Figure 17-17. Black Diamond Pumping Station sump. (a) Plan; (b) Section A-A.

Critique of Example 17-7

The behavior of the steeply sloping (2% gradient) approach pipeline is entirely satisfactory, but the LWL shown on the plans is too low to prevent the hydraulic jump in the approach pipe from reaching the sump. Of course, the LWL can be readily changed.

In the sump, the smooth sides sloping at 60 degrees keep solids from sticking during pump-down. The pumps broke suction when the submergence of the bells was about 1 D. At this water depth, the area of the water surface was too large to confine the scum sufficiently so that all scum could be ejected on a single pump-down, and a second pump-down was needed. During an inspection, it appeared that operating the two pumps simultaneously would possibly remove the scum in a single pump-down, but that operation was not tested. The designer has stated that, in a redesign, the walls would be made vertical from the floor to 1 D above the pump intakes so as to restrict the area to which scum would be confined upon pump-down.

The bell is too large and the pump entrance velocity is very low and might not suck out sand and gravel quickly. Nevertheless, cleaning is accomplished with reasonable dispatch. However, the designer stated he would reduce the size of the flat floor next time.

The success of the Black Diamond Pumping Station demonstrates the practicality and usefulness of the concepts explained in Chapter 12.

17-8. References

1. ASCE Manual 37, *Design and Construction of Sanitary and Storm Sewers,* American Society of Civil Engineers, New York (1970).
2. Metcalf & Eddy, Inc., *Wastewater Engineering: Collection and Pumping of Wastewater,* McGraw-Hill, New York (1981).
3. AISI, *Modern Sewer Design,* American Iron and Steel Institute, Washington, DC (1980).
4. Parmakian, J., *Waterhammer Analysis,* Dover Publications, New York (1981).

5. Dicmas, J. L., *Vertical Turbine, Mixed Flow, & Propeller Pumps,* McGraw-Hill, New York (1987), Figures 6.6, 6.8, 6.9, and 6.10.
6. Sanks, R. L., G. M. Jones, and C. E. Sweeney, "Improvements in pump intake basin design," EPA 600/ R-95/041, RREL-CR. Order No. PB 95-188090. National Technical Information Service, 5285 Port Royal Road, Springfield, VA 22161 (1995).

Chapter 18

System Design for Water Pumping

BAYARD E. BOSSERMAN II
RICHARD J. RINGWOOD
MARVIN DAN SCHMIDT
MICHAEL G. THALHAMER

CONTRIBUTORS
Pat H. Bouthillier
Roland S. Burlingame
A. L. Charbonneau
Allen W. Peterson
David M. Reeser
Joseph W. Steiner
Jerald D. Underwood
Eric L. Winchester
James R. Wright
Eugene K. Yaremko

The purposes of this chapter are: (1) to describe methods for determining the required capacity and pressure for water pumping stations; (2) to present recommendations on selecting and sizing the pumps; (3) to discuss control methods for operating pumps; and (4) to provide representative examples of pumping station design. Examples of pumping station design may be worked only in the units originally used, whereas some may be worked in both SI and U.S. customary units.

18-1. Types of Water Pumping Stations

For the purposes of discussion, water pumping stations are considered to fall into five general categories or systems:

- Source (such as a well) pump discharging into an elevated tank
- Raw water pumping from a river or lake
- In-line booster pumping into an elevated tank
- High service pumping of finished water at high pressure
- Distribution system booster without a storage tank in the piping system.

Functionally, there may be little difference between high service and distribution or in-line booster pumping or between low service (raw water pumping to a treatment plant) and booster pumping.

18-2. Pumping Station Flow and Pressure Requirements

Water pumping stations are fundamentally different from wastewater pumping stations because they do not have to be sized to pump at high peak flow rates.

A wastewater pumping station must be able to pump whatever wastewater flow enters it, but a water pumping station can be designed to take advantage of water storage reservoirs in the system.

Flow Requirements

Water pumping stations are usually designed to supply water to an area in which the required demand is reasonably well defined or can be projected to a reasonable degree. In a water distribution system, the demand is a combination of customer needs and fire flow requirements. Average annual per capita water consumption, peak hour, and maximum daily demands vary widely depending on factors such as climate, income levels, population, and the proportions of residential, commercial, and industrial users. Typical water consumption values are 560 to 760 L/cap · d or 150 to 200 gal/cap · d. More detailed data on per capita water demand are readily available [1–5]. Most water supply utilities and cities also have extensive records on water consumption.

Fire flow requirements are usually dictated by the fire code adopted by the local jurisdiction. Formulas based on population or on the individual buildings [where area, height, type of construction, occupancy, installation of automatic sprinklers, and proximity (and thus exposure) to other structures] may be useful. For example, the fire flow requirement can be estimated using an early National Board of Fire Underwriters formula. In SI units, the formula is

$$Q = 232P^{0.5}(1 - 0.1P^{0.5}) \tag{18-1a}$$

in which Q is the fire demand flow in cubic meters per hour and P is the population in thousands. In U.S. customary units, the formula is

$$Q = 1020P^{0.5}(1 - 0.1P^{0.5}) \tag{18-1b}$$

where Q is fire demand flow in gallons per minute and P, again, is the population in thousands. Be wary of computing fire flows based on formulas that include only the population factor because they can give unrealistically low flow values in comparison to a fire flow demand that might actually occur. For example, in a town of 1000 people, P would be 1 and Q would be $230\,\text{m}^3/\text{h}$ (1000 gal/min), which may be inadequate. The presence of a lumber yard, for example, would require a high water flow allocated for fire. Furthermore, design flows for pumping stations and water mains depend on the reservoir storage capacity available. Generally, the design flow should be the larger of (1) peak hourly demand, or (2) maximum daily demand plus fire flow.

For commercial areas, the flow demand is usually estimated as flow rate per area. Demands of 18.7 to $47\,\text{m}^3/\text{ha} \cdot \text{d}$ (2000 to 5000 gal/acre · d) are not unusual in mixed commercial and residential areas [2–5]. Zoning classifications greatly affect these unit flows. In one large water agency on the East Coast of the United States, water consumption varies from 18.7 to $700\,\text{m}^3/\text{ha} \cdot \text{d}$ (2000 to 75,000 gal/acre · d) depending on the zoning classification.

Pressure Requirements

Service connection pressures during normal system operations should lie between the following limits:

- Minimum pressure = 210–280 kPa (30–40 lb/in.^2)
- Maximum pressure = 410–550 kPa (60–80 lb/in.^2)

Some cities and agencies further define acceptable minimum pressures as, for example,

- 276 kPa (40 lb/in.^2) for maximum daily flow
- 207 kPa (30 lb/in.^2) for peak hourly flow
- 138 kPa (20 lb/in.^2) for maximum daily flow plus fire flow.

A frequent practice in water districts is to have zones with a 45- to 60-m (150- to 200-ft) difference in elevation from top to bottom. This corresponds roughly to the maximum 410 to 550 kPa (60 to 80 lb/in.^2) pressure range above. A major reason for the maximum pressure of 550 kPa (80 lb/in.^2) is that household plumbing fixtures—especially water heaters—cannot withstand greater pressures.

In sparsely populated areas, it is possible to establish zones with a 90-m (300-ft) difference in elevation and to serve the lower areas through pressure-reducing valves. It is also possible to use booster pumping and storage in alternating zones (e.g., pump from zone 1 to

Example 18-1
Flow Requirements in a Small Town

Problem: A town has a population of 10,000. Find the requirements of the water supply system if (1) an adequate reservoir is to be built, (2) an existing but somewhat inadequate reservoir exists, and (3) there is no reservoir.

Solution: Note that data and calculations are rounded off to the customary two (or sometimes three) significant figures. The assumptions are as follows:

Item	SI Units	U.S. Customary Units
Population	10,000	10,000
Water consumption	758 L/cap · d	200 gal/cap · d
Fire flow rate (from Equation 18-1)	197 L/s	3100 gal/min
Fire flow time	8 h	8 h

So the demand flow rates are as follows:

Average daily	10,000 × 758/(1440 × 60) = 88 L/s	10,000 × 200/(1440 min/d) = 1400 gal/min
Maximum daily	2.5 × avg daily = 220 L/s	2.5 × avg daily = 3500 gal/min
Peak hour	4 × avg daily = 352 L/s	4 × avg daily = 5600 gal/min
Average daily + fire flow	280 L/s	4500 gal/min
Maximum daily + fire flow	420 L/s	6600 gal/min

(1) Storage reservoir. If a properly sized water storage reservoir is provided in the system, it would be reasonable to design the pumping station with a firm capacity equal to the maximum daily demand—in this example, 220 L/s (3500 gal/min). The reservoir would supply additional water to the system during peak hour and fire flow demands.

Although the detailed sizing and design of water storage reservoirs are beyond the scope of this book, some general criteria are as follows:

- Peaking storage equals 25 to 50% of the average daily demand
- Fire flow storage equals fire flow for 3 to 8 h duration
- Emergency storage required for loss of power supply with no standby power supply or alternative water supply available equals two to three days of maximum daily demand.

The total required storage, then, can range from the volume required by one of these criteria to the volume required by the sum of all of the criteria. The actual size of the storage reservoir depends on the pumping station capacity and design considerations, such as whether standby power is provided for the pumping station. In this example, if standby power is omitted, the required storage is:

Storage	SI Units	U.S. Customary Units
For peaking	$0.25(88\,\text{L/s})(86{,}400\,\text{s/d}) \times 10^{-3}\,\text{m}^3/\text{L} = 1900\,\text{m}^3$	$0.25(1400\,\text{gal/min})(1440\,\text{min/day}) =$ 504,000 gal
For fire flow	$(197\,\text{L/s})\ (3600\,\text{s/h})\ (8\,\text{h}) \times 10^{-3}\,\text{m}^3/\text{L} = 5700\,\text{m}^3$	$(3{,}100\,\text{gal/min})\ (609\,\text{min/h})(8\,\text{h}) =$ 1,500,000 gal
For emergency (2 days of storage at maximum daily demand)	$(2\,\text{d})(220\,\text{L/s})(86{,}400\,\text{s/d})\times 10^{-3}\,\text{m}^3/\text{L} = 38{,}000\,\text{m}^3$	$(2\,\text{d})(3500\,\text{gal/min})(1440\,\text{min/d}) =$ 10,100,000 gal
Total	$45{,}600\,\text{m}^3$	12,100,000 gal

Notice that by far the largest storage volume is due to the emergency storage requirement, which results from the assumption that the pumping station lacks standby power or that the water supply is temporarily lost.

Selection of pumps. When considering this reservoir, a reasonable design would be to provide four $320\,\text{m}^3/\text{h}$ or 89 L/s (1400 gal/min) pumps identical for ease in stocking the same spare parts.

- One pump would normally operate for the average daily flow.
- The second and third pumps would come on as demands increase and the pressure consequently decreases up to the maximum daily flow. The nominal peak flow would be $3 \times 320 = 960\,m^3/h$ or 267 L/s ($3 \times 1400 = 4200$ gal/min).
- The fourth pump would serve as standby when one of the pumps must be taken out of service for repairs.

In this example, the installed maximum operating capacity of $960\,m^3/h$ (4200 gal/min) is slightly greater than the required maximum daily flow of 220 L/s (3500 gal/min) because of the decision to use pumps of the same size. The actual total capacity would be less than the installed maximum because of the increased headloss at higher flows.

(2) Inadequate reservoir. Reservoir capacities are often marginal or even substandard. In a congested urban area, it may not be feasible to construct additional reservoir capacity. Even in relatively open areas, topography may be so flat that additional reservoirs are unwarranted. In such situations, constructing additional pumping facilities may be the only feasible alternative.

Consider the previous example, but assume it has been determined that sufficient reservoir capacity is available to satisfy peak hour and short-term maximum daily demands, but not average daily plus fire flows. In such a situation, the pumping station must satisfy both (1) maximum daily flow, and (2) average daily flow plus fire flow. A possible design might provide:

Demand	SI Units	U.S. Customary Units
Average daily	One pump at 88 L/s	One pump at 1400 gal/min
Fire flow	One pump at 197 L/s	One pump at 3100 gal/min
Maximum daily	Both of the pumps operate	Both of the pumps operate
Standby	Another pump at 197 L/s	Another pump at 3100 gal/min

(3) No reservoir in the system. The difference between the first part of this example, 18-1(1), and this part, 18-1(3), is that in this part, the pumping station must supply all of the water demands— there is no reservoir to make up peak hour and fire flow demands. Thus, the pumping station should be designed to supply the larger of (1) a peak hour demand of 352 L/s (5600 gal/min), or (2) a maximum daily demand plus fire flow of $220 + 197 = 417$ L/s ($3500 + 3100 = 6600$ gal/min).

A reasonable design might consist of the following number and sizes of pumps.

One pump at 101 L/s	One pump at 1600 gal/min
Two pumps at 159 L/s	Two pumps at 2500 gal/min
One standby pump at 159 L/s	One standby pump at 2500 gal/min

In all of the examples, consideration must also be given to system operation at very low flows, such as might occur at night. In the first and second parts of this example, the pumps could be shut off and the reservoir could supply the small amount of water required. In this part, there is no reservoir, so a pump must continue to operate. It is usually not feasible to allow a pump to operate at flows less than, say, 33% of its optimum capacity, so there are three alternatives.

- Install a small pump that is sized to deliver the low flow needed.
- Provide a variable-speed drive on the smaller pump to reduce its output. The variable-speed drive would be controlled to maintain a set minimum system pressure.
- Add a hydropneumatic tank (see Example 18-5).

Controls for the pumping station in all three parts of this example can be based either on the pressure sensed in the pumping station discharge or on the change in the level of a reservoir. Pressure at a pumping station could, depending on system geometry, be difficult for control. A control scheme might be as follows:

- Demand on the system causes a decline in water level in the reservoir or a decrease in the pressure in the system.
- As soon as the pressure at the pumping station falls to Pl, the first pump comes on.

- If the pressure continues to fall to P2, the next pump comes on.
- This procedure continues until (if necessary) all pumps are running.
- Eventually, the water level in the storage tank rises and fills the tank and the system pressure starts to rise.
- As the pressure rises, the pumps are stopped. The order in which the pumps are shut off can be as simple as shutting them off in the order in which they started. A more complicated scheme is to run combinations of pumps of various capacities to provide small changes in flow. In this part of the example, the following combinations of pumps can be run to meet differing demands.

Average daily:	One pump at 100 L/s or one pump at 159 L/s	One pump at 1600 gal/min or one pump at 2500 gal/min
Maximum daily:	One pump at 100 L/s or one pump at 159 L/s Total of 259 L/s or two 159 L/s pumps for a total of 318 L/s	One pump at 1600 gal/min or one pump at 2500 gal/min Total of 4100 gal/min or two 2500 gal/min pumps for a total of 5000 gal/min
Peak hour or maximum daily plus fire flow:	Two 159 L/s pumps = 318 L/s plus one 100 L/s pump = 100 L/s Total of 418 L/s	Two 2500 gal/min pumps = 5000 gal/min plus one 1600 gal/min pump = 1600 gal/min Total of 6600 gal/min

Note that the lack of a reservoir results in operating inefficiencies. Large pumps must be provided for peak hour and fire flows and must consequently operate at lower flows and less efficiency at average and intermediate demands.

Utilizing either (1) another smaller pump (one-fourth to one-half the size of the smallest pump above), or (2) a variable-speed drive on the smaller unit can result in an even finer degree of control in response to system demand and pressure.

All these flows are nominal. Increases in friction head reduce the higher flows. Accurate flows can be calculated only by due consideration of the system H-Q curve.

---

3 to 5) and to service intermediate zones (e.g., zones 2 and 4) through pressure-reducing valves until additional pumping and storage facilities can be justified.

18-3. Raw Water Pumping from Rivers and Lakes

Raw water may be pumped from a river, a natural lake, or an artificial reservoir. A potable (treated) domestic water supply may be pumped directly or indirectly through a distribution reservoir or into a water distribution system. A raw water supply may be pumped to a water treatment plant from the source either before or after passing through desilting basins.

Depending on the source and ultimate use of the water, raw water pumping facilities are generally a combination of only three basic components: (1) the raw water intake structure, (2) the pumping facilities, and (3) the screening facilities, which may or may not be required. Design variations and arrangement of these basic components depend on the imagination, ingenuity, experience, and judgment of the design engineer.

Most raw water pumping facilities are shore installations. The intake may be placed below the lowest water level on record or the lowest level of reservoir drawdown determined by storage requirements. The intake may be placed (1) at a point in deep water such as a lake or impoundment, (2) near the shoreline in shallow water, or (3) directly on shore with an excavated channel to deeper water. Compared with wastewater pumping, there is greater flexibility in placing the pumping facilities for a water supply. The pumping station may be built in the intake structure at any of the points cited above, or it may be located remotely from the intake. The connection between the intake and the pumping station may be any of the following:

- A suction pipe in a shallow trench to a higher elevation and directly connected to the pumps.

The pump would not be more than 6 m (20 ft) (including pipeline losses) above the lowest water level at the intake. The pumps must be equipped with adequate priming devices.
- An excavated open channel leading to the pump suction well.
- A horizontal pipeline from the intake directly to the pumps.
- A tunnel (in larger installations) connecting the intake to a remote pumping station.
- Combinations of any of these.

It may be desirable to incorporate the intake works and/or the pumping and screening facilities into a water supply dam. In an earth dam, the intake may be:

- A simple gate structure built into and parallel to the upstream slope, near the toe of the dam, with a conduit leading to a pumping station at or beyond the downstream toe.
- A tower inlet, located near the upstream toe, with an access bridge from the top of the dam. The tower can include pumps and have trash racks, screens, or multiple inlet ports. With this type of inlet, a conduit may also be carried through the dam to a pumping station downstream.
- A concrete dam intake section may be in the dam's upstream face, with or without racks, screens, pumps, and multiple ports.

Because many options and alternatives are available, the choice is based on the siting considerations of the land and water environment. The most important factor influencing design is the depth of the raw water source. Depending on the design flows (pumping rates) and pumping heads, it may be difficult to deal with a large variation in water surface elevation due to reservoir drawdown or natural hydrological variations. Water quality variations at different levels in a deep lake or reservoir may require multiple port inlets to the intake structure to allow taking the best quality of water available at all times.

Deep reservoirs in temperate climates turn over (sometimes rapidly) in the fall or spring or both. Turnover is caused by cold and dense surface water overlaying warmer (less dense) layers. The unstable condition causes bottom layers to be suddenly brought to the surface, often with bottom mud. Normally, the water quality at or near the bottom of a deep reservoir is inferior to that of shallower depths. It may be devoid of or low in oxygen and high in manganese and/or iron due to decayed vegetation. Because the water quality at any given depth undergoes seasonal variations, multiple ports in the inlet works of deep reservoirs are desirable to draw the best water. Top and bottom waters can be continuously mixed to a uniform quality by extending a perforated air hose or pipe across the bottom of a reservoir and supplying air from a compressor on shore, an arrangement that may allow a single inlet in the intake structure. A sampling program should be carried out throughout the year to determine water temperatures and quality at various depths.

Surface conditions in lakes and impoundments influence the intake design. In warm, humid climates, aquatic plant growth, such as water hyacinth and duckweed, is a problem. A floating boom arrangement may prevent plant growths and surface trash from entering the intakes and pumps. In cold climates the intake works and pumping arrangements must also be designed to prevent damage from ice and freezing.

River Intake Design Considerations

This subsection is an abridgement of extensive material in *Proceedings* [6].

Low Water

The determination of design low water is important for an intake that must operate year-round. Design low water is an important factor in the design of a direct intake, in the layout of an infiltration gallery, and in the pump setting and wet well floor elevation. Design low flow can be obtained through a routine low-flow frequency analysis of water survey data. If there are no data for the stream, either a regional analysis or the analysis of transposed data may be required.

Cross-section data can then be used to calculate a design low stage. The calculations must begin at a downstream control section or at a control reach in the river. Consider the likelihood of channel changes that could significantly change low-flow levels over the expected life of the intake structure. The degradation of a rapids or a downstream control section or the migration of rapids from downstream to upstream of the site would result in lower water levels, as would the development of a downstream cutoff. The development of a large bar downstream, or a cutoff just upstream, could cause higher water levels. The likelihood of such an occurrence and the magnitude of the effect cannot be determined analytically. Any allowance made must be a matter of judgment based on a study of comparative aerial photographs and observations made in the field.

High Water

The elevation of design flood high water is also an important factor in determining the elevation of the working floor of the pumphouse and in determining the design of any river training works that might be associated with the intake. In many instances, design scour depths can also be determined from mean flood depths.

The discharge of the design flood can be estimated by analyzing the water survey data. The corresponding stage can be computed by using the cross-section data and the channel and flood plain roughness estimates. Direct stage measurements at one or two high flows of known magnitude are most useful for increasing the accuracy of the calculated design high water. High-water marks, information from local residents, and backwater computer programs are also very useful in simulating water levels once the channel characteristics have been determined.

The choice of design high- and low-water levels should be based on the aggradations of the river reach under consideration because estimated water levels and streambed levels can be dramatically reduced at an intake if the river reach is subjected to degradation. Alternatively, on an aggrading reach such as the backwater above a storage dam, water and streambed levels will rise, which results in more frequent flooding during periods of high flow and possible inundation of the intake with sediment.

Once the frequencies of the low- and high-water levels have been ascertained, the probability of levels above or below these values during the life of the structure can be determined. For a 50-year design life of the structure and design high- or low-water levels with a return period of 100 years, there is a 60% assurance that the high-water levels will be exceeded at least once in the life of the structure (see Figure 18-1). For the low-water-level condition (for a similar design life and return period), the assurance of lower water levels would also be 60%.

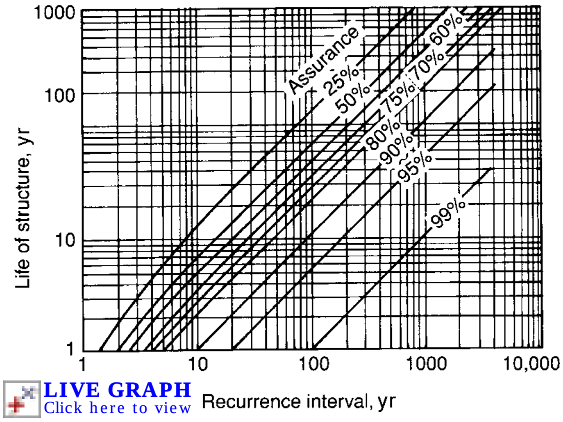


Figure 18-1. Probability of flood damage during the life of a structure.

Trash and Debris

Large amounts of trash and debris, floating or suspended, are likely to be carried by the stream at high discharges. River structures that project high above the bed can trap the debris and can sometimes cause the formation of an adjacent floating island that impedes the operation of the intake. Log booms can be constructed to deflect the floating debris away from the intake, but booms are effective only if the angle between the direction of the surface velocity and the boom is small. Suspended debris, such as waterlogged wood and coal, can be drawn into the system and can clog screens or trash racks. These materials can be removed by adequate backflushing, and frequent backflushing is usually required during periods of high discharge. Trash racks should be designed with horizontal bars exposed to the flow. The racks should be recessed into the face of the intake structure to minimize the hang-up of trash on the upstream side of the rack.

Ice Jams

Design high water in regions of below-zero weather may often result from ice jam flooding. For example, the water levels during spring break-up on the Athabasca River at Fort McMurray, Alberta, can be 6 m (20 ft) above the summer high flood levels because of ice jams. Evidence for such high-water levels may be discovered during the field survey, from either ice scars on tree trunks or information from local residents. Ice-jam data can be found in the records of various government departments and public and private agencies, in the archives of local newspapers, in railway engineering and maintenance records, or even in old diaries of church missions.

Frazil Ice

Few water intakes in northern regions with freezing temperatures are immune from frazil ice problems. A length of river in which rapids or highly turbulent areas are combined with reaches of open water and subzero temperatures can generate large quantities of frazil ice that can result in very large ice thickness

downstream with a high potential for jamming and possible ice damage to structures in this reach.

To combat active frazil ice, amorphous plastics, such as polyethylene and other special coatings on steel bars, or slightly heated surfaces can be used. Frazil ice does not form on wooden racks nearly as readily as on steel ones. A short frazil run and large rack openings will result in a small amount of ice deposit with no significant headloss.

Mechanical rack cleaning is successful if the frazil run is not heavy. A motor drives a device with teeth that mesh with the openings up and down the screen. But if the frazil run is heavy, the mechanical scrapers may seize.

Recent research has shown that an economical way to get rid of frazil ice problems is to use screens with large openings and to coat the steel bars with plastic resins, polyethylene coatings, or silicone grease.

Screens for Fish Protection

Some regulatory authorities have insisted that fish be protected by intake screens with small slots (2 mm or 0.08 in.) and approach velocities of no more than 0.15 m/s (0.5 ft/s). Such small slots are completely impractical in the presence of frazil ice. In severe climates, the only protective measure that works well appears to be electric shock.

During the preliminary stages of design, be sure to obtain approvals of conceptual plans from the U.S. Army Corps of Engineers, the regional U.S. EPA, the state department that deals with environmental protection, and, perhaps, the state fish and game authority.

Stable Channels and Mild Climates

If the river channel is stable and the water levels are controlled within reasonable limits either by an upstream dam or by a weir in the channel, if ice is not a serious problem, and if the intake capacity required is relatively small (say, no more than about 1 m^3/s or 20 Mgal/d), a relatively simple intake structure may suffice. Some of the types are:

- Tower intake with multiple, screened ports on the downstream face
- Infiltration galleries along the bank
- Transverse perforated pipe under the streambed (protected with a rock overlay)
- Weir and side channel
- Forebay or lagoon constructed beside the river with a shore intake, as per Figure 18-2.

Lake Intake Design

Tower Intakes

Tower intakes are built in deep water, have a bridge to the shore or to the crest of a dam, and may or may not have screening or pumping facilities. These intakes are used when taking water from varying levels

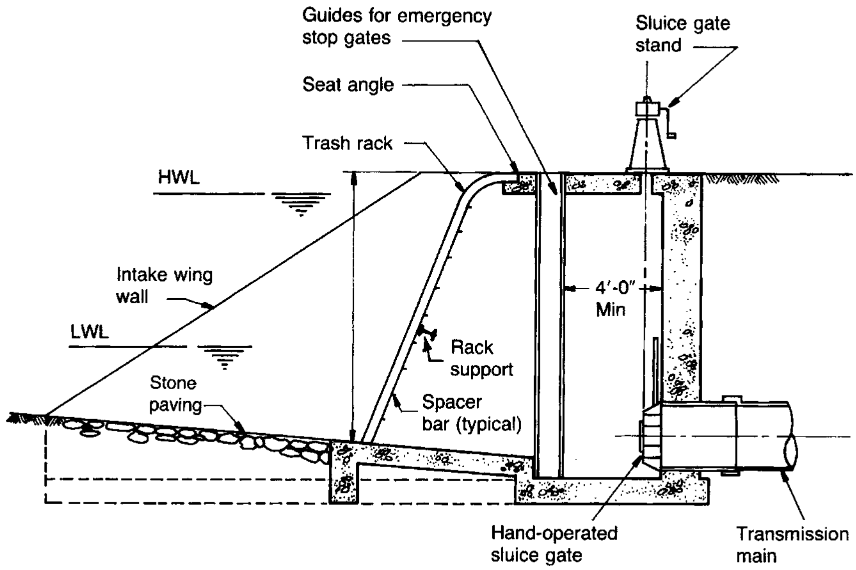


Figure 18-2. A typical shallow shore intake with a hand-cleaned trash rack. After Camp Dresser & McKee.

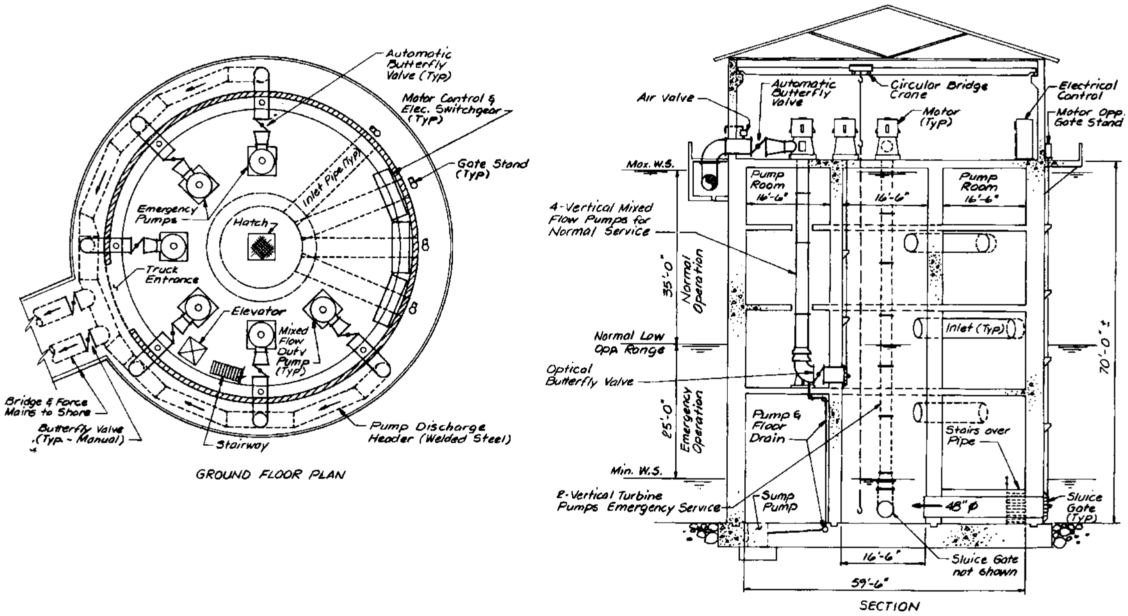


Figure 18-3. Tower intake. Courtesy of Camp Dresser & McKee.

in a deep, newly constructed reservoir or in dewaterable reservoirs, so dry or shallow water conditions prevail throughout the construction period. Because most tower intakes are in deeper water, they are usually circular in plan. The configuration (circular, square, or hexagonal) is based on (1) structural considerations, (2) water depth, (3) the need for trash racks or screens, (4) pump space, if required, and (5) whether the tower may be dewatered under full or partly full reservoir conditions.

The cost of the access bridge may be a substantial part of the total intake works cost. The intake, therefore, should be placed as close to the shore as possible. If pumps are to be installed, the service bridge also provides a means of carrying the pipeline and power cables to shore, and it must be substantial enough to truck equipment (such as pumps, motors, and screening segments) from the shore to the intake and simultaneously carry the load of the discharge pipeline. High tower intake and bridge piers may impose relatively heavy, concentrated loads on the foundation—particularly if the reservoir is drawn down to the point where buoyancy is lost. The foundation conditions must be carefully investigated prior to design, and if the intake tower is to be dewatered under full reservoir conditions, the tower weight must be able to resist water buoyancy and prevent flotation.

A typical tower intake including vertical dry well pumps that take suction from a central well is shown in Figure 18-3. Water can be taken from four different elevations. Installing vertical wet well pumps would eliminate the dry well with a substantial cost savings even though the wet well would have to be enlarged to provide adequate spacing between the pumps. The advantage of installing wet well pumps is the reduced cost. The disadvantage is pump maintenance difficulties. If an impeller becomes clogged or damaged, the pump must be pulled to the motor room floor for servicing. In a shallow setting, it is relatively simple to pull the pump, but in a deep setting, such as the one shown in Figure 18-3, it is time consuming. If clogging or pump damage is periodic, the dry well configuration is advantageous because of the relative ease of operations and maintenance. For example, the pump impellers are readily accessible for inspection, maintenance, and repair because the pumps can be disassembled at the pump room floor level without disturbing either the motor or the pump column and without having to pull the entire pump.

A horizontal pump installation would require the exterior shell of the tower to be substantially larger. Furthermore, pump room floor-level motors could be subjected to flooding to the full depth of the intake—about 21 m (70 ft). The summary of the advantages and disadvantages of wet well versus dry well installations and of vertical versus horizontal pumping units is given in Table 25-6.

Alternative Tower Intake Designs

One alternative (if incorporated into a new dam) is to eliminate the pumps but install bar racks and extend a pipe from the bottom of the tower through the dam to a downstream pumping station. The pumps can be horizontal to make them more easily accessible. All piping would be protected from freezing. A valve or sluice gate in the outlet pipe within the intake structure can allow dewatering for periodic pipe inspection. If the dam is made of earth, a number of cutoff walls around the pipe as it passes through the dam should be arranged to prevent seepage, which, if not prevented, ultimately results in failure of the earth dam.

Another variation is to build an intake with two compartments into the upstream face of a concrete gravity dam. Trash racks are installed at the entrance of the first compartment, with openings at various elevations to admit water to the second compartment. A pipe at or near the bottom of the second compartment passes through the base of the dam to a pumping station either at the downstream face or at some convenient point below the dam. This arrangement (as with all intakes) may supply water by gravity to a treatment plant or distribution reservoir without using a pumping station. All intakes may be equipped with fine-mesh traveling water screens if required.

No raw water pumping station can be said to be typical, but many aspects of the concerns noted above are embodied in Example 18-2, in which there is both a side inlet and a tower intake in a river where ice creates problems.

Example 18-2
Raw Water River Intakes and Pumping Stations

Billings, Montana, with a population of 87,000 in 1988, lies in the Yellowstone River Valley at an elevation of 940 to 1100 m (3100 to 3700 ft). The climate is relatively severe with minimum and maximum mean temperatures of -11.2 and -1.2°C (11.8 and 29.9°F) in January and 14.5 and 30.4°C (58 and 86.6°F) in July. Recorded flows on the Yellowstone River have varied from 12 to 1970 m^3/s (430 to 69,500 ft^3/s). In 1987, the average flow was 138 m^3/s (4869 ft^3/s). The turbidity varies from 2 to 1500 nephelometric turbidity units (NTU), and the average in 1987 was 35 NTU. Spring floods occasionally carry large cottonwood trees, and summer flow carries leaves and algae. During the winter, heavy pack ice and frazil ice create serious problems. Pumpage in 1987 averaged 0.8 m^3/s (18.3 Mgal/d or 210 gal/cap · d) and a maximum of 1.79 m^3/s (40.9 Mgal/d—an average of 470 gal/cap · d). The 11 reservoirs have a total storage of 91,700 m^3 (24.2 Mgal).

The site of Billings' water supply is shown in Figure 18-4. Before 1954, however, the only supply was at Inlet No. 1, which then consisted of three fixed screens at the shore line.

Problem: The problems prior to 1954 were (1) to obtain a reliable supply of water at low river flow, (2) to be able to handle the high turbidities more effectively, and (3) to achieve an overall reliability greater than could be obtained with a single inlet.

Solution: The solution to these problems consisted of (1) constructing a tower intake (called Inlet No. 2) at the deepest portion of the river, and (2) improving the original inlet.

Inlet No. 2. The construction of the intake tower is shown in Figures 18-5 and 18-6. The tower is connected to the shore by a walkway bridge with the pump suction pipes buried under the river bed, as shown in Figure 18-7a. The bedrock is shale soft enough for steel sheet piles to be driven into it. The low service pumping station is on the shore. The pump room floor is low enough to provide flooded suction for the horizontal split-case pumps. One of the smaller pumps is shown in Figure 18-7b.

Inlet No. 1. Some years later, a staged series of improvements were made at Inlet No. 1. The first was the construction of six 1200-mm (48-in.) reinforced concrete pipes (RCPs), which extended for some distance into the river. The second was the construction of the contact basin, the diffuser baffle, and the L-structure (Figure 18-8) designed to collect the flow and pass it through a rapid-mix basin consisting of two vertical mixers in tandem. Chemicals are piped to the L-structure from the treatment plant nearby. Potassium permanganate is added at the rapid-mix basin all year. Alum is also added from March through September, during which time the turbidity exceeds 20 NTU and complete treatment is desirable. Flocculation takes place in the contact basin, sometimes assisted by two horizontal propeller mixers driven by

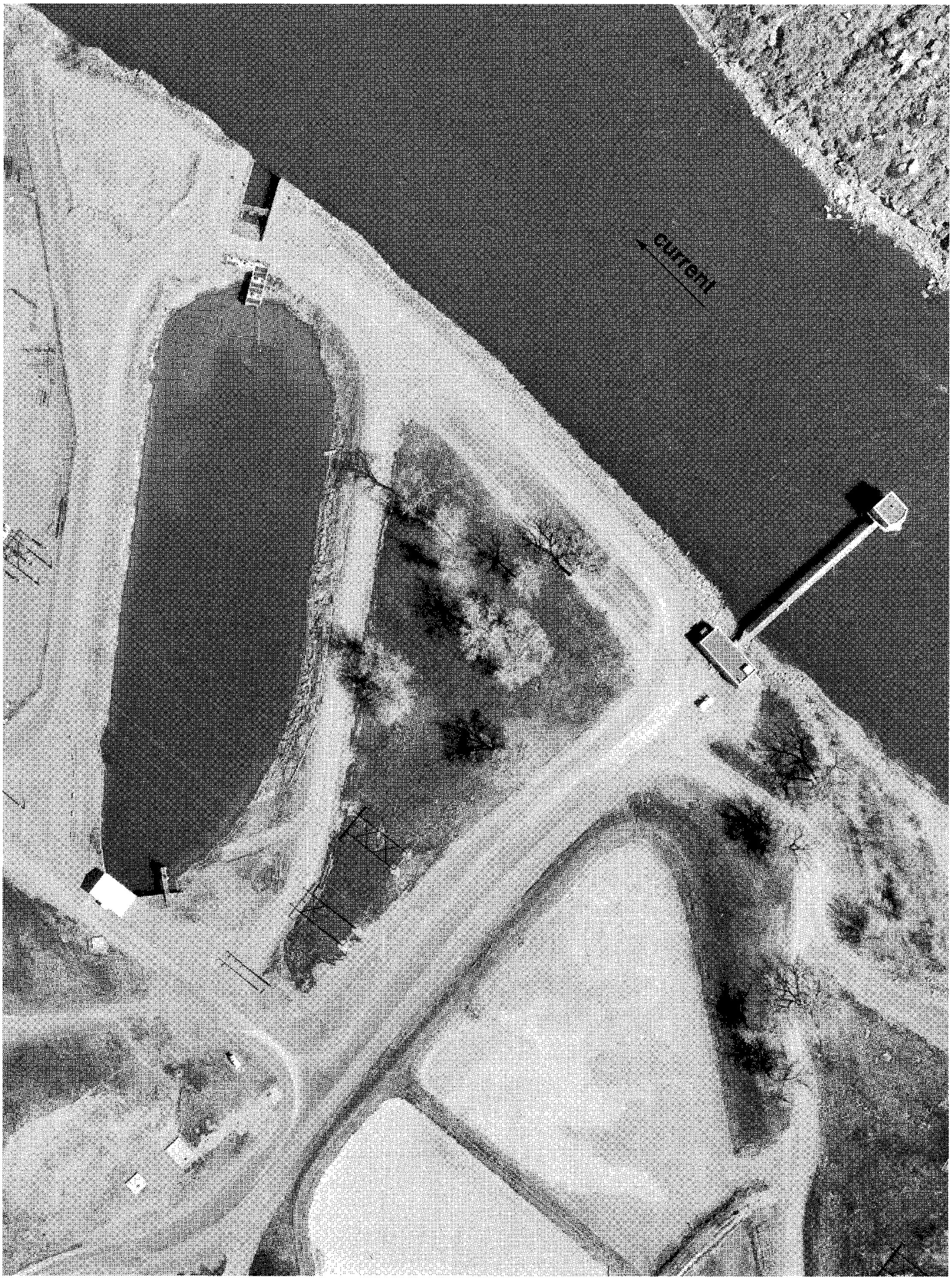


Figure 18-4. Site of the City of Billings Intake No. 1 (upper left) and Intake No. 2 (right center). Aerial photograph courtesy of Christian, Spring, Sielbach & Associates.

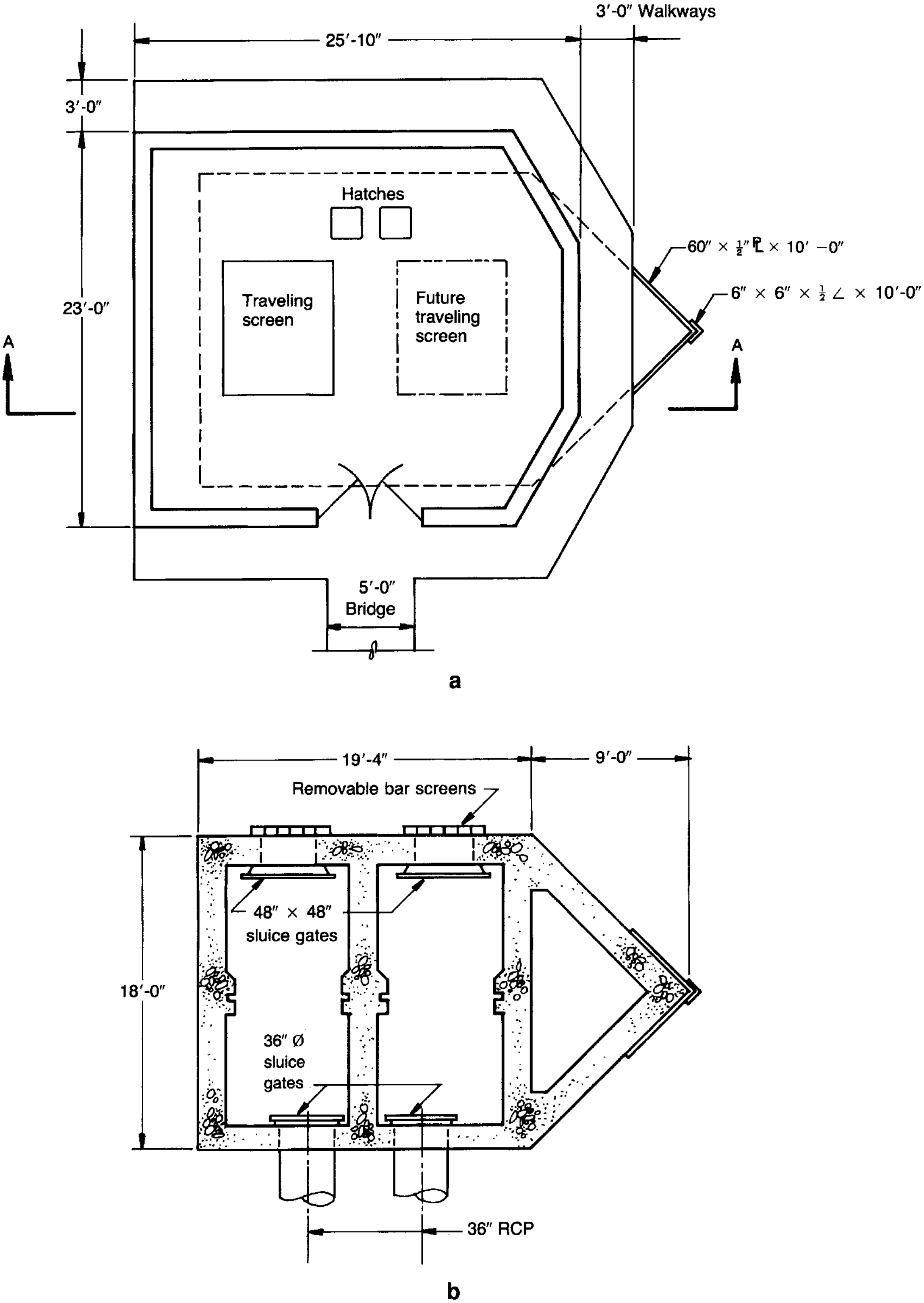


Figure 18-5. Tower intake, Inlet No. 2. (a) Floor plan; (b) Section B-B from Figure 18-6.

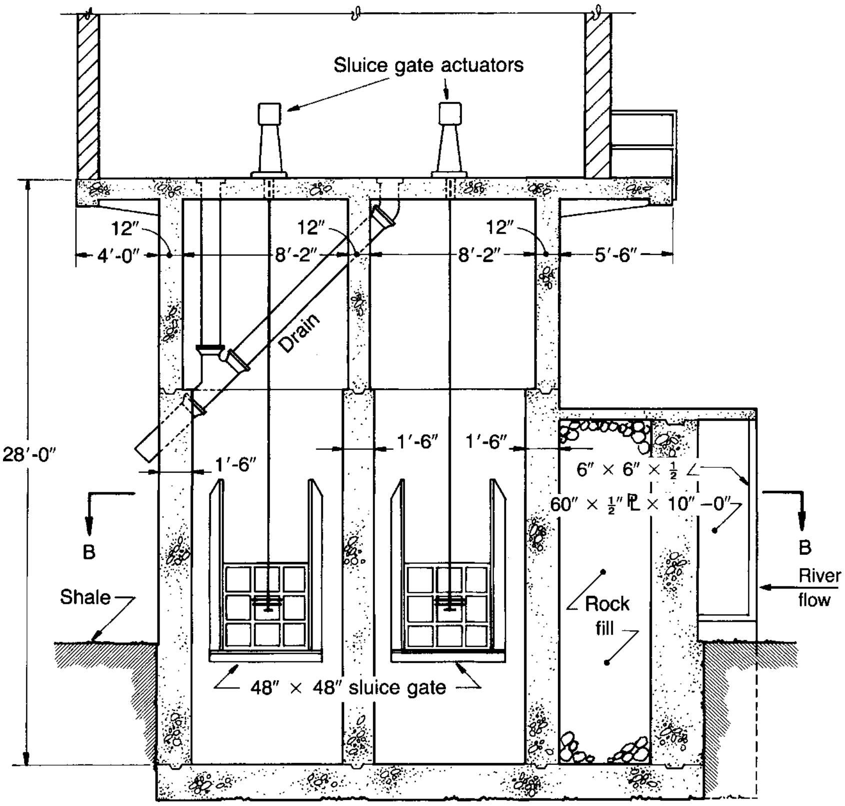


Figure 18-6. Section A-A from Figure 18-5.

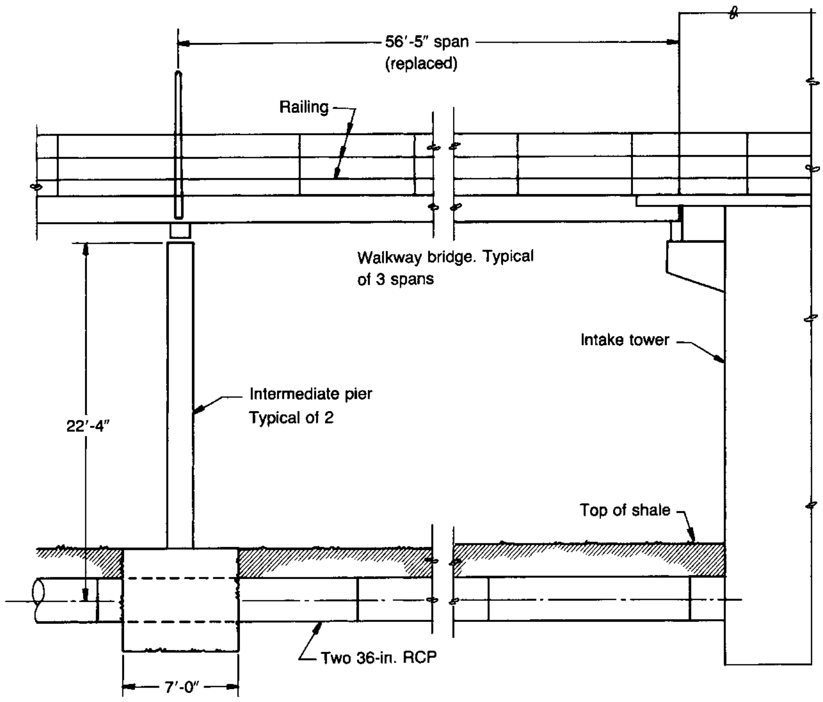
submersible motors carried on barges. Water is generally treated by direct filtration from October through February. The diffuser baffle in the contact basin inhibits short circuiting.

A third improvement at Inlet No. 1 was to drive sheet piling along the inlet and river both to protect the banks and to make it easier for a backhoe to remove sand and silt from the inlet. The next improvement was to cut off the six 1200-mm (48-in.) pipes and install canal gates at the cut ends. The final improvements were to construct the inlet baffle with three skimmer gates at the entrance of the inlet channel. The skimmer gates are raised and lowered in separate grooves by cables and hand winches so that a slot of any desired height can be positioned at any desired elevation to exclude sand and silt and to prevent the entrance of floating debris. Because there is almost no hydrostatic force on the skimmer gates, they are lightly built.

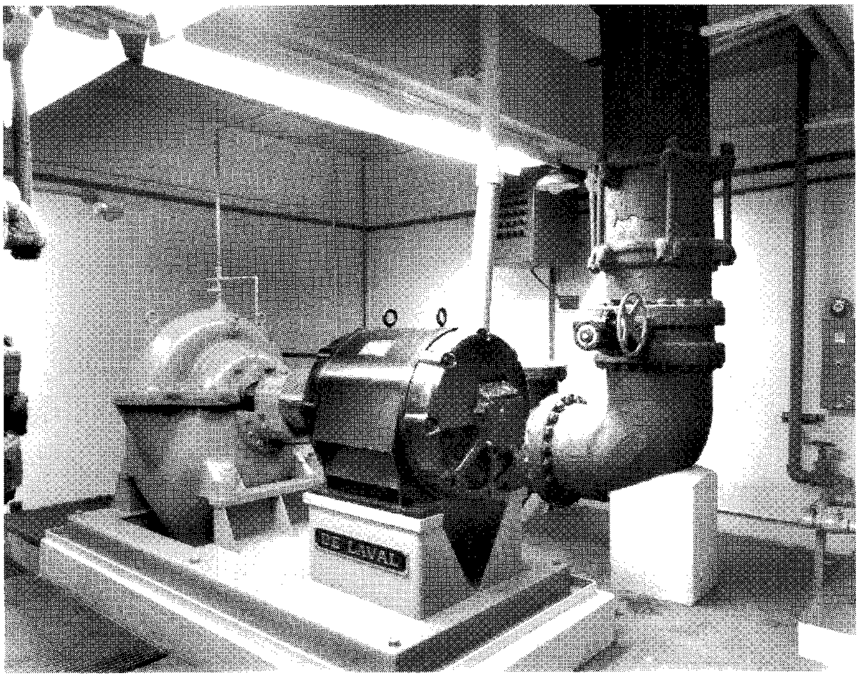
Operational experiences. Inlet No. 1 has been very satisfactory. Frazil ice is not a problem, and debris is kept out by positioning the three skimmer gates to draw water from below the water surface but above the river bottom to exclude heavy materials. Once a year, sand and silt are removed from the channel with a backhoe. The contact basin is cleaned with a floating dredge, a job that takes all summer to complete. The dredged material is stored in holding ponds.

Inlet No. 1 is used most of the time, partly because it is trouble free, but—more important—because it contains the L-structure, which is used as part of the treatment process.

Inlet No. 2. Inlet No. 2 is used for emergency service when Inlet No. 1 is closed for repairs; it is also used during periods of very low flow when the water can be treated by direct filtration. Frazil ice causes serious problems at times. Because it clogs the steel screens, which must be cleaned by hand (an onerous task), Inlet No. 1 is used (if possible) whenever frazil ice is forming.



a



b

Figure 18-7. Bridge, suction pipes, and pump at Inlet No. 2, Billings Water Treatment Plant. (a) Bridge and pump suction pipes; (b) one of the four horizontal split-case pumps. Photograph by R. L. Sanks.

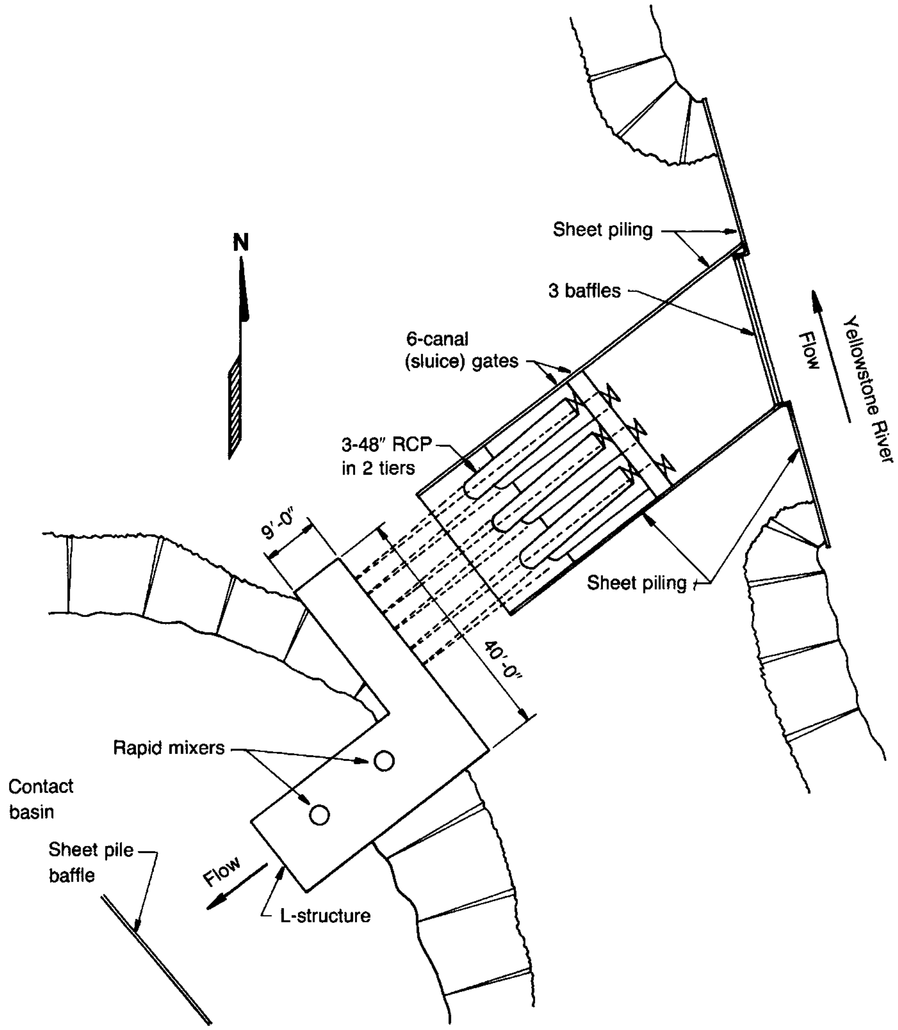


Figure 18-8. Inlet No. 1 at Billings Water Treatment Plant.

Pack ice causes ice jams and, at times, a pileup of ice at both the intake tower and the walkway. Ice jams have formed downstream, and they have caused the water level to rise 3.3 m (11 ft) in 25 min. Water has, at times, flowed over the bridge. Large cottonwood trees are sometimes caught either on the intermediate piers or on the walkway and can damage railings. To remove the trees, a cable must be attached and they must be winched upstream where the branches can be sawed off. Workers are reminded of the hazards of this duty by the shuddering of the bridge in such circumstances. A section of the bridge was once carried away. The walkway and one intermediate pier have since been repaired, and the nose of the intake tower has been sheathed with 12-mm ($\frac{1}{2}$ -in.) steel plate.

Object lessons for designers. Designers of river works would do well to be guided by thoughtful contemplation of the three decades of operating experiences described above. Some conclusions are as follows:

- Side inlets are generally satisfactory if the problem of low water can be circumvented. Perhaps training dykes or a low weir would help to supply water during low flows.

- Any inlet may have to be closed for repairs. Either design so that an inlet can be partially closed or provide for an alternate emergency inlet.
- Make a very thorough study of the river and its seasonal variations before beginning to design the river works. Studying records of low and high flow is not enough; frazil ice, pack ice, trees, debris, and fish can create serious problems.
- If pack ice or trees can be carried by the river, search for the highest ice jam that has occurred. Interview local residents and use ice scars on tree trunks for data. Set floor elevations of towers and walkways high enough to exceed the previous records, and then add another 3 m (10 ft) of elevation for additional safety.
- Consider designs that tend to keep pack ice away from the mouth of the intake. Ice tends to pile up on protruding objects in a river.
- Avoid intermediate piers in walkways. A single span for such light loads can easily bridge 60 m (200 ft).
- Sheath the nose of any towers or piers with steel up to the top of the highest ice jam that can occur.
- With reference to Figures 18-5 and 18-6, some engineers would prefer (1) thicker walls—particularly the nose walls, (2) anchoring the nose wall to the underlying shale with heavy reinforcing steel bars (or even railroad rails) grouted into deep [e.g., 6-m (20-ft)] holes to prevent overturning, and (3) using flanges or grooved-end couplings (for longitudinal restraint and to allow for movement due to temperature) instead of bell and spigot joints in any pipe (such as the sloping floor drain in Figure 18-6) that is not buried.
- For gratings and screens, consider the use of solid plastics or plastic coatings on metal or other measures to combat frazil ice. At one time, excess steam from a nearby power plant was considered for keeping frazil ice off the screens, but the heating requirements were enormous. Locating the intake in a quiescent water zone and keeping the intake velocities very low is a good defense.
- Note that pack ice is unpredictable both in its ability to form ice jams and in the forces it imposes on structures. Because failures of bridge piers elsewhere in the United States have occurred with loss of life, river works that are subject to extraordinary forces of nature should be designed very conservatively indeed. Calculations are meaningless when the forces cannot be accurately quantified, so follow the old engineering adage to "build it Hell for stout."

18-4. Raw Water Pumping from Aqueducts

Functionally, low-service water pumping to a treatment plant from a raw water aqueduct is somewhat similar to booster pumping. However, the booster pump in an in-line pumping station must be coordinated with the supply pumps—not an easy problem. But with raw water pumping, only a portion of the flow is taken and the problems of coordination are less severe or even absent. Furthermore, the pump suction head in pumping from a canal or aqueduct does not usually fluctuate very much, whereas the suction head from a reservoir, for example, might vary by 30.5 m (100 ft) or more.

The design of a pumping station taking suction from an existing raw water aqueduct is shown in Example 18-3.

Example 18-3
Raw Water Pumping from an Aqueduct

The project, located in the north San Francisco Bay area in California, was needed to supply 5 Mgal/d of raw river water to an existing 5 Mgal/d municipal water treatment plant (operated 24 h/d) in which the process train consists of chemical mixing, flocculation, sedimentation, filtration, and chlorination. There are four filter beds, each with a sustained capacity of 875 gal/min, although the use of two beds (at 1750 gal/min) is average. The treated water enters a small clear well with a capacity of 150,000 gal, and from there it is pumped to a series of distribution storage reservoirs on the basis of demand.

The raw water is high in turbidity, particularly during the spring and early summer period of maximum river flow. The treatment plant was previously supplied from the same raw water source via a 40-year-old 36-in. diameter pressure pipeline that passes by the treatment plant site (see Figure 18-9). In the new system, the source of water is a surge-controlling standpipe (see Figure 7-4) 150 ft in diameter and 22 ft high on a new 72-in. state aqueduct. A 14-in. isolation valve and a flow-metering station to supply the treatment plant are provided as a part of the state aqueduct system. The ground surface profile along the supply pipeline is shown in Figure 18-10.

The long-term future demand was not expected to exceed the previous plant capacity of 5 Mgal/d, but the treatment plant was then 35 years old (in 1987) and due for rehabilitation, and the treatment process needed to be upgraded to accommodate revised U.S. EPA water quality standards.

Problem: Design a system to supply the municipal treatment plant with water from the standpipe on the 72-in. state aqueduct. The municipality insists on variable output at any flow from 875 to 3500 gal/min.

Solution:

Design decisions. The following are the major design decisions that were required:

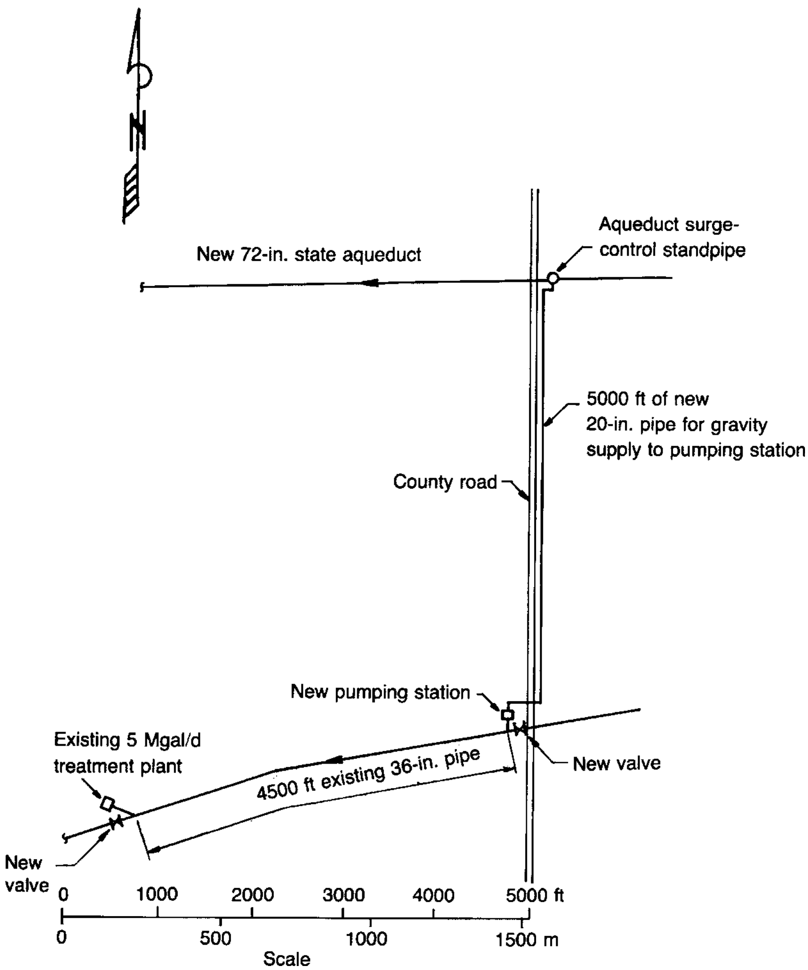


Figure 18-9. Site plan, Example 18-3.

LIVE GRAPH
Click here to view

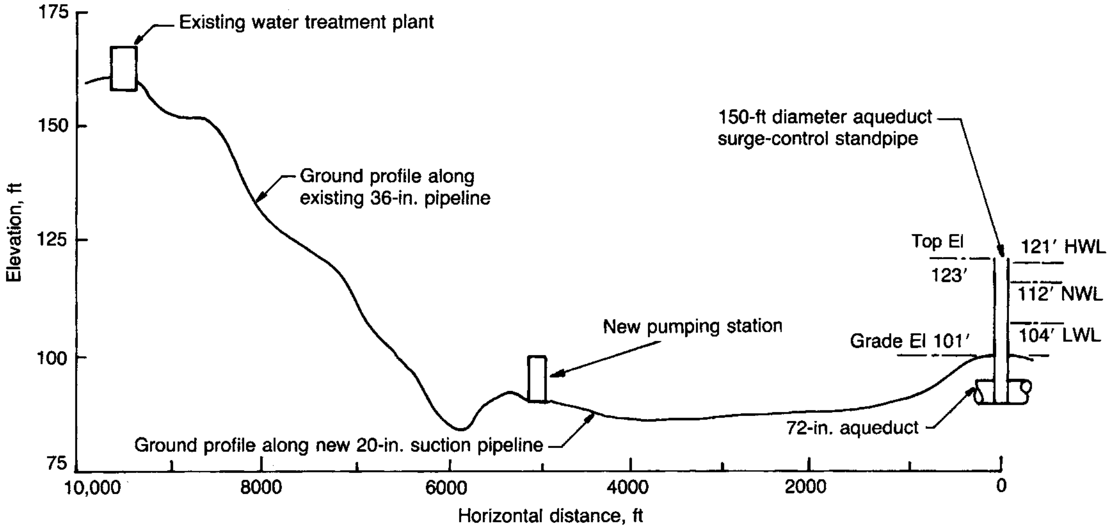


Figure 18-10. Profile of pipelines. For the existing water treatment plant, the influent channel HWL = 160 ft. For the new pumping station, floor elevation = 89 ft, grade elevation = 88 ft, and pump suction elevation = 76.5 ft.

- Pumping station location
- New pipeline requirements and size
- Type, number, and capacity of the pumps
- Type of driver
- Type of pumping station structure
- Pump-control system.

Pumping station location and pipeline route. The location considerations included:

- Gravity supply from the surge tank
- Site availability
- Pipeline right-of-way
- Access
- Power supply
- Environmental factors
- Security
- Minimum new pipeline construction.

These considerations led to the selection of the site shown in Figures 18-9 and 18-10. Any location between the standpipe and the selected site would have been satisfactory if land were available. The alignment paralleling the county road fell within an existing pipeline easement and also permitted the use of the existing 36-in. pipeline to deliver the pumping station discharge to the treatment plant.

A direct route from the standpipe to the treatment plant was discarded for several reasons:

- A greater length of new pipeline construction
- Right-of-way acquisition through multiple property ownerships
- To avoid environmental impact.

New pipeline. With the selected pumping station location, the new discharge piping is limited to the pump manifold and the connection to the existing 36-in. pipeline. Upstream and downstream valves are needed to isolate this portion of the 36-in. pipeline.

A present-worth analysis was performed on a range of pipe sizes for the (approximately) 5000 ft of suction pipeline from the 14-in. standpipe connection to the pumping station. Based

on the installed cost of ductile iron pipe (the owner's preference), the power cost for average flow conditions at \$0.065/kW · h per 25-year life expectancy, and interest at 8.0%, a pipeline 20 in. in diameter was determined to be the most economical size.

System curve. Pipeline friction losses were calculated by first determining the equivalent length of each size of pipe in the system to take into account system entrance and exit losses and losses due to fittings (elbows, tees, increasers, and reducers), flowmeters, and valves. With equivalent pipe lengths tabulated, system friction losses were calculated by a "pipe loss" computer program using the Darcy–Weisbach formula with the following relative roughnesses, ε/D:

Pipe	ε/D
Ductile iron (12, 14, and 20 in.)	0.003 in. (cement lining)
Steel (10 and 14 in.)	0.0005 in. (asphalt coating)
Old cement-lined steel (36 in.)	0.008 in. (bad shape)

The results of calculations for maximum static lift and total dynamic head (TDH) are tabulated below, and the system H-Q curve is shown in Figure 18-11.

Flow, gal/min	Pumps operating	Friction loss, ft[a]	Static lift, ft	TDH, ft
3500	2	48	56	104
1750	1	12	56	68
875	1	3	56	60

[a]For 5000 linear ft (LF) of new 20-in. pipe, 4500 LF of old 36-in. pipe, and all valves, meters, and fittings.

Pump suction conditions at maximum flow are as follows:

Minimum standpipe water level	+104 ft
Suction piping loss at maximum flow	17 ft
Pump suction elevation (centerline of connection to pump barrel)	76.5 ft
	Subtotal +11.5 ft
Barometric pressure head	+33.9 ft
Vapor pressure head at 70°F	0.8 ft
	Subtotal +44.6 ft
Safety factor at 10%[a]	4.5 ft[a]
NPSHA (worst condition)1	Total +40.1 ft

[a]However, readers are warned that a safety factor of less than 5 ft or less than 1.35 × NPSHR may be risky (see Section 10-4 and Figure 10-13).

Pump and driver selection. The most suitable type of pump for this application was considered to be a vertical turbine pump with the bowls encased in a vertical "can" or suction barrel with a flanged connection to the suction pipeline manifold. One alternative would be a horizontal split-case centrifugal pump, but the vertical turbine pump is lower in cost, occupies less floor space (thus reducing the cost of the building), and (with the concept of using multiple bowls) is more flexible.

The logical number of pumps to satisfy maximum and average flow requirements would be two units with a capacity of 1750 gal/min each to deliver the maximum flow of 3500 gal/min at

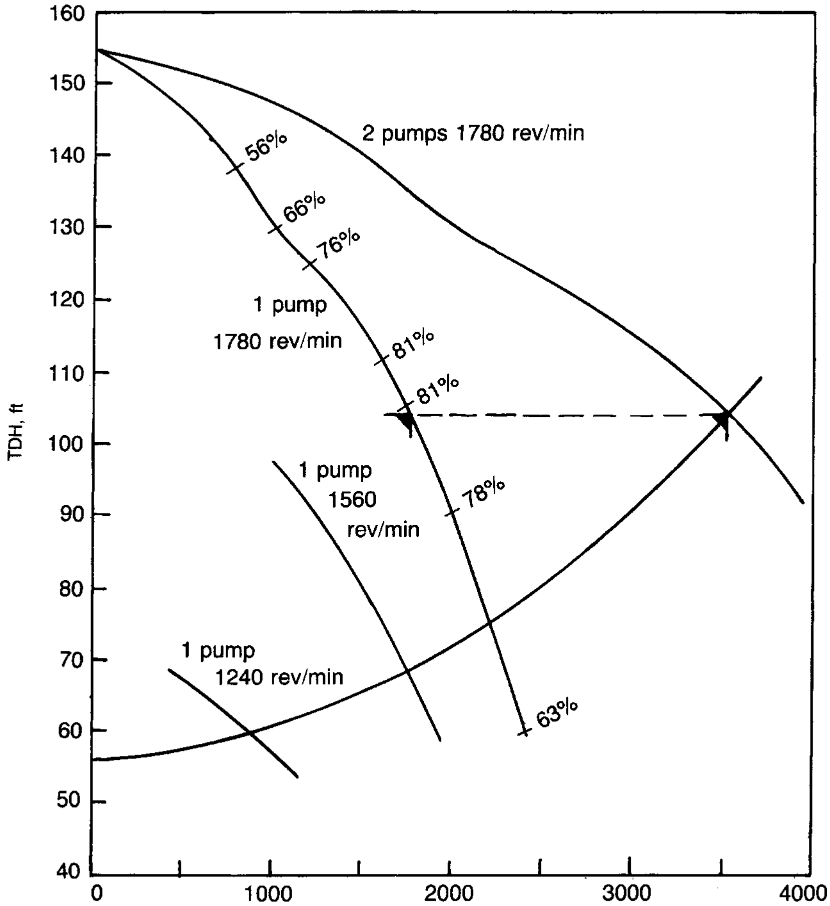


Figure 18-11. System head-capacity and pump characteristic curves.

a TDH of 104 ft. One pump operating alone would deliver more than 1750 gal/min due to the reduction in pipeline friction loss. If constant-speed pumps were used, the flow rate controller at the treatment plant could compensate for this difference, but to supply the minimum flow requirement of 875 gal/min would require either a third, smaller pump sized for this condition or excessive throttling (by the flow rate controller). In this application, however, the owner specified pump drives for operation at any flow between 875 and 3500 gal/min; hence, variable-speed (V/S) drives are required. The V/S drives permit the use of two identical duty pumps of 1750 gal/min capacity. A third unit is used for standby service.

Adjustable-frequency drives (AFDs) were selected because of their higher efficiency in comparison with eddy-current couplings (but see Figure 15-25, Section 15-11, and Table 15-3).

The pumps selected were two-stage, barrel-type vertical turbines to discharge 1750 gal/min at a head of 104 ft with 7.6-in. impellers at 1770 rev/min. The minimum permissible flow per pump is 600 gal/min. The pump performance curve is shown in Figure 18-11. Curves for reduced speeds were obtained by using the affinity laws (see Section 10-3), which are quite accurate for these pumps. The power output of the pump is found from Equation 10-6b

$$hp = \frac{qH}{3960} = \frac{1750\,\text{gal/min} \times 104\,\text{ft}}{3960} = 46$$

As noted in Table 15-3, the efficiency of AFD converters is about 96%, the motor efficiency is also about 96%, but extra motor losses are about 3% and transformer losses are about 2%, so the net AFD efficiency is about 88%. Because the pump has an efficiency of 80.5%, the total motor output must be at least

$$hp = \frac{46}{0.88 \times 0.805} = 65$$

A 75-hp motor would be needed.

Pumping station structure. Although the pumping station is located in a temperate climate suitable for an outdoor installation, the owner elected an indoor installation to reduce and facilitate maintenance, to protect the equipment better, and to improve security. The foundation conditions at the site were favorable. The building is of concrete block construction with a pier supported, beam-and-girder concrete foundation and concrete floor and roof. Eight poured-in-place piers 2 ft in diameter by 20 ft deep were installed, one at each building corner and two along each side wall. Inside dimensions are 32 ft by 18 ft in plan and an interior height of about 9 to 9.5 ft. Removable 6-ft-square roof hatches are located over each pump to facilitate installation and removal. In addition to the pumping units, the station houses the electrical switchgear and control equipment, including the AFDs.

Ventilation is provided by air inlet screens and three 3500 ft^3/min wall-mounted exhaust fans. Building heating is not required. Pump suction piping connects to the pump barrels 12.5 ft below the floor level. The above-floor pump discharge passes through a slow-closing check valve and shut-off valve before turning down through the floor with a 90-degree bend for below-grade manifolding and connection to the existing 36-in. pipeline. The plan and a section are shown in Figures 18-12 and 18-13.

Pump-control system. The treatment plant is staffed 24 h/d, but the pumping station is visited only periodically for inspection and maintenance. The control system is therefore

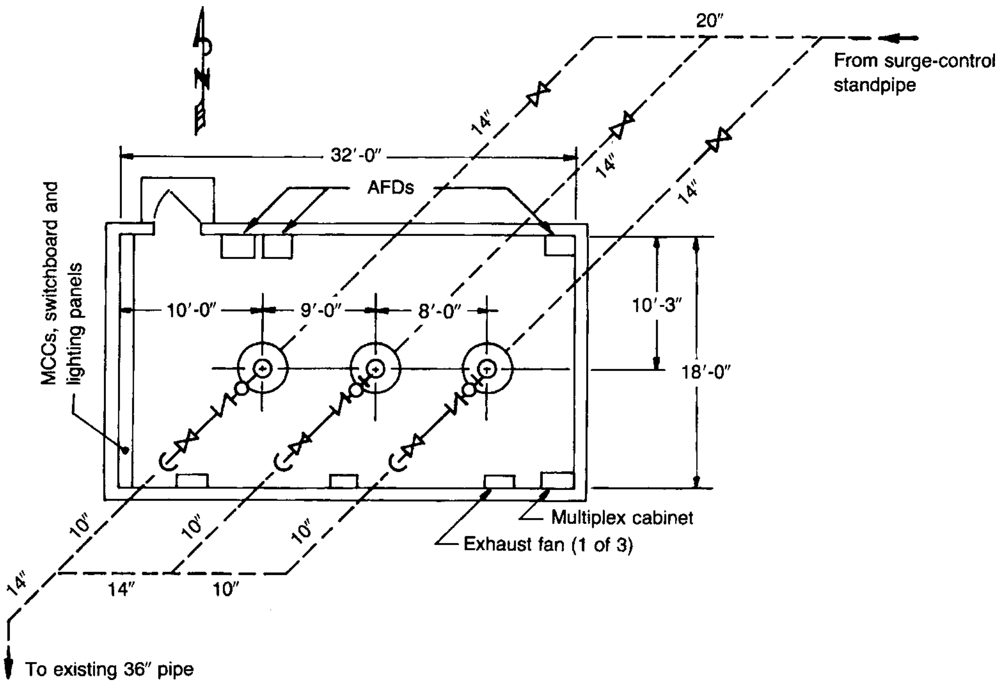


Figure 18-12. Plan of the pumping station.

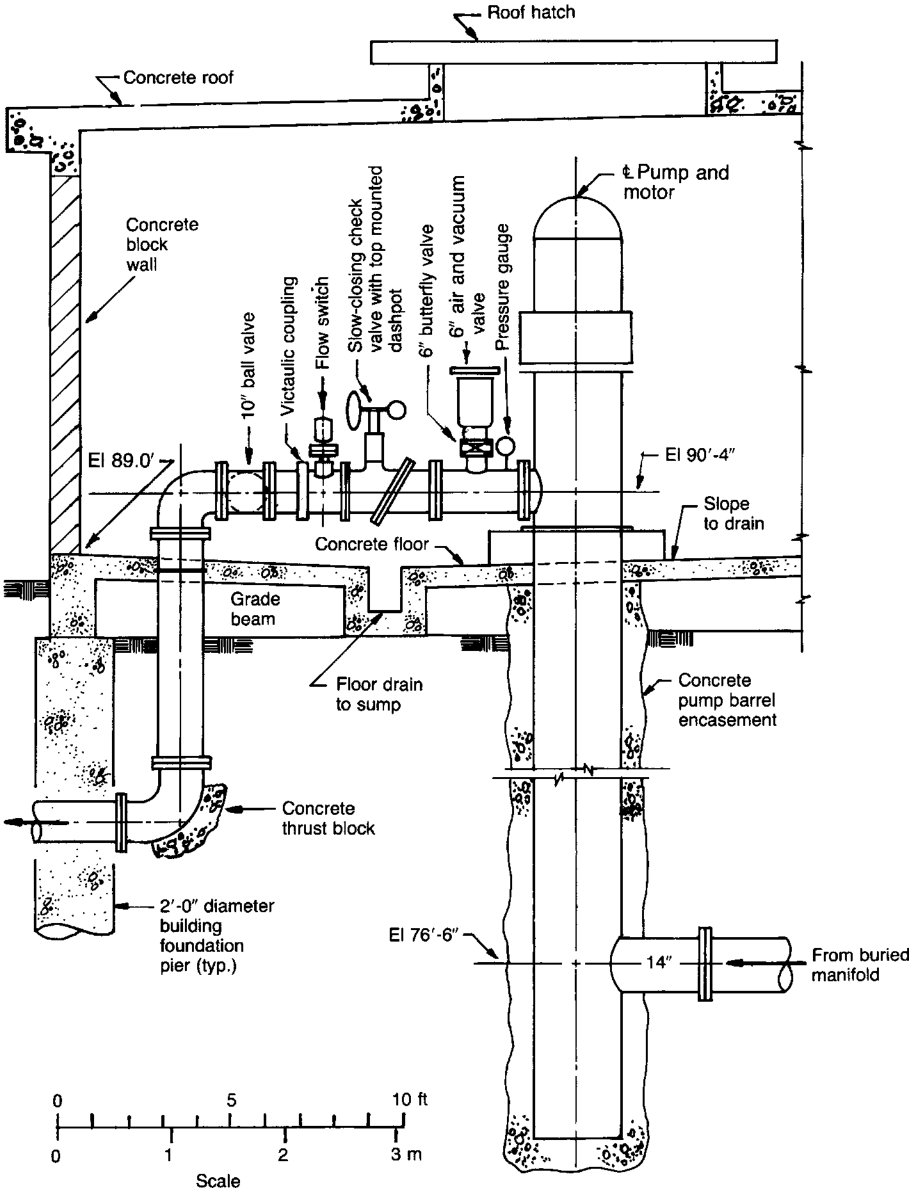


Figure 18-13. Section through the pump (typical of all three pumps).

designed for remote control from the treatment plant. The treatment plant operator can select which pumps to operate and can set the desired flow rate. A "local/remote" switch is provided at the AFD cabinet at the pumping station. "Local" operation at the pump house is used only for maintenance and testing.

A multiplexing unit between the pumping station and the filter plant control room links the input–output signals over telephone lines for remote start–stop and analog display and control of the pumps. Analog signals for pressure (at each pump suction and at a common point on the pump discharge), level, flow, and speed are multiplexed for remote indication at the filter plant control room panel. Pressure gauges are also mounted on each pump discharge for local observation. At the control room panel, a three-position switch selects the pump

group (1–2, 2–3, 1–3), and another switch selects the lag-lead pump. If pump No. 1 is selected as the lead pump and group 1–2 is selected to run, then pump No. 3 is automatically on standby. A start button will start pump No. 1 as the lead pump.

A flow input signal compares the flow set point against the actual flow and adjusts the pump speed until maximum flow is reached. If the flow requirement is greater than the flow output of pump No. 1, pump No. 2 will start automatically and share the load with pump No. 1 until flow requirements have been fulfilled. A reverse sequence occurs when flow requirements decrease. Overriding emergency controls to shut off the operating pumps automatically are:

- A high water level in the treatment plant influent channel
- Low flows (which indicate possible pump failure)
- Low pump suction pressure and high pump discharge pressure.

A schematic process and instrument diagram (P&ID) of the system is shown in Figure 18-14.

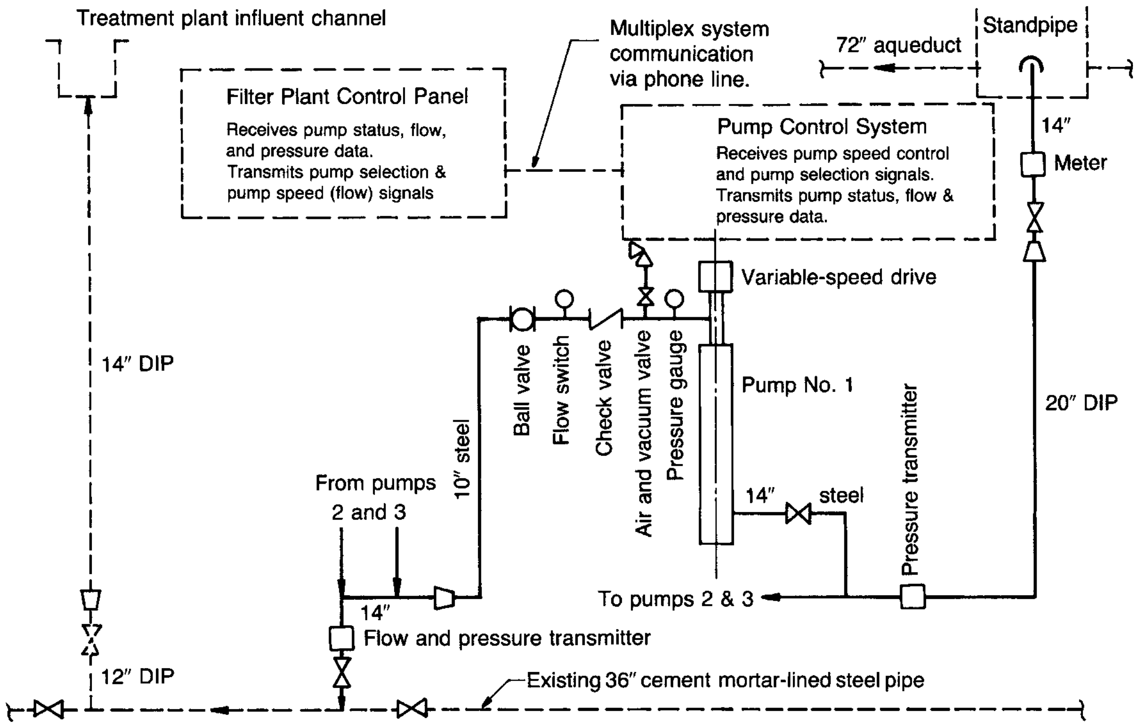


Figure 18-14. Piping and instrumentation diagram for Example 18-3.

Critique of Example 18-3

In most circumstances, cement-mortar lining is superior to asphalt coatings and should always be considered (see Sections 3-2, 3-3, and 4-1). Some waters are aggressive toward cement mortar, so asphalt and other coatings (Table 4-8) are still used.

The use of variable-speed pumps in this example would normally be unjustifiable. Flows close to 100, 50, and 25% of a given flow rate can be achieved by using constant-speed pumps of two sizes. The operating time per shift at the treatment plant could be adjusted to permit the plant to produce either 3500 gal/min with four filters in use or 2250 gal/min with three filters in use. Thus, the filters would always operate within 15% of rated capacity. If the owner requires V/S drives despite their disadvantages (see Section 15-1), the engineer in a practical world has no choice.

The use of a slip drive (such as an eddy-current coupling) instead of an AFD may be worth an extensive investigation. In Example 29-1 of the first edition of this book, the eddy-current coupling was slightly more cost-effective than an AFD despite the greater efficiency of the AFD. Comparisons depend on the effect of a frequency analysis of expected flow rates on the cost of energy; they also depend on capital cost. Eddy-current couplings are becoming more expensive and, as few companies make them any more (except for MagnaDrive), they are becoming obsolete. On the other hand, AFDs are becoming cheaper. Consider also obsolescence, replacement cost of short-lived components, maintenance, and reliability.

Fittings in the discharge header are crowded. The pressure gauge location (so close to the discharge elbow) may induce some inaccuracy and fluctuation in the gauge readings. The combination of one grooved-end coupling and a slanting flange makes dismantling easy; another solution would be a grooved-end coupling on the discharge elbow and another in a horizontal plane just above the floor at the down-turned elbow. A pair of sleeve couplings (not shown on the drawings) separated by two or three pipe diameters is normally needed outside of (but adjacent to) the building to allow for differential settlement. As this building is set on piers, the sleeve couplings were not needed. Refer to Example 12-3 for a comparison of design approaches.

18-5. Well Pumps with Elevated Tanks

The most common well pumps are vertical turbines. High service pumps connected to a clearwell are usually of horizontal split-case, vertical split-case end-suction, or vertical turbine barrel design (see Chapter 11). Selection criteria and a discussion of the advantages and disadvantages of vertical turbine pumps are presented in Section 25-6.

As discussed in Chapter 4, cement-mortar-lined ductile iron or steel pipe is most frequently used. Isolation valves are usually gate or butterfly valves (see Chapter 5). Check valves should be either the swing type with outside lever and spring or outside lever and counterweight type with an oil-filled dashpot. The slanting disc type with a top-mounted dashpot and/or bottom buffer can also be used (see Chapter 5). If very quick closure is required, the center post "silent" check valve is a good choice. If valve slam or water hammer (or both) are likely to occur with a check valve (see Chapters 5, 6, and 7), a pump-control valve should be used instead. Pump-control valves are usually used if the piping is larger than 300 mm (12 in.). The purpose of these valves is to control surges in the discharge piping that occur when the pump starts and stops. The pump motor control center (MCC) is interconnected with the pump-control valve so the pump starts and stops against a closed valve. In discharge manifolds up to about 400 mm (16 in.) in size, these valves are frequently of the diaphragm-actuated or piston-actuated globe type. In larger installations, ball or cone valves equipped with motorized, pneumatic, or hydraulic actuators are frequently used.

Climate Considerations

Heat, cold, wind, dust, precipitation, and moisture are natural elements that must be considered in the design of any waterworks facility. Weatherproof enclosures for electrical equipment and devices and well-maintained paint, coatings, and linings for pumping units, piping, valves, and appurtenances may be adequate in moderate climatic conditions. More severe climatic conditions require additional protective measures. A good starting point is a field inspection of existing facilities in the proposed project area in which the type and adequacy of protection are being investigated.

High temperature is primarily a concern for electrical equipment such as switchgear, motor starters, and instrument control panels. Temperatures exceeding 30°C (85°F) tend to reduce equipment life and affect the accuracy of instruments. Increasing degrees of protection of equipment from excessive heat can be provided as follows:

- Provide sun shade
- Enclose in a vented building with an exhaust fan
- Enclose in an air-conditioned building.

Temperatures below freezing are of great concern. Short-duration temperatures slightly below 0°C (32°F) require that small-diameter piping, valves, and water-holding appurtenances be protected against freezing. In colder climates, *all* of the piping, valves, and appurtenances must be protected against freezing and, in addition, structures and piping must be protected from damage due to frost heave. Piping, valves, and appurtenances that need not be accessible may be protected from both freezing and the effects of frost heave by being buried below the frost line. Piping, valves, and appurtenances that must be accessible may be given increasing degrees of protection from freezing as follows:

- Wrap with electrical "heat strips" and insulating materials (called "heat tracing"). Some engineers

report poor experience with heat tracing and find that it usually lasts three years or less.
- Enclose in a vault with or without insulating materials or space heaters.
- Enclose in a heated building.

Windblown sand and dust particles can damage painted or coated surfaces, bind moving parts, infiltrate electrical equipment cabinets and enclosures, and increase the maintenance. Electrical equipment must be maintained in a dust-free condition to provide quiet, reliable service. Enclosing station equipment and piping in an air-conditioned building offers the best protection from windblown sand and dust particles. Protection against precipitation and moisture is provided by:

- Properly specified, applied, and maintained paints and coatings
- The use of weatherproof enclosures for exposed electrical equipment and devices
- The use of space heaters in vaults or motor enclosures
- Enclosure in a climate-controlled building.

Using buildings to protect pumping station equipment from the outside environment requires careful consideration of the resultant inside environment:

- Enclosing electrical equipment in a vented building provides a degree of protection from heat but not from dust.
- Enclosing electrical equipment in an air-conditioned building provides heat and dust control and a more pleasant environment for service or operating personnel.
- Enclosing pumping units in a heated building to protect them from winter cold requires proper ventilation and, perhaps, cooling to prevent temperature buildup in the summer due to motor or engine heat.
- Variable-speed motor controllers produce significant amounts of heat that must be properly vented if enclosed in a climate-controlled building; generally, most switchgear is rated for a 40°C (104°F) ambient temperature, and it is difficult to obtain switchgear of higher ratings.

Wells

Location

The water supply of many (perhaps most) small communities comes from wells. Hydrogeologists, well drillers, and engineers share the responsibility of locating, testing, drilling, and casing wells. An engineer usually selects the pumps, designs the system, and assumes overall responsibility for the project. The location of wells depends on many factors, including:

- The distance from other wells and the possibility of mutual interference of drawdown surfaces
- The distance from possible pollution sources, such as 30.5 m (100 ft) from a known septic tank or downstream from a chemical or solid-waste dump site
- The land available for purchase or lease
- The location of a good groundwater aquifer; information can be obtained from local well drillers, previous well-drilling reports, health departments, or geological surveys.

Water Quality

A test well is normally drilled to determine the sustained yield, drawdown distances, and water quality. As the well is drilled, a soil log is kept to record soil type versus depth. This information, together with an electric well log, is used to determine the best water-bearing soil strata. The electric well log (a measure of the soil's resistivity) can be used by an experienced technician to determine at what soil strata levels a well screen should be positioned to provide the desired yield. In several states, such records must be submitted as part of the permit process for wells. A chemical analysis of the water in each aquifer should be made to select those aquifers with the highest water quality commensurate with suitable yield. Unwanted aquifers can be sealed.

Water Treatment

Well waters usually have a high content of dissolved solids and are usually "hard." Hard waters can be softened (and often demineralized to some extent) by lime-soda treatment, or they can be either softened or demineralized by such means as ion exchange or reverse osmosis [7]. All of these treatments are expensive. Well waters for potable use should be chlorinated even if the water from the aquifer contains no bacteria. Because both wells and distribution systems can become contaminated, residual chlorine is needed to protect the public health. Unless the well water is clean and free from sand and silt, a settling basin (or even a filter) is required for a municipal well supply. Many wells contain gases (such as carbon dioxide,

methane, and/or hydrogen sulfide) that can be removed by aeration or, more completely, by vacuum. Iron and manganese can also be removed by aeration, lime-soda softening, or chlorination followed by filtration, or they can be removed by ion exchange.

Casing

Wells are always cased—usually with steel. The casing can be placed against the soil, or the bore hole can be enlarged to permit the placement of a layer of gravel, usually at least 150 mm (6 in.) thick, between the soil and casing. The "gravel pack" increases the yield of the well and reduces the velocity of inflow at the face of the soil, which thereby reduces the inflow of sand and silt. Samples of water and soil from the test hole can be used to design the gravel pack for maximum effectiveness in preventing the entrance of particulate. In the aquifer, the casing is either perforated at closely spaced intervals by a special tool to allow water to enter or the casing is interrupted by well screens.

Development

The well is "developed" by pumping at a high rate (considerably higher than normal pumping rates) to wash fine particles out of the aquifer so that it can behave like a large, underloaded sand filter and, thus, produce clear water. Some wells produce sparkling, clear water; others never stop producing silty water. During a well's development, nearby wells or bore holes can be tested to establish the drawdown curve and to aid the hydrogeologist in predicting the safe yield. Walton [8] describes groundwater tests in detail, and his text is accompanied by a diskette containing test programs in BASIC.

Well Head

The design of the valving system at the well head is critical. When the pump starts after a period of rest, the water (at water table level) flows at high speed to the well head because the TDH is initially zero. If the speed is not reduced by the cushion of air in the well, the pipe or well casing may break and pumps may be displaced. The air must be allowed to escape slowly enough so that the moving column of water in the well strikes both the closed check (or pump-control) valve and the stationary column of water in the transmission main too gently to cause an undue pressure surge. If the first surge of water from the aquifer contains sand and silt, the water is usually wasted. Valving arrangements for well heads are shown in Figures 7-9, 7-10, and 7-11 in Section 7-6, as well as in Example 18-4.

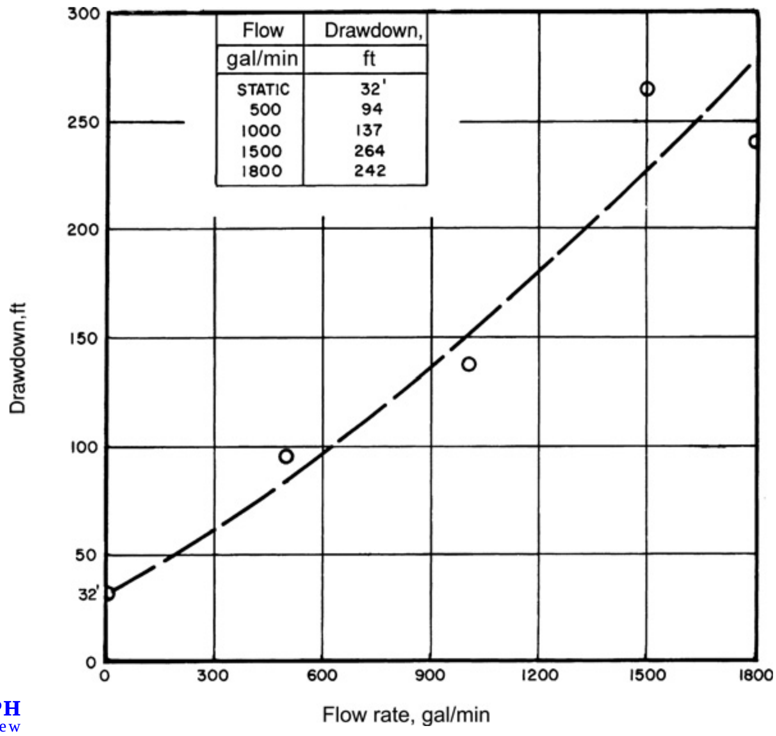
Example 18-4
Design of a Deep Well Pumping Station

Problem: Maxwell (population 1800) needs a second, reliable water source at 500 gal/min minimum. The well location selected is 2000 ft from an existing 100,000 gal storage tank with a low water level 107 ft and a high water level 127 ft above the ground, which is level between the well and the tank. Based on the results of well tests, well screens were placed between depths of 595 to 605 ft and 720 to 724 ft. The drawdown curve is shown in Figure 18-15.

The problem is to (1) plot the system H-Q curve, (2) select the pump, (3) select the driver, (4) design the pump layout at the well head, and (5) plan the chlorination facilities. The transmission main has been selected by others.

Solution: As outlined in Chapter 17, a well-documented record of all verbal communications, letters, calculations, and equipment selections should be maintained.

(1) Transmission main. Ordinarily, comparative analyses of initial cost plus energy costs over a 20-year (or longer) period would be made for two or three pipe diameters to select the optimum. The well, however, is the only water source on one side of a railroad and, because that area is growing, the owner wants the largest economically feasible transmission main, which is an 8-in. diameter pipe. The profile of the pipe is shown in Figure 18-16. Data for TDH calculations at three points are as follows:



LIVE GRAPH
Click here to view

Figure 18-15. Pump drawdown curve for well. Courtesy of Psomas and Associates and Raymond Vail and Associates.

	Flow rate, gal/min		
Item	300	500	700
Flow rate, ft^3/s	0.67	1.11	1.56
Velocity, ft/s (8-in. ID)	1.92	3.18	4.47
Velocity head, $v^2/2g$	0.057	0.157	0.310
Dynamic losses, maximum	0.51	1.27	2.37
287-ft lined steel pipe, $C = 140$[a] plus			
2000-ft PVC pipe, $C = 150$	3.03	7.79	14.51
TDH			
Maximum pumping head:			
Tank water level (above grade)	127	127	127
Well water level (below grade)	78	108	137
287-ft lined steel pipe, $C = 140$	0.5	1.3	2.4
2000-ft PVC pipe, $C = 150$	3.0	7.8	14.5
Minor losses: inlet, gate valve, check valve, 10 elbows, outlet, $K = 7.3$[b]	0.4	1.1	2.3
TDH	209	245	283
Minimum pumping head:			
Tank water level (above grade)	107	107	107
Well water level (below grade)	60	83	109
287-ft lined steel pipe, $C = 140$	0.5	1.3	2.4
2000-ft PVC pipe, $C = 150$	3.0	7.8	14.5
Minor losses: inlet, gate valve, check valve, 10 elbows, outlet, $K = 5.2$b	0.3	0.8	1.6
TDH	171	200	235

[a]Equation 3-9b.

[b]From Tables B-6 and B-7: K_{ent} (through screen) = 1; $K_{gate} = 0.1$ to 0.2; $K_{check} = 0.6$ to 2.6; $K_{elbow} = 0.25$; $K_{exit} = 1$. Equation 3-16: $h = Kv^2/2g$.

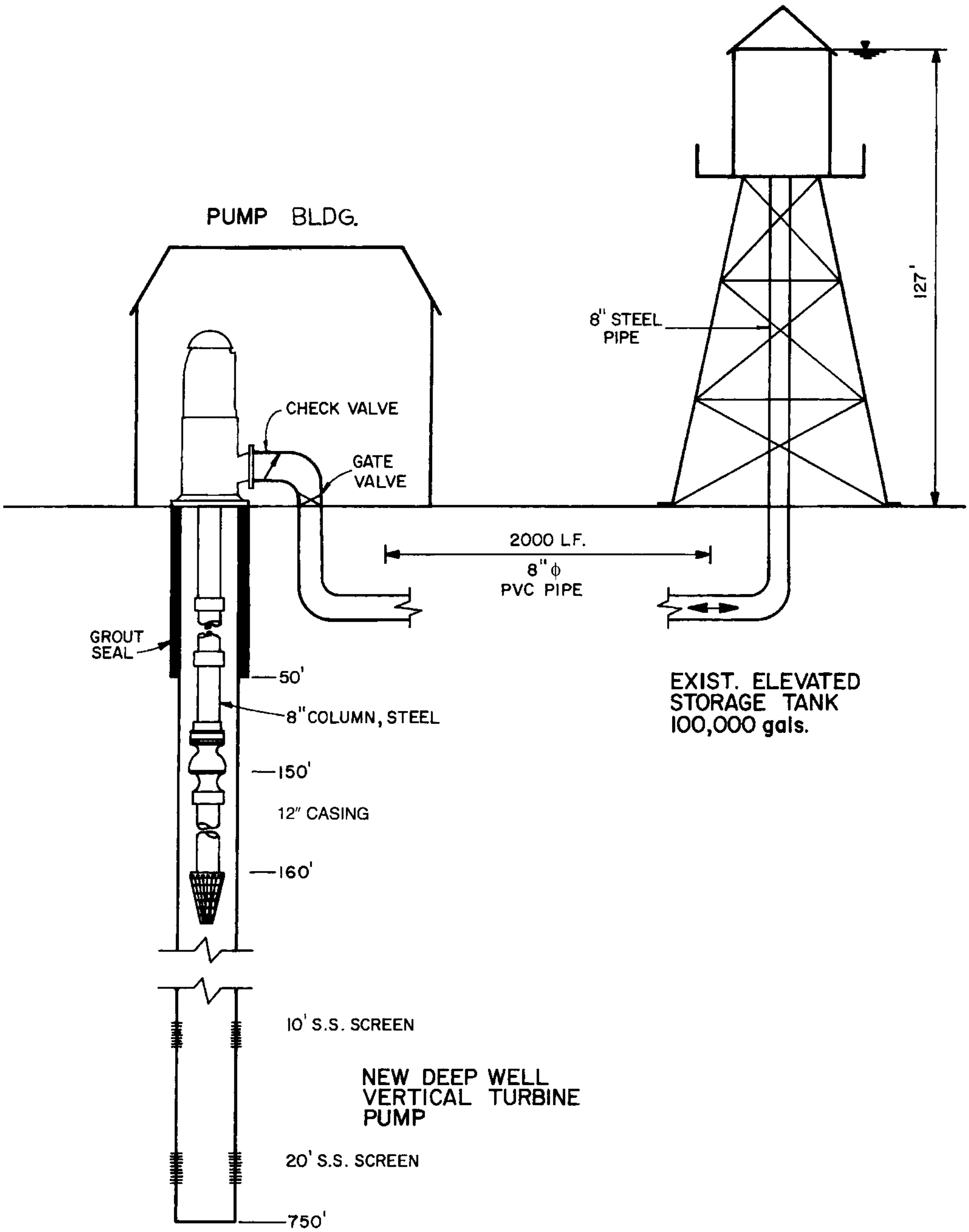
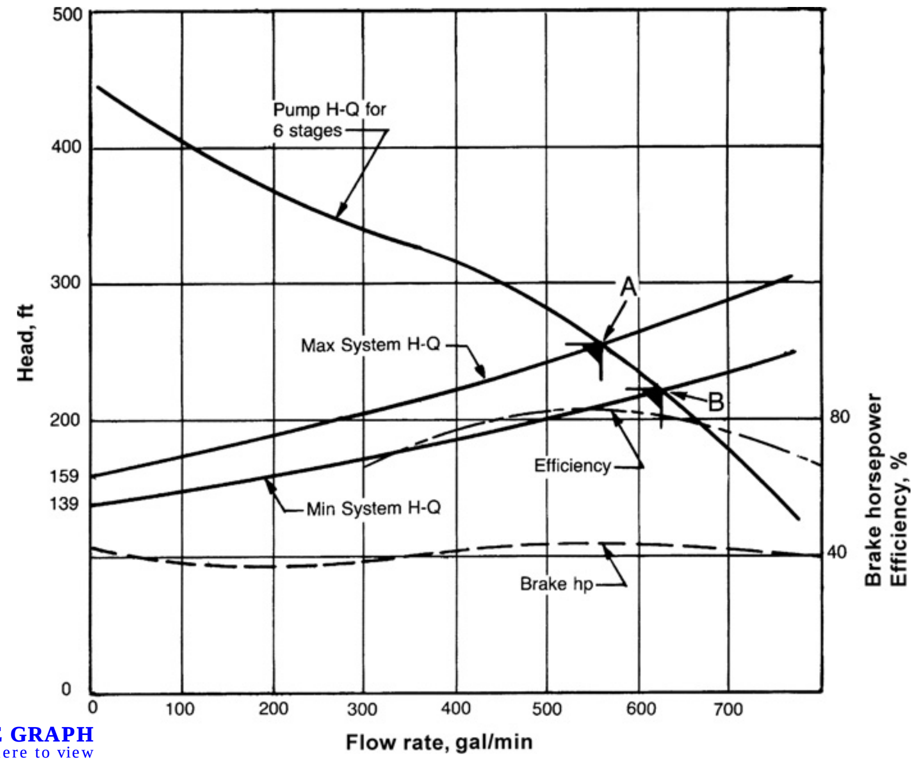


Figure 18-16. Well and pipe profile for Example 18-4. Courtesy of Psomas and Associates and Raymond Vail and Associates.

Plot the system curve. The system H-Q curve is a band as shown in Figure 18-17. Hence, the pump can operate anywhere along the pump curve between points A and B.

(2) Select pump. Consult a manufacturer's catalog for a vertical turbine pump that will discharge at least 560 gal/min at a TDH of 255 ft and 630 gal/min at a TDH of 220 ft (from the system curve in Figure 18-17). See Section 12-10 for the selection process. At six bowls, the head per stage for a candidate pump would be about 43 ft, which is reasonable. The head for a



LIVE GRAPH
Click here to view

Figure 18-17. System head-capacity and pump performance curves for Example 18-4. Courtesy of Psomas and Associates and Raymond Vail and Associates.

single stage of the turbine pump is multiplied by 6 to obtain the total pump H-Q curve, as shown in Figure 18-17.

Refinements in the calculation of total head or energy due to water density, derating for the number of stages, whether bowls are enameled, impeller material, and so on can be found in the manufacturer's catalog. The pump operates between points A (560 gal/min) and B (630 gal/min).

The NPSHR at 630 gal/min is 16 ft from the manufacturer's catalog. The NPSHA is calculated as follows:

- Lowest bowl elevation = 150 ft below grade
- Drawdown at 630 gal/min = 126 ft below grade (worst case)

- Net submergence = +24 ft
- Atmospheric pressure head = +33.9
- Vapor pressure head (60°F) = −0.6
- Entrance loss = −0.25
- Safety factor (for wells) = −5
- NPSHA = 52.1 (obviously adequate). Use 52 ft.

The safety factor usually suggested for NPSHA is 2 ft, but due to the uncertainty of drawdown for a well, larger safety factors should be used. A safety factor of 20 ft might be too little in some circumstances.

Other considerations in pump selection are allowable internal pressure in the bowls, shaft stretch, and hydraulic losses between casing and pump. Ordinarily, the manufacturer should be consulted for these concerns, but for this problem, such concerns are trivial because the pump is not deep and the head is not large.

The total pumping head is 255 ft, and about 5 ft per 100 ft of pump column should be added to allow for friction and turbulence losses.

The bowl head pressure $= 255 + (5 \times 160/100) = 263\,\text{ft} = 114\,\text{lb/in.}^2$ (also satisfactory). Shaft stretch, thrust-bearing load, and bearing life should be calculated by the manufacturer; the calculations should be certified and given to the designer for checking. Pumps from several manufacturers should be investigated and the selection made on the basis of efficiency as well as on the quality of the pump. Specifications can be written to include a penalty/reward clause for efficiency. Quality can be attained by writing specifications as explained in Section 12-4.

(3) Select the driver. As shown in Figure 18-17, the operating point depends on the drawdown, the piping losses, and the elevated tank water level, so operation could occur at any point along the pump curve between point A (560 gal/min at 225 ft) and B (630 gal/min at 220 ft). From Equation 10-6b the water power is:

$$hp = \frac{56 \times 25}{396} = 36$$
$$hp = \frac{63 \times 22}{396} = 35$$

The pump efficiency is 82%, so the brake horsepower at the pump shaft is:

$$hp = \frac{36}{0.82} = 44$$

which is confirmed by the plot of horsepower in Figure 18-17.

The motor selected should be the next higher motor size manufactured, or 50 hp. A check of the pump curve shows that a 50-hp motor can power the pump along the full range of the pump curve. Note that there are minor losses caused by shaft bearing and thrust bearing friction. Such losses should be computed by the manufacturer, certified, and transmitted to the designer for checking.

(4) Pump layout. The pump and piping layout is shown in Figure 18-18, and well details are shown in Figure 18-19. Items of special note are as follows:

- An air release valve to vent air on pump start-up. The venting is critical. It must cushion the shock of opening the check valve and starting the movement of water in the transmission main. Data for calculating the size of the orifice must be obtained from the valve manufacturer.
- A check valve to minimize water hammer (see Chapter 7). A "silent" (center-post-guided, globe-style) check valve or a swing check could be used.
- An isolation gate or butterfly valve.
- An advantage of the low velocity in the transmission main is that the simple valving system shown in Figure 18-18 was found by analysis to be adequate for water hammer control.
- A pump bearing lubrication system. Use water lubrication in most potable water installations.
- A pump packing seal or gland or stuffing box.
- A ½- to ¼-in. tap for a pressure gauge (see Figure 20-6).
- A 1-in. tap for chlorine feed addition.
- A direct-coupled, hollow-shaft motor with a top adjusting nut to adjust for shaft stretch or abrasive wear.
- Other pump features are available and important for particular installations and should be discussed in detail with the pump manufacturer. Some features include thermal protection and winding heaters.
- The pump motor is energized by pressure switches set to respond to the level of water in the elevated reservoir.

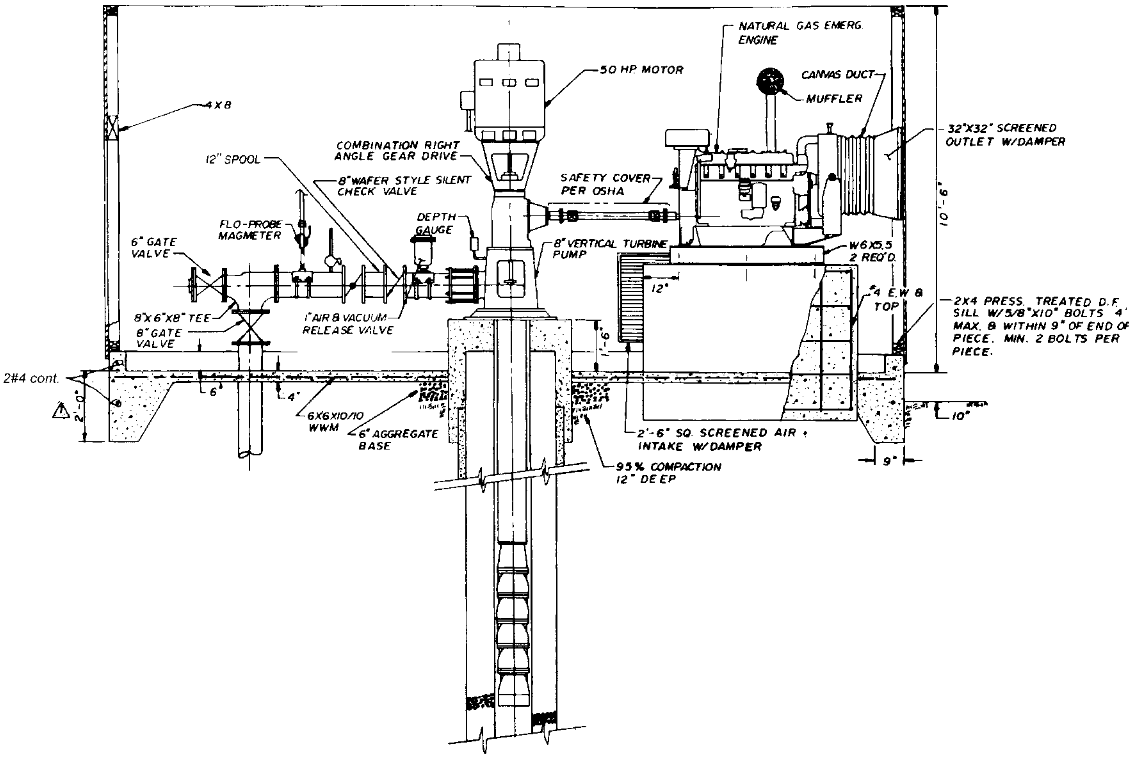


Figure 18-18. Pump and piping detail for Example 18-3. Courtesy of Psomas and Associates and Raymond Vail and Associates.

- If an extended power outage is likely, auxiliary power is required. Because the system has storage sufficient for 3 h of supply at peak day demand, a manually started engine directly connected to the pump through a right angle gear is satisfactory. If an engine is installed, use it to pump for a 1-h period at least twice per month (once per week is better).

(5) Chlorination facilities. The State Department of Health required the well water to be chlorinated. If the well were operated continuously at 560 gal/min, the chlorine gas use at 2 ppm (a relatively low dosage) would be:

$$560\,\text{gal/min} \times \frac{1 \times 10^6\,\text{gal/d}}{694\,\text{gal/min}} \times \frac{8.34\,\text{lb}}{\text{gal}} \times \frac{2\,\text{lb}}{10^6\,\text{lb}} = 13\,\text{lb/d}$$

This well, however, is one of three available and is used only part time, so the amount of chlorine used would be small. Hypochlorite is more expensive than compressed chlorine gas, but it is safer and the equipment is less expensive, so a batch hypochlorination system was installed. The system, shown schematically in Figure 18-20, consists of a 60-gal PVC batch tank and a metering pump connected to the well pump discharge pipeline. The pump controller also starts the metering pump.

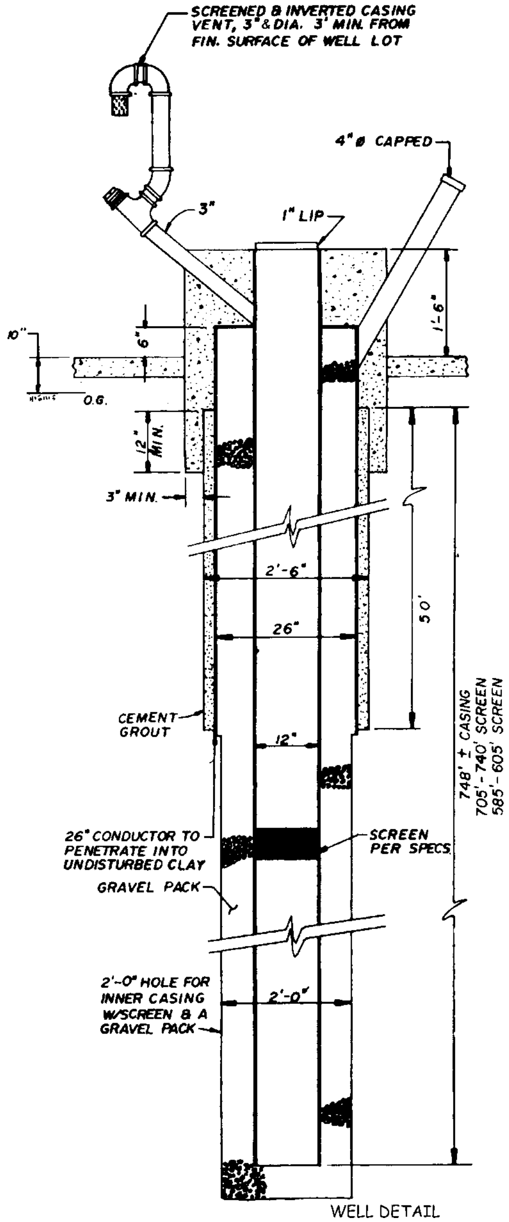


Figure 18-19. Well detail for Example 18-3. Courtesy of Psomas and Associates and Raymond Vail and Associates.

Critique of Example 18-4

The pump setting 150 ft is relatively shallow, so a lineshaft pump is proper. For deep wells with pump settings greater than 600 ft, for crooked wells, or for wells where noise must be suppressed, consider the use of submersible pumps.

The simple valving system at the well head is adequate for this particular well. Neither valve slam nor excessive surge occurs because the water velocity is very low. It is important to make a water hammer analysis. If valve slam or excessive surges can occur, use a more sophisticated valve system such as one of those discussed in Section 7-6.

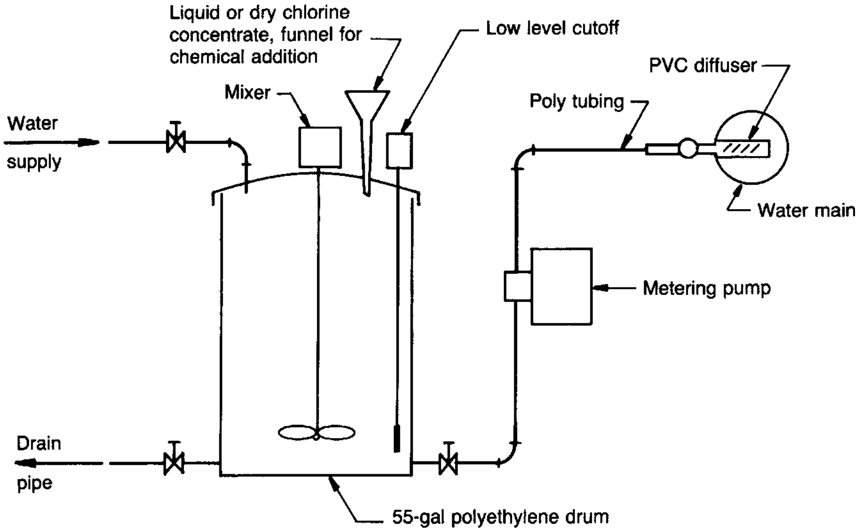


Figure 18-20. Batch hypochlorinator schematic.

Some engineers would specify a harness for the sleeve coupling at the pump discharge elbow. Others would note that if the 8-in. gate valve body is not overstressed by the shear load imposed, nothing can move and no harness is needed. The sleeve coupling is crowded in its location, and two alternatives are (1) a flange adapter, and (2) a grooved-end coupling. Some engineers would also provide an intermediate pipe support for the discharge header, but the header is only 6 ft long and can bridge that distance.

The device to the right of the Flo-Probe Magmeter® in Figure 18-18 is a sampling tap. Taps for a pressure gauge and chlorine addition are included but not shown in the figure. When the pump starts, a water-air mixture explodes out of the air release valve, so drain piping (also not shown) leads from the valve past the pump and engine.

The space between the 6-in. gate valve and the wall is only about 2 ft. More clearance (at least 30 or 36 in.) should be provided.

18-6. Booster Pumping Stations

Booster pumps can be divided into two types: in-line and distribution. In-line boosters take suction from an incoming pipeline, pressurize all the water, and discharge it into another pipeline. Distribution boosters take suction typically from storage (although sometimes from a portion of the flow in a pipeline) and maintain a given pressure (within limits) for supply in a distribution system at wide ranges of demand.

In-line Booster Pumping Stations

The potential advantages of an in-line booster pumping station include: (1) the pipeline on the suction side of the booster station can be designed for a lower pressure rating, which thereby reduces the pipeline construction costs; (2) all of the water in the system need not be pumped at maximum system pressure, so energy costs are reduced; and (3) the primary pumping station (e.g., the source pumping station at a clear well or reservoir) need not be designed for the high-pressure conditions that are necessary for only a part of the entire water system.

The potential disadvantages include: (1) additional pumping station construction cost, unless the cost of the primary pumping station can be reduced; (2) additional pumping station O&M costs; (3) increased operational complexity; (4) additional electric power substation required; (5) complicated analysis and control of system hydraulic transients; and (6) the possible need for other facilities, such as access roads and power lines.

General comments concerning the typical transmission main in-line booster pumping station, illustrated schematically in Figure 18-21, are as follows:

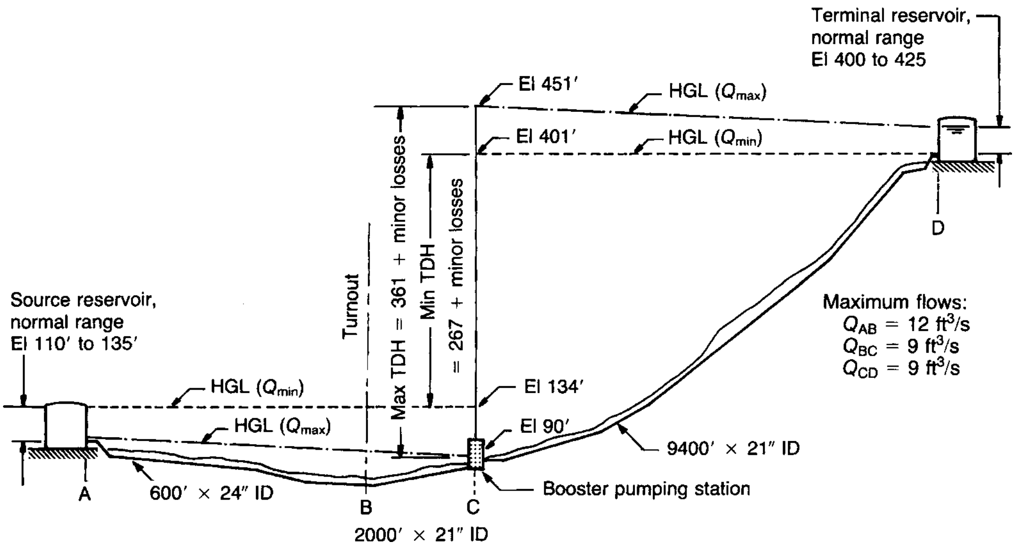


Figure 18-21. Transmission main with an in-line booster pumping station. Gravity flow out of the source reservoir. After Boyle Engineering Corp.

- Operationally, the booster pumping station could be located anywhere between point A and point C. The relationship of the minimum suction side hydraulic grade line (HGL) to the ground is the limiting factor. The critical HGL (based on the lowest reservoir water level and the peak flow rate) should exceed ground level by at least 3 to 6 m (10 to 20 ft).
- The operating pressure in pipeline segment AC is significantly reduced compared with the alternative of placing the pumping station at point A instead of at point C.
- Energy can be saved if the portion of flow delivered through the turnout at point B does not require boosting.
- The booster pumps must be capable of producing the required flow rate at the maximum TDH, and they must operate without cavitation at the minimum TDH.
- Both suction and discharge pipelines should be analyzed for transients, and facilities should be designed as necessary to prevent water hammer.

The addition of a primary pumping station (Figure 18-22) creates a system that is hydraulically and operationally more complex than the one in Figure 18-21. General comments about Figure 18-22 are as follows:

- Capital and annual O&M costs of the booster pumping station must be offset by the reduction in the costs of the primary pumping station and the pipeline segment AB due to a lower HGL.
- The HGL between points A and B depends on the pump H-Q curves selected for each station.
- The rate of flow produced at both stations must be equal at all times, so careful matching of the pumping units at the two locations is required.
- The pipeline transients must be analyzed and a surge control system must be designed.
- Provision must be made to shut down each pumping station on high or low pressure if a power outage occurs at the other station.
- Depending on pump selection, it may be necessary to allow gravity backflow from point B to point A to produce the proper back pressure for the pumping units at point A to start against.

Distribution Booster Pumping Stations

Examples of booster pumping station installations that provide pressure to a municipal water supply upper service zone are schematically shown in Figures 18-23 and 18-24.

The following comments relate to the system shown in Figure 18-23:

- The system hydraulics are controlled by the water level in the elevated reservoir.
- The pumps are started by pressure switches located at the booster station as pressure drops due to customer demand.

- Pumping in excess of demand refills the elevated reservoir. When the reservoir is full, the altitude valve closes and causes a pressure rise, and a pressure switch sequentially shuts off the booster pumps at set time intervals.
- The discharge of each booster pump is equipped with a pump-control valve or some other device to limit the transient pressures of start-up and shutdown.
- Rapid pressure fluctuations in this type of system often cause customer dissatisfaction. Service can be improved by adding a hydropneumatic tank at the booster.

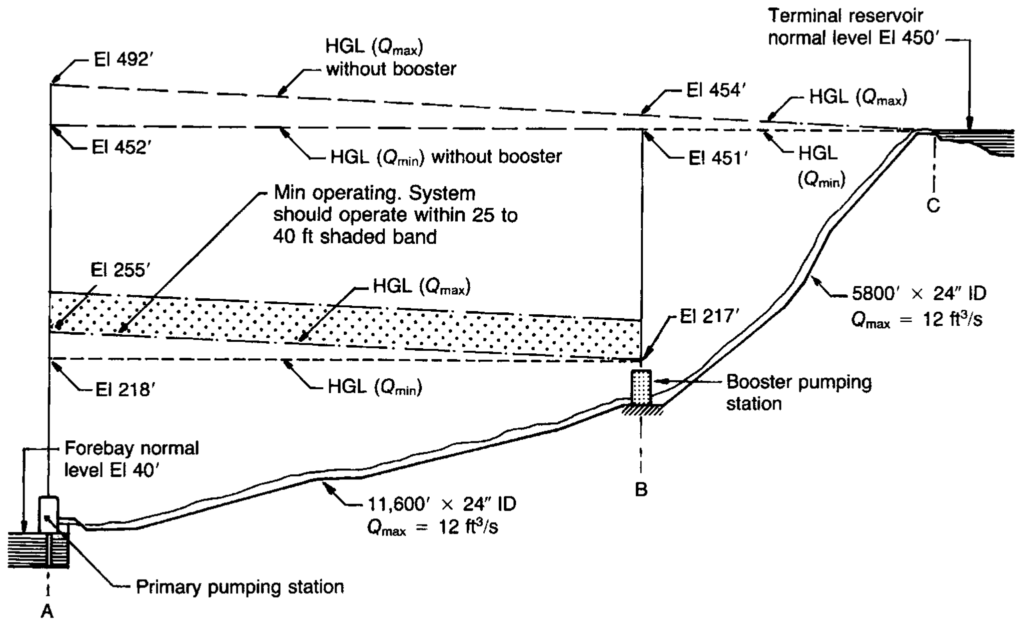


Figure 18-22. Transmission main with a primary pumping station and a booster pumping station. After Boyle Engineering Corp.

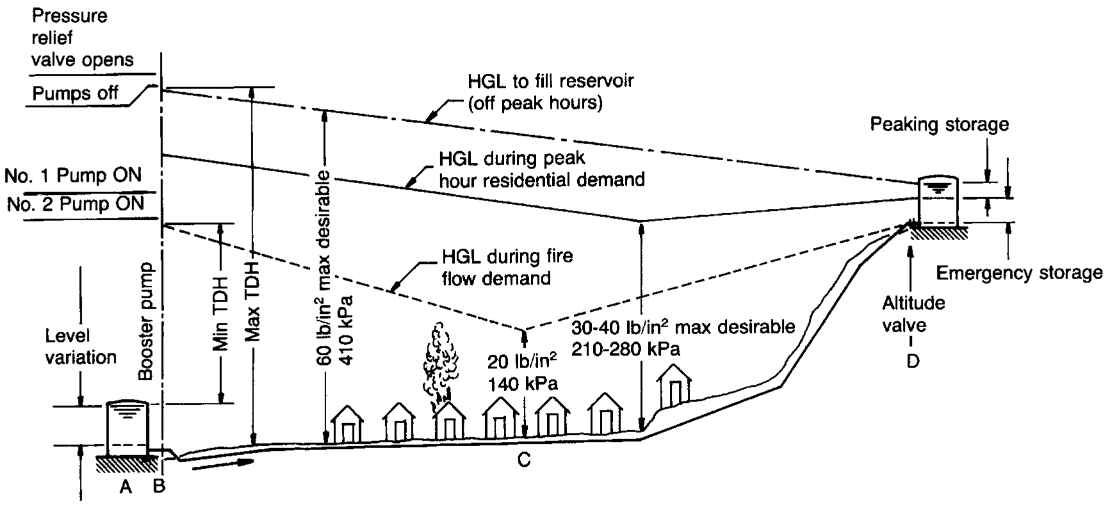


Figure 18-23. Distribution system booster with reservoir control of the hydraulic gradeline. After Boyle Engineering Corp.

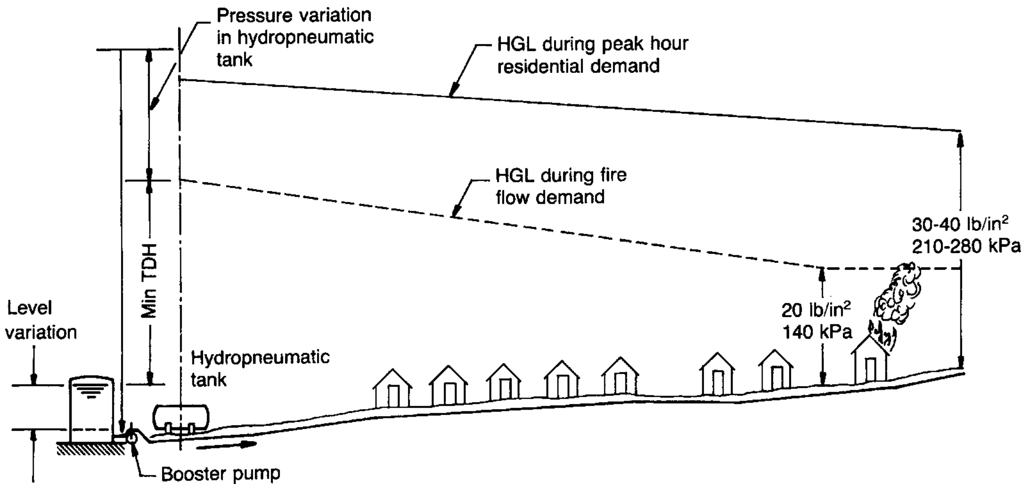


Figure 18-24. Distribution system booster without reservoir control of the hydraulic gradeline. After Boyle Engineering Corp.

Comments on the system in Figure 18-24 are as follows:

- An elevated reservoir is assumed not to be economically justified, due to a small service area and flat terrain.
- A hydropneumatic tank maintains system pressure within prescribed limits. The pumps are started and stopped based on water level and pressure in the hydropneumatic tank.

Example 18-5
Small Distribution System Booster without Reservoir

Problem: Design a booster pumping station for a small residential distribution system pressure zone serving 200 persons. The booster is to be located adjacent to and pump from an existing ground-level welded steel reservoir. It has been determined that an elevated storage reservoir cannot be justified economically. A hydropneumatic tank is to be included in the design to maintain system pressure within prescribed limits. Because the climate is mild with temperatures rarely below freezing, an outdoor installation is acceptable.

Solution: Note that differences in results between SI units and U.S. customary units are due to the number of significant digits used.

Data and assumptions.

Item	SI Units	U.S. Customary Units
Population	200	200
Water consumption	946 L/d · cap	250 gal/d · cap
Fire flow rate		
(minimum rate per local code)	31.5 L/s	500 gal/min
Demand flow rates		
Average daily	200 × 946 L/d = 2.2 L/s	200 × 250 gal/d = 35 gal/min
Maximum daily	2.5 × avg daily = 5.5 L/s	2.5 × avg daily = 87 gal/min
Peak hour	4.0 × avg daily = 8.8 L/s	4.0 × avg daily = 139 gal/min
Suction reservoir operating elevations		
Maximum water surface	103.9 m	341 ft
Minimum water surface	100.0 m	328 ft

Item	**SI Units**	**U.S. Customary Units**
Ground-level elevations		
At booster station	98.1 m	322 ft
At system high point	104.9 m	344 ft
At system low point	89.9 m	295 ft
System pressure requirements		
Maximum pressure	517 kPa	75 lb/in.^{2}
Peak hourly flow	207 kPa	30 lb/in.^{2}
Max daily flow plus fire flow	138 kPa	20 lb/in.^{2}

Define the booster peak design flow rate. Because the distribution system is without storage, a reasonable peak design flow rate for the booster station is the sum of the maximum daily average flow rate and the required fire flow rate.

(5.5 + 31.5) L/s = 37 L/s (87 + 500) gal/min = 587 gal/min

Define the allowable pressure variation in the hydropneumatic tank. The maximum operating pressure in the hydropneumatic tank should be the pressure that, during static conditions, produces the maximum allowable system pressure at the lowest customer service in the pressure zone. Estimate the water surface in the hydropneumatic tank to be 1.5 m (5 ft) above ground level.

The maximum operating pressure at the hydropneumatic tank is:

$$(517\,\text{kPa} + 89.9\,\text{m}/0.102\,\text{m/kPa}) - [(98.1\,\text{m} + 1.5\,\text{m})/0.102\,\text{m/kPa}] = 1398\,\text{kPa} - 976\,\text{kPa} = 422\,\text{kPa}$$

$$[75\ \text{lb/in.}^2 + 295\,\text{ft}/2.31\,\text{ft/(lb/in.}^2)] - [(322\ \text{ft} + 5\,\text{ft})/2.31\,\text{ft/(lb/in.}^2)] = 203 - 142 = 61\,\text{lb/in.}^2$$

The minimum operating pressure in the tank must be at least as great as the larger of the following:

- The pressure required to deliver the peak hourly flow rate with at least 207 kPa (30 lb/in.^{2}) throughout the system, or
- The pressure required to deliver the maximum day average flow plus the required fire flow at a pressure of 138 kPa (20 lb/in.^{2}) or more to the most critical hydrant location.

These determinations require a system network analysis, which is typically accomplished using computer programming. For a small system, however, it can readily be done by hand using the Hardy–Cross method. For this problem, the following assumptions have been made:

- The latter criterion governs (i.e., the delivery of maximum day average flow plus fire flow).
- The critical hydrant is located at the system high point.
- The drop in system pressure from the hydropneumatic tank to the critical hydrant due to pipeline friction and other system losses is 96 kPa (14 lb/in.^{2}).

The minimum operating pressure at the hydropneumatic tank is:

$$(138\,\text{kPa} + 104.9\,\text{m}/0.102\,\text{m/kPa}) + 96\,\text{kPa} - [(98.1\,\text{m} + 1.5\ \text{m})/0.102\,\text{m/kPa}] = 286\,\text{kPa}$$

$$[(20\,\text{lb/in}^2 + 344\,\text{ft}/2.31\,\text{ft/(lb/in.}^2)] + 14\,\text{lb/in}^2 - [(322\,\text{ft} + 5\,\text{ft})/2.31\,\text{ft/(lb/in.}^2)] = 41\,\text{lb/in.}^2$$

The allowable pressure variation in the hydropneumatic tank can thus be between 286 kPa (41 lb/in.^{2}) and 422 kPa (61 lb/in.^{2}), or a 136-kPa (20-lb/in.^{2}) differential. Differential pressures from 136 kPa (20 lb/in.^{2}) to 207 kPa(30 lb/in.^{2}) are considered normal for hydropneumatic systems.

Pumping unit selection. The following subjects must be addressed relative to pumping unit selection:

- Horizontal split-case pump versus vertical turbine pump
- Electric motor versus natural gas (or diesel) engine driver
- The number and capacity of pumping units.

Considerations that influence the choice between horizontal and vertical pumps include the suitability of available pump curves to meet the full range of system head-capacity requirements, pump speed, pump efficiency, construction costs, space requirements, available suction pressure, and owner preference. Pump curves from several manufacturers of both vertical turbine and horizontal split-case pumps were compared with the system requirements. The primary problem in pump selection was finding a pump that would operate at both the upper and lower head conditions with reasonable efficiency. The pump curve that best met the conditions was that of a five-stage, vertical turbine pump operating at 1760 rev/min. A vertical turbine barrel pump has another advantage compared with a split-case pump—it can operate safely with a lower suction hydraulic gradeline.

Factors influencing the choice of an electric motor or an engine drive include equipment costs, maintenance costs, space requirements, the availability and dependability of electric power, the availability of natural gas, the costs of these alternative energy sources, and, again, owner preference (see Chapters 13, 14, and 25). These factors vary from area to area and from time to time. For this problem, it is assumed that an economic evaluation of initial installation costs plus present worth of long-term energy costs favors electric motors. A power outage in excess of a few minutes, however, would mean *no water* because the system has essentially no storage. (Consideration will be given to minimal emergency storage in the hydropneumatic tank later in this example problem.) It is thus vital to contact the utility supplying electric power to obtain an outage history. If outages have been significant, then one of the following options should be selected:

- Natural gas (or diesel) engines as main drivers
- Electric motors with direct-connected natural gas or diesel engines as back-ups
- Electric motors with a natural-gas-powered or a diesel-engine-powered standby generator.

For this problem, the outage history is assumed to be very good with no outages longer than 5 min experienced within the last two years. Thus, electric motors are to be used, but only after careful review with and approval by the owner of the proposed booster and distribution system.

The number and capacity of pumping units are determined on the following basis: (1) the pumps normally have to produce flows only up to the peak hour demand flow rate of 8.8 L/s (139 gal/min) and (2) the pumps must also be capable of occasionally producing the peak design flow rate of 37 L/s (587 gal/min). Because these flow rates are likely to occur for durations of an hour or more, they must be supplied by the pumps because hydropneumatic tank storage is inadequate for such periods. The pumps should also be selected so that, when new, they include a "wear allowance" of 5 to 10%. In effect, the pumps should initially produce excess flow at the design head such that after five years or so of operation and the resultant loss of capacity due to wear, the pumps will still produce the design flow rate.

Standby (back-up) units for each size of pump are desirable for domestic water supply systems and probably mandatory where the system does not include storage. For a system of this size, the following tentative combinations should be considered:

- One primary pumping unit plus an identical standby unit (a total of two units)
- Two identical primary pumping units plus one identical standby unit (a total of three units).

The selection of the pumping unit combination to be utilized in the booster station design should be based on the following considerations:

- Costs of pumping units, electrical switchgear, piping, and valving—which favor fewer, larger units
- Operational flexibility and system reliability—which favor more, smaller units

- System surge control—which favors more, smaller units, but the fundamental problem is always power failure
- Energy cost—which favors more, smaller units if a "demand charge" is a significant element of the power rate schedule
- Hydropneumatic tank size and cost—which favor more, smaller units
- Space requirement—which favors fewer, larger units.

For this example, the preferred pump combination is found to be three identical pumping units—two duty pumps and one standby unit.

Develop system head-capacity and pump characteristic curves. It is necessary to establish the full range of pumping conditions to select pumping units properly. As shown below, the total dynamic head on the pumps varies. Assume minor losses between the reservoir and the hydropneumatic tank at 1.8 m (6 ft). (The minor loss estimate should be verified during the design of the booster station piping.)

The maximum hydropneumatic tank pressure and minimum suction reservoir level is $\mathrm{TDH} = \mathrm{HGL_{tank}} - \mathrm{HGL_{res}} + \text{minor losses}$.

$$\begin{aligned}\mathrm{TDH} &= (98.1\,\mathrm{m} + 1.5\,\mathrm{m} + 422\ \mathrm{kPa} \\ &\quad \times 0.102\,\mathrm{m/kPa}) - 100.0\,\mathrm{m} + 1.8\,\mathrm{m} \\ &= 44.4\,\mathrm{m}\end{aligned} \qquad \begin{aligned}\mathrm{TDH} &= [322\,\mathrm{ft} + 5\,\mathrm{ft} + 61\,\mathrm{lb/in.}^2 \\ &\quad \times 2.31\,\mathrm{ft/(lb/in.}^2)] - 328\,\mathrm{ft} + 6\,\mathrm{ft} \\ &= 146\,\mathrm{ft}\end{aligned}$$

This maximum TDH is plotted as a horizontal line in Figure 18-25.

The minimum hydropneumatic tank pressure and maximum suction reservoir level is $\mathrm{TDH} = \mathrm{HGL_{tank}} - \mathrm{HGL_{res}} + \text{minor losses}$.

$$\begin{aligned}\mathrm{TDH} &= (98.1\,\mathrm{m} + 1.5\,\mathrm{m} + 286\,\mathrm{kPa} \\ &\quad \times 0.102\ \mathrm{m/kPa}) - 103.9\,\mathrm{m} + 1.8\,\mathrm{m} \\ &= 26.7\,\mathrm{m}\end{aligned} \qquad \begin{aligned}\mathrm{TDH} &= [322\,\mathrm{ft} + 5\,\mathrm{ft} + 41\,\mathrm{lb/in.}^2 \\ &\quad \times 2.31\,\mathrm{ft/(lb/in.}^2)] - 341 + 6\,\mathrm{ft} \\ &= 87\,\mathrm{ft}\end{aligned}$$

This minimum TDH is shown as a horizontal line in Figure 18-25.

The minimum hydropneumatic tank pressure and minimum suction reservoir level is critical because the suction pressure is minimum. $\mathrm{TDH} = \mathrm{HGL_{tank}} - \mathrm{HGL_{res}} + \text{minor losses}$.

$$\begin{aligned}\mathrm{TDH} &= (98.1\,\mathrm{m} + 1.5\,\mathrm{m} + 286\,\mathrm{kPa} \\ &\quad \times 0.102\,\mathrm{m/kPa}) - 100.0\ \mathrm{m} + 1.8\,\mathrm{m} \\ &= 30.6\,\mathrm{m}\end{aligned} \qquad \begin{aligned}\mathrm{TDH} &= [322\,\mathrm{ft} + 5\,\mathrm{ft} + 41\,\mathrm{lb/in.}^2 \\ &\quad \times 2.31\,\mathrm{ft/(lb/in.}^2)] - 328\,\mathrm{ft} + 6\,\mathrm{ft} \\ &= 87\,\mathrm{ft}\end{aligned}$$

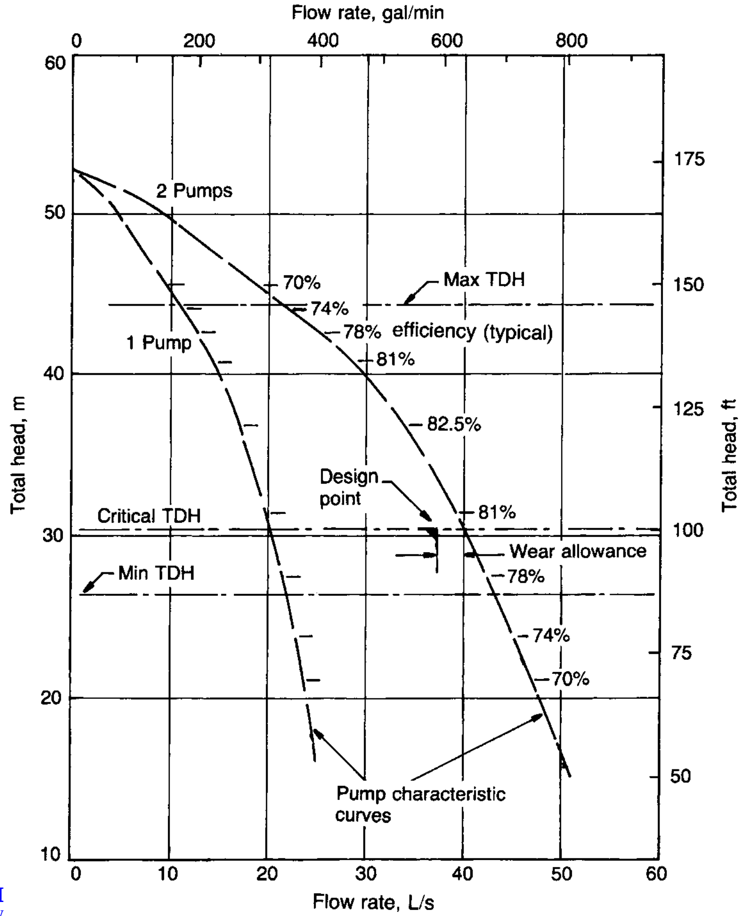
This critical TDH (30.6 m or 100 ft) is also shown as a horizontal line in Figure 18-25. The pumps must be able to discharge the peak design flow rate at this critical TDH, so the "design point" for the system is 37 L/s at 30.6 m (587 gal/min at 100 ft).

The next step is to find a pump curve that satisfies the following requirements when two such pumps are operating:

- Operate with acceptable efficiency (70% or better) at both maximum and minimum system heads.
- Operate without cavitation at minimum system head. Thus, NPSHA must equal or exceed the NPSHR as shown on the manufacturer's pump curves.
- Operate at the "design point" increased by a flow rate of 5 to 10% for wear allowance.

Available pump characteristic curves are presented electronically or graphically in pump manufacturers' catalogs, and the manufacturers' sales representatives should be consulted to assist designers in the selection of the best-suited pumps. A sample pump characteristic curve that satisfies the design requirements is plotted in Figure 18-25 for one pump and for two pumps operating in parallel.

The electric motor must be able to drive the pump at any combination of head and discharge from the maximum to the minimum TDH. The maximum required power can be found by trial, but pump manufacturers' catalogs usually show the point where the power demand peaks. For the pumping unit selected, the peak occurs at a flow rate of 18.3 L/s (290 gal/min) and a TDH of 34.1 m (112 ft) at a pump efficiency of 82%. From a combination of Equations 10-6 and 10-7, the motor output power is:



LIVE GRAPH
Click here to view

Figure 18-25. Booster pump head-capacity curves. The pump characteristic curves are based on identical pumps selected from the manufacturer's data: vertical turbine, five-stage, 1760 rev/min, 185-m (7304-in.) bowls. After Boyle Engineering Corp.

$$P = \frac{qH}{102\,\mathrm{E_P}} = \frac{18.3\,\mathrm{L/s} \times 34.1\,\mathrm{m}}{102 \times 0.82} = 7.46\,\mathrm{kW} \qquad P = \frac{qH}{3960\,\mathrm{E_p}} = \frac{290\ \mathrm{gal/min} \times 112\,\mathrm{ft}}{3960 \times 0.82} = 10.0\,\mathrm{hp}$$

Determine the required hydropneumatic tank capacity. The operation of a hydropneumatic tank is based on the following relationship (if constant temperature is assumed):

$$P_1 V_1 = P_2 V_2$$

where P is the absolute pressure, V is the volume of air, and the subscripts 1 and 2 indicate initial and final, respectively.

A hydropneumatic tank is typically designed to be at least 10% filled by water at the minimum station pressure. The tank volume is normally selected to limit pump cycling to four to six cycles per hour. For this problem, assume a minimum volume of 10% and a maximum of five cycles per hour.

Select the following pump operational criteria:

- First pump on when tank pressure falls to 314 kPa (45 lb/in.^{2}) or 32.0 m (104 ft) WC
- Second pump on when tank pressure falls to 286 kPa (41 lb/in.^{2}) or 29.2 m (95 ft) WC
- Second pump off when tank pressure rises to 394 kPa (57 lb/in.^{2}) or 40.2 m (132 ft) WC
- First pump off when tank pressure rises to 422 kPa (61 lb/in.^{2}) or 43.0 m (141 ft) WC.

Both one- and two-pump operations need to be analyzed to determine which governs the size of the hydropneumatic tank. The minimum hydropneumatic tank volume for the one-pump operation is found as follows:

Determine the percentage of tank water volume used in the one-pump operation, that is, the pressure change from 314 kPa (45 lb/in.^{2}) gauge to 422 kPa (61 lb/in.^{2}) gauge. The tank water volume at minimum pressure 286 kPa (41 lb/in.^{2}) gauge was previously assumed at 10%. Atmospheric pressure (see Table A-6 or A-7) is 101 kPa (14.7 lb/in.^{2}) at the site. From Equation 18-2,

$$(286 + 101)\ \text{kPa} \times (100 - 10) = (314 + 101)\ \text{kPa} \times (100 - \%\ \text{water})$$
At 314 kPa, water = 16.1%

$$(41 + 14.7)\ \text{lb/in.}^2 \times (100 - 10) = (45 + 14.7)\ \text{lb/in.}^2 \times (100 - \%\ \text{water})$$
At 45 lb/in.^{2}, water = 16.0%

$$(286 + 101)\ \text{kPa} \times (100 - 10) = (422 + 101)\ \text{kPa} \times (100 - \%\ \text{water})$$
At 422 kPa, water = 33.4%
Volume used = 17.3%

$$(41 + 14.7)\ \text{lb/in.}^2 \times (100 - 10) = (61 + 14.7)\ \text{lb/in.}^2 \times (100 - \%\ \text{water})$$
At 61 lb/in.^{2}, water = 33.8%
Volume used = 17.8%

Determine the average capacity of the single-pump operation varying between its on and off points. Assume the suction reservoir is at its maximum operating level [water surface elevation = 103.9 m (341 ft)], because that results in increased pump discharge (reduced TDH) and is therefore a critical case for hydropneumatic tank sizing. The hydropneumatic tank pressure for the one-pump operation varies from 314 kPa (45 lb/in.^{2}) to 422 kPa (61 lb/in.^{2}). The average tank pressure is 368 kPa (53 lb/in.^{2}). The average one-pump TDH is calculated as follows (see Figure 18-26):

$$\text{TDH} = 98.1\,\text{m} + 1.5\,\text{m} + (368\,\text{kPa} \times 0.102\,\text{m/kPa}) - 103.9\,\text{m} + 1.8\,\text{m} = 35.0\,\text{m}$$

$$\text{TDH} = 322\,\text{ft} + 5\,\text{ft} + [53\,\text{lb/in.}^2 \times 2.31\,\text{ft/(lb/in.}^2)] - 341\,\text{ft} + 6\,\text{ft} = 115\,\text{ft}$$

From the one-pump curve (Figure 18-25), the pump capacity (new) for a TDH of 35.0 m (115 ft) is 18.0 L/s (286 gal/min).

Determine the hydropneumatic tank minimum volume using the following relationship: minimum volume = ½ pump capacity × ½ cycle time/portion of tank used (expressed as decimal):

$$\text{Minimum volume} = \tfrac{1}{2} \times 18\,\text{L/s} \times \tfrac{1}{2} \times 720\,\text{s}/0.173 = 18{,}700\,\text{L}$$

$$\text{Minimum volume} = \tfrac{1}{2} \times 286\,\text{gal/min} \times \tfrac{1}{2} \times 12\,\text{min}/0.178 = 4820\,\text{gal}$$

Determine the percentage of tank water volume used in the two-pump operation:

$$(286 + 101)\ \text{kPa} \times (100 - 10) = (394 + 101)\ \text{kPa} \times (100 - \%\ \text{water})$$
Percentage water at 394 kPa = 29.6%
Volume used = 19.6%

$$(41 + 14.7)\ \text{lb/in.}^2 \times (100 - 10) = (57 + 14.7)\ \text{lb/in.}^2 \times (100 - \%\ \text{water})$$
Percentage water at 57 lb/in.^{2} = 30.1%
Volume used = 20.1%

Determine the average capacity of the second pump operating between its on and off points with the first pump operating continuously. Again assume the suction reservoir is at its maximum operating level [water surface elevation = 103.9 m (341 ft)]. The hydropneumatic

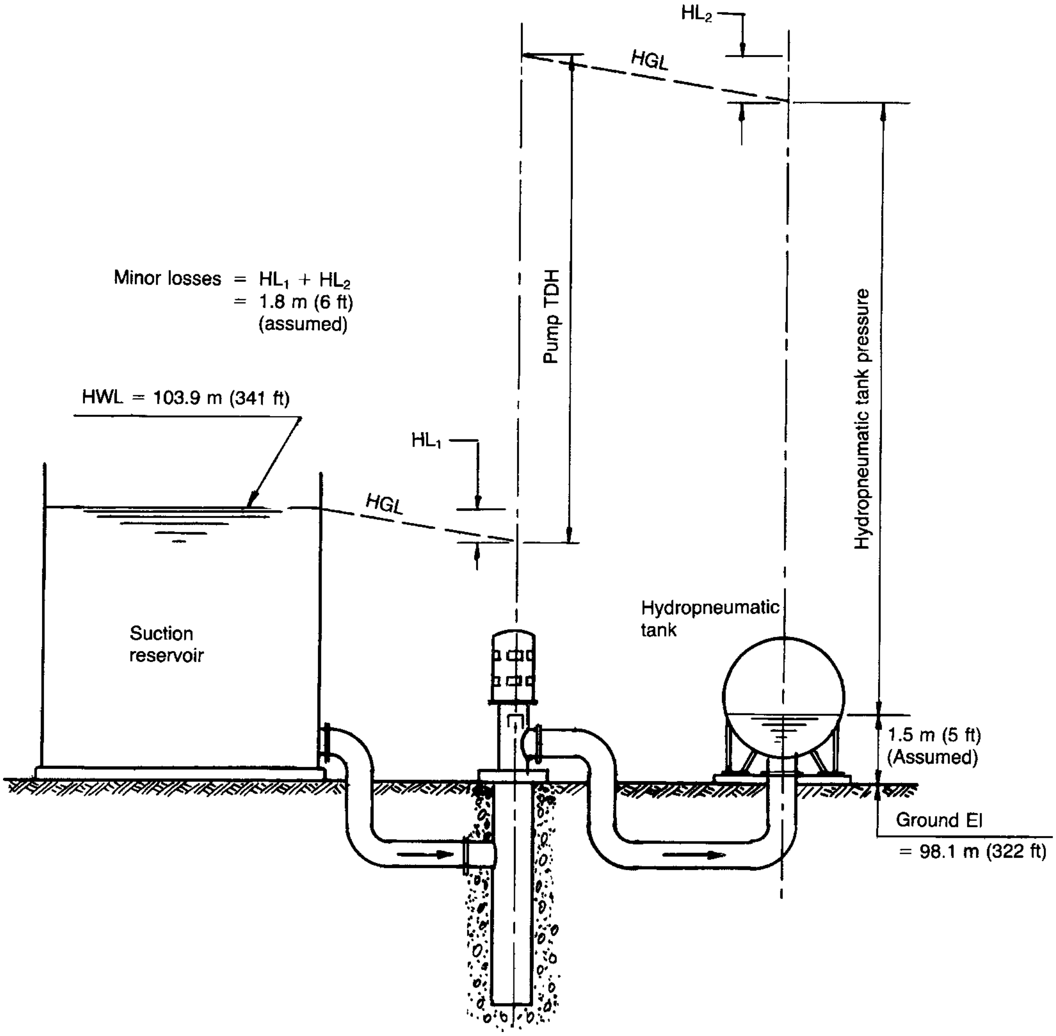


Figure 18-26. Booster pump hydraulic schematic. After Boyle Engineering Corp.

tank pressure for the second pump operation varies from 286 kPa (41 lb/in.^{2}) to 394 kPa (49 lb/in.^{2}). The average second pump TDH is calculated as follows (see Figure 18-26):

$$\begin{aligned}\text{TDH} &= 98.1\,\text{m} + 1.5\,\text{m} + (340\,\text{kPa} \\ &\quad \times 0.102\,\text{m/kPa}) - 103.9\,\text{m} \\ &\quad + 1.8\,\text{m} = 32.2\,\text{m}\end{aligned}$$

$$\begin{aligned}\text{TDH} &= 322\,\text{ft} + 5\,\text{ft} + [49\,\text{lb/in.}^2 \\ &\quad \times 2.31\,\text{ft/(lb/in.}^2)] - 341\,\text{ft} \\ &\quad + 6\,\text{ft} = 105\,\text{ft}\end{aligned}$$

From the one-pump curve (Figure 18-25), the capacity of the second pump for a TDH of 32.2 m (105 ft) is 19.2 L/s (304 gal/min).

Determine the minimum hydropneumatic tank volume for the operation of the second pump:

$$\begin{aligned}\text{Minimum volume} &= {}^1\!/_2 \times 19.2\,\text{L/s} \\ &\quad \times {}^1\!/_2 \times 720\,\text{s}/0.196 \\ &= 17{,}600\,\text{L}\end{aligned}$$

$$\begin{aligned}\text{Minimum volume} &= {}^1\!/_2 \times 304\,\text{gal/min} \\ &\quad \times {}^1\!/_2 \times 12\,\text{min}/0.201 \\ &= 4540\,\text{gal}\end{aligned}$$

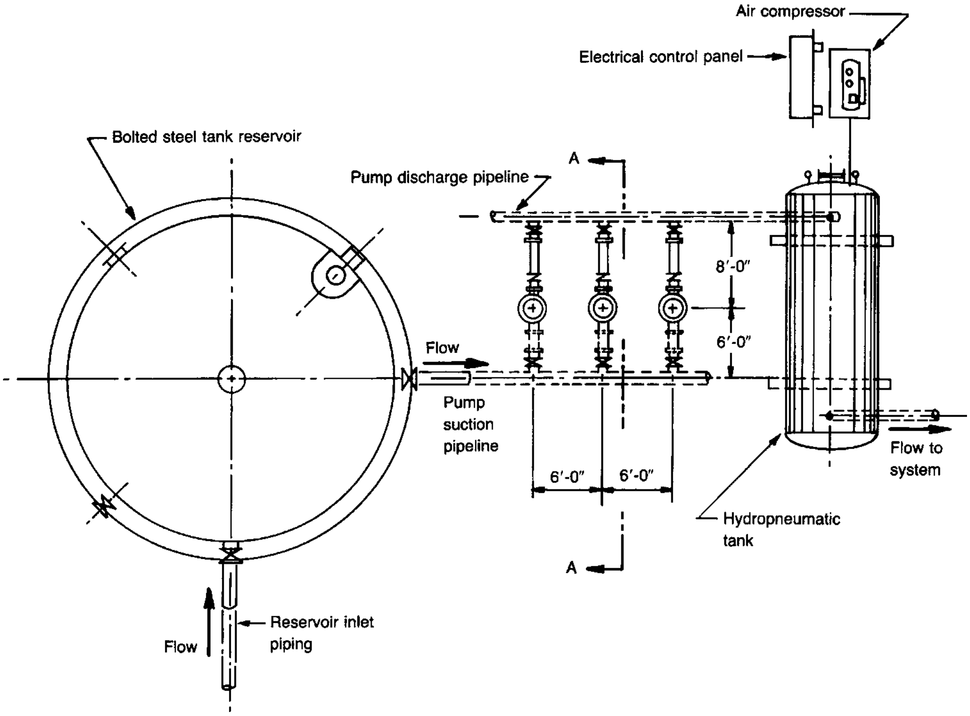


Figure 18-27. Booster pumping station plan. After Boyle Engineering Corp.

Thus, the minimum hydropneumatic tank volume is 18,700 L (4820 gal) as determined by single-pump operation.

Because the system is vulnerable to power outages, reliability relative to short-duration power outages and for average day system demands could be improved by increasing the tank water volume at minimum pressure to an amount greater than 10% (e.g., 30%). Determine the minimum tank volume based on single-pump operation.

$$(286 + 101)\ \text{kPa} \times (100 - 30) = (314 + 101)\ \text{kPa} \times (100 - \%\ \text{water})$$

At 314 kPa, water = 34.7%

$$(41 + 14.7)\ \text{lb/in.}^2 \times (100 - 30) = (45 + 14.7)\ \text{lb/in.}^2 \times (100 - \%\ \text{water})$$

At 45 lb/in.^{2}, water = 34.7%

$$(286 + 101)\ \text{kPa} \times (100 - 30) = (422 + 101)\ \text{kPa} \times (100 - \%\ \text{water})$$

At 422 kPa, water = 48.2%
Volume used = 13.5%

$$(41 + 14.7)\ \text{lb/in.}^2 \times (100 - 30) = (61 + 14.7)\ \text{lb/in.}^2 \times (100 - \%\ \text{water})$$

At 61 lb/in.^{2}, water = 48.5%
Volume used = 13.8%

$$\text{Minimum volume} = {}^1\!/_2 \times 18\ \text{L/s} \times {}^1\!/_2 \times 720\ \text{s}/0.135 = 24{,}000\ \text{L}$$

$$\text{Minimum volume} = {}^1\!/_2 \times 286\ \text{gal/min} \times {}^1\!/_2 \times 12\ \text{min}/0.138 = 6220\ \text{gal}$$

Thus, the hydropneumatic tank would be about 29% larger than the minimum required. It would, however, provide additional reserve capacity at slightly reduced pressure in the event of power outage. This reserve capacity would amount to the volume between 30 and 10% water, or 4800 L (1244 gal)—equivalent to 35 min reserve capacity at average daily flow rate. This additional investment in tank size would probably avoid an occasional depressurization of the system and time-consuming repressurization.

System layout. A sample layout of the required facilities is illustrated in Figures 18-27 and 18-28.

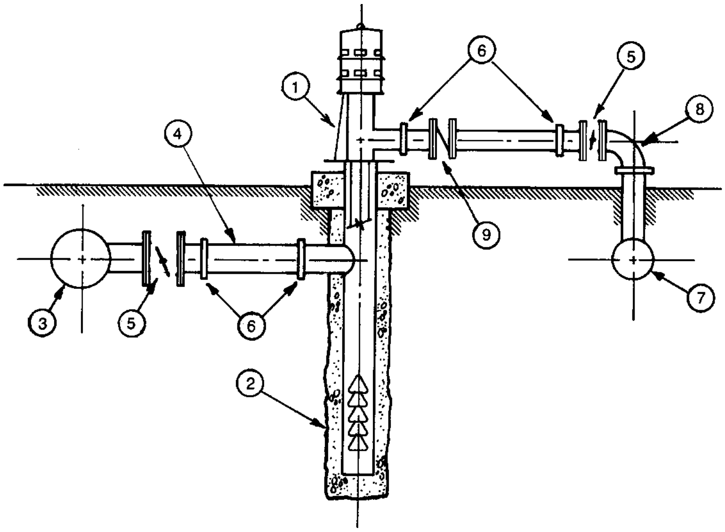


Figure 18-28. Section A-A from Figure 18-27. 1, Vertical turbine pumping unit; 2, pump "can" enclosed in concrete; 3, suction manifold pipe; 4, suction header; 5, butterfly valve for isolation; 6, Victaulic® couplings; 7, discharge manifold pipe; 8, discharge header; 9, check valve. Small appurtenances (such as air release valves, pressure switches, valve supports, motor connection boxes, and pressure gauges) are not shown. After Boyle Engineering Corp.

Packaged Booster Pumping Stations

At least one manufacturer (see Section 11-10, Subsection "Multistage Centrifugal Pumps") markets packaged booster pumps with built-in variable speed controls designed to satisfy demands for very small to rather large flows for houses, high-rise buildings, or subdivisions at little change of pressure. Some (but not all) of the foregoing calculations can be avoided because (1) integrated diaphragm tanks are sized correctly for the pumps and the application, and (2) the speed controls ramp the pump speed up or down automatically as more pumps are switched on and off.

18-7. Retrofitting Large Pump Basins

Large pump basins (and sometimes small ones as well) are sometimes afflicted with cross currents that cause rotation in pump intakes severe enough to prevent pumps from reaching their required design capacity. New rectangular basins with large pumps must have long partition walls (per ANSI/HI 9.8–1998, Figure 9.8.1) to guide water at uniform velocities into the pump intakes to prevent rotation, and such basins must therefore be very large. The submerged barrel intake has been successfully used for many years without partition walls (1) for water pumping stations, (2) for both secondary and primary wastewater treatment plant effluent, and (3) particularly for retrofitting existing sumps where the pumped fluid is devoid of large solids. The advantages are that swirling and vortices are almost entirely suppressed and the velocity distribution in the throat of the suction bell is well within ANSI/HI 9.8 recommendations even when there are strong currents above the can. The reason is seen in Figure 18–29. Submerged can intakes might also be economical for a new basin if the size and complexity of the basin could thereby be sufficiently reduced. See Section 12–3 for more discussion.

The design of submerged barrel intakes is described in Example 18–6.

Example 18–6
Dublin–San Ramon Effluent Transport Station

The outstanding feature is the use of barrel (can) intakes for pumping secondary effluent at this wastewater treatment plant.

Water flowing radially toward the pump column above the outer barrel turns downward into the annular area between the pump column and the outer can and flows downward a short distance that allows pressure to be equalized circumferentially around the pump column. When the water enters the annular space between inner and outer barrels, the vanes, which extend to the cone under the pump suction bell, stop any tendency for rotation. Model tests of these 17,400 gal/min pumps revealed a maximum swirl of 0.5 degrees, a maximum velocity variation of 7%, and a maximum turbulence (time-varying fluctuations of velocity at a point) of 8.4%. These results are superb.

Design is simple. Submerge the outer barrel sufficiently to limit the average radial (horizontal) velocity of water (flowing toward the pump column) above the rim of the barrel to 0.3 m/s (1 ft/s). Keep the area of the flow pathway either prismatic or diminishing all the way from the top of the outer can to produce either constant or increasing velocity toward the bell. Do not allow velocities to decrease at any section along the streamlines. In this respect, the design of Figure 18–29 could be improved slightly by making the fillet and cone somewhat smaller (as shown by the dashed lines) and by lowering the pump bell to a floor clearance of D/4 to prevent the area of the flow pathway from expanding under the rim of the suction bell. This modification keeps the area of the flow pathway at $\pi D^2/4$ as the water flows between the inner and outer cylinders all the way into the mouth of the bell. The length of the vanes should be at least as great as the spacing between them. It is physically impossible to extend the vanes to the apex of the cone. Terminate them where necessary to prevent overcrowding.

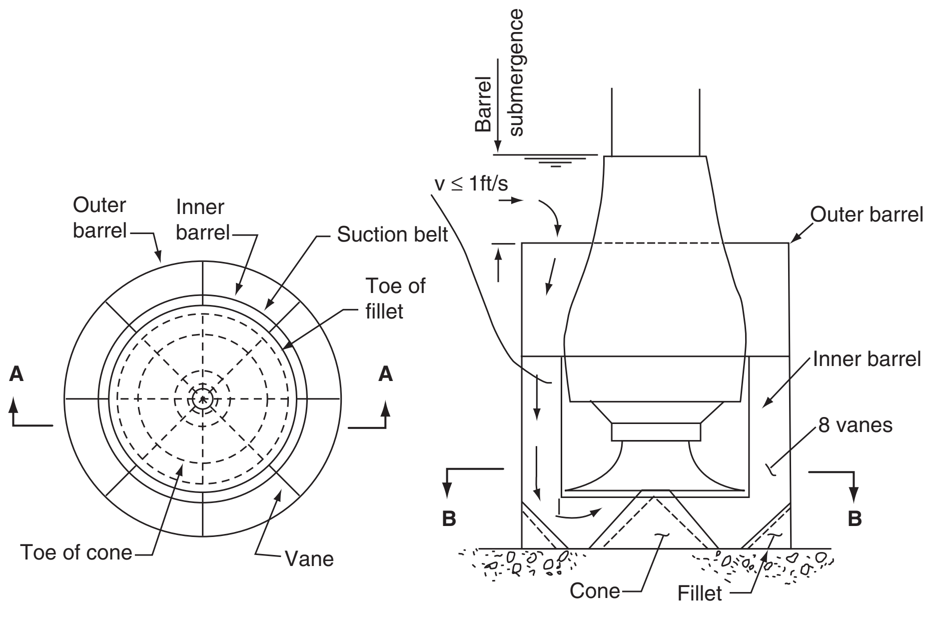


Figure 18–29. Submerged barrel (can) intake at Dublin-San Ramon Effluent Transport Station. Courtesy Brown and Caldwell, Fairbanks-Morse Pump Corp., and ENSR.

18-8. References

1. Metcalf and Eddy, Inc. *Wastewater Engineering: Collection, Treatment, Disposal, and Reuse,* 2nd ed., McGraw-Hill, New York (1978).
2. "Water supplies—Fire flow tests." Special Bulletin No. 42, National Board of Fire Underwriters, New York (July 1958).
3. Fair, G. M., J. C. Geyer, and D. A. Okun, *Water and Wastewater Engineering, Vol. 1, Water Supply and Wastewater Removal,* John Wiley, New York (1966), pp. 5-1–5-21.
4. Babbitt, H. E., J. J. Doland, and J. L. Cleasby, *Water Supply Engineering,* 6th ed., McGraw-Hill, New York (1967).
5. Peavy, H. S., D. R. Rowe, and G. Tchobanoglous, *Environmental Engineering,* McGraw-Hill, New York (1985).
6. Sanks, R. L., C. W. Reh, and A. Amirtharajah (Eds.), *Conference Proceedings, Pumping Station Design for the Practicing Engineer,* Vol. III, Department of Civil Engineering and Engineering Mechanics, Montana State University, Bozeman, MT (1981), pp. 1173–1214 and 1247–1256. Available (1) as photocopies or (2) as Vol. III on interlibrary loan from the university library.
7. Sanks, R. L., *Water Treatment Plant Design for the Practicing Engineer,* Ann Arbor Science Publishers, Ann Arbor, MI (1978).
8. Walton, W. C., *Groundwater Pumping Tests, Design & Analysis,* Lewis Publishers, Chelsea, MI (1987).

18-9. Suggested Reading

1. Walski, T. M., *Analysis of Water Distribution Systems,* Van Nostrand Reinhold, New York, (1984).

Chapter 19

System Design for Sludge Pumping

CARL N. ANDERSON
DAVID J. HANNA

CONTRIBUTORS
Robert H. Brotherton
George R. Brower
Geoffrey A. Carthew
Michael C. Mulbarger
Wyett C. Playford

The purpose of this chapter is to introduce the key issues involved in the design of sludge pumping systems. A number of significant problems, not usually met in the design of other pumping systems, are encountered in pumping sludge. These problems include:

- Variability in the nature of the pumped material
- Resulting variability in the frictional headloss characteristics
- Sludge behavior (as a non-Newtonian fluid) that makes traditional design (such as that for water pumping) inappropriate
- The need to understand complex fluid mechanics and not rely on "rule-of-thumb" approaches.

Due to the unusual and complex fluid mechanics associated with the wide varieties of sludges, designers are cautioned (1) to treat each sludge pumping application as a unique design problem, and (2) to develop site-specific design criteria based on detailed evaluation of the specific sludge characteristics. The discussion in this chapter is limited to such problems of sludge systems as the selection of pumps, pipe, valves, and cleaning and flushing stations. Problems associated with scum, screenings, grit, ash, and chemical slurry pumping are not addressed, but note that these materials behave, in general, entirely differently from wastewater sludges.

The key to pumping sludge is to use (1) a pump properly sized to develop sufficient head, and (2) a smooth pipe sized to produce the proper velocity (neither too high nor too low) without constrictions or projections and with as few bends as possible. The key to maintaining such a system is to have large, easily opened cleanouts on the pump, at any elbows on the suction side of the pump, and (where possible) at all elbows on the discharge side of the pump. Quick-disconnect air and/or water hose connections on both the suction and discharge sides of the pump are desirable where possible. Sludge lines should rarely be smaller than 150 mm (6 in.) and should preferably be larger. Glass-lined pipe is superior where high concentrations and large amounts of grease are present. Glass lining is expensive, however, whereas cement-mortar lining—which is also satisfactory—is not. The elimination of elbows and constrictions is of more practical importance than extreme smoothness for reducing friction.

The procedures presented in this chapter are applicable only to short pipelines. Long pipelines [1.6 km (1 mi) or longer] present a host of complex problems, briefly stated in Section 19-5. In general, extensive field testing and a thorough understanding of the literature and past experience with sludge pumping to develop engineering judgment is required for the successful design of systems for transporting sludge over long distances.

19-1. Hydraulic Design

Some of the traditional, well-known difficulties in evaluating the hydraulics of sludge flow [1] are as follows:

- Sludge is nonhomogeneous and has variable, peculiar properties.
- Parameters useful for water (such as Reynolds number) do not directly apply to sludge, which is a non-Newtonian fluid that may or may not behave as a Bingham plastic. Consequently, viscosity cannot be treated as a constant in pressure-drop calculations, and special methods must be used to calculate the friction loss.
- Friction losses decrease with decreasing solids concentration, a lower proportion of volatile solids, and increasing temperature. The suspended solids concentration is generally accepted as the most dominant variable.
- Due to the nature of biological solids, fresh undigested sludges and sludges from combined wastewater behave more erratically than digested sludges.
- Sludge flow can be either laminar or turbulent. Both flow regimes, shown in Figure 19-1, are widely used.

Flow in Pipelines

Sludges are most efficiently moved within a treatment system or between locations by pumping through pipelines. The unique flow characteristics of sludge present unusual difficulties in estimating the frictional flow characteristics, and the literature contains some information on hydraulic design parameters for several types of sludges, the bulk of which is on digested and raw sludges [3–8]. Before 1970, designers typically relied on empirical rules and unreliable methods for predicting frictional headloss, although the fundamental theory was available [4]. Better understanding was provided by the literature of the 1970s [9–12], and still more valuable knowledge has been gained from later studies [2, 13–18]. A brief, noncomprehensive summary of fluid mechanics applicable to sludge systems is presented below.

Headloss in Laminar Flow

Sludges generally behave as non-Newtonian fluids. Because frictional headloss depends on the fluid

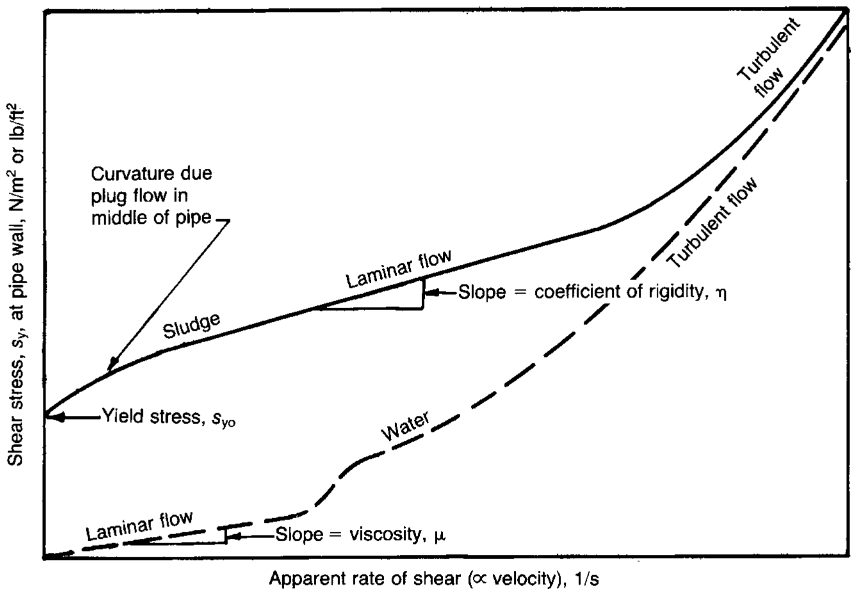


Figure 19-1. Comparison of wastewater sludge and water flowing in pipes. After EPA [2].

rheology (viscosity, elasticity, plasticity) as well as on pipe diameter and flow velocity, it can be many times the headloss for water. Thixotropic behavior (resistance to flow until the shearing force is significant), grease accumulation on pipe walls, the increase of friction with an increase of solids, and (to a lesser extent) a high proportion of volatile solids contribute to the uncertainty of predicting headloss.

For water (a Newtonian fluid), pressure drop due to flow is directly proportional to the fluid velocity and viscosity under laminar flow conditions. When critical velocity is reached, the flow becomes turbulent. Critical velocity is a function only of Reynolds number (see Chapter 3 and Figure B-1). Unlike water, sludges often move in the laminar flow region where, for non-Newtonian fluids such as sludge, the pressure drop is not proportional to flow. The precise Reynolds number at which turbulent flow characteristics are encountered is uncertain for sludges. Convention has evolved to an accepted definition of two critical velocities for sludge: a lower critical velocity (below which flow is laminar) corresponding to a Reynolds number (***R***) of 2000 and an upper critical velocity (above which flow is turbulent) corresponding to an ***R*** of 3000. As discussed below, there is some debate about the best way to evaluate the laminar-turbulent transition. At any rate sludge behaves much like a Bingham plastic, a substance with a straight-line relationship between shear stress and flow only after flow begins. A Bingham plastic is described by two constants: (1) the yield stress (Figure 19-2) and the coefficient of rigidity (Figure 19-3).

An efficient procedure for using both constants is presented in depth by Mulbarger et al. [13], whose curves are included here as Figures 19-2 through 19-5. Designers are urged to provide additional data from known projects and to perform the suggested testing to develop site-specific data, particularly when pumping sludge long distances. Once the two parameters, the yield stress and the coefficient of rigidity, are obtained, laminar flow headlosses and the two critical velocities can be obtained from Equations 19-1 through 19-4. Babbitt and Caldwell performed early pioneering work in this study of hydraulics using a method by Bingham [3, 4, 6]. The Bingham equation is applicable only under laminar flow, and its derivation includes an approximation that makes it conservative at very low velocity. The Buckingham equation [15] may be used to avoid this approximation, but it is more difficult to solve. Further difficulties arise in measuring the two parameters, s_y and η, and because sludge is not exactly a Bingham plastic. Even so, the Bingham equation is useful for estimating laminar flow headlosses. In SI units, the equation is

$$\frac{H}{L} = \frac{16 s_y}{3D_{\rho g}} + \frac{32_{\eta v}}{\rho_g D^2}$$

LIVE GRAPH
Click here to view

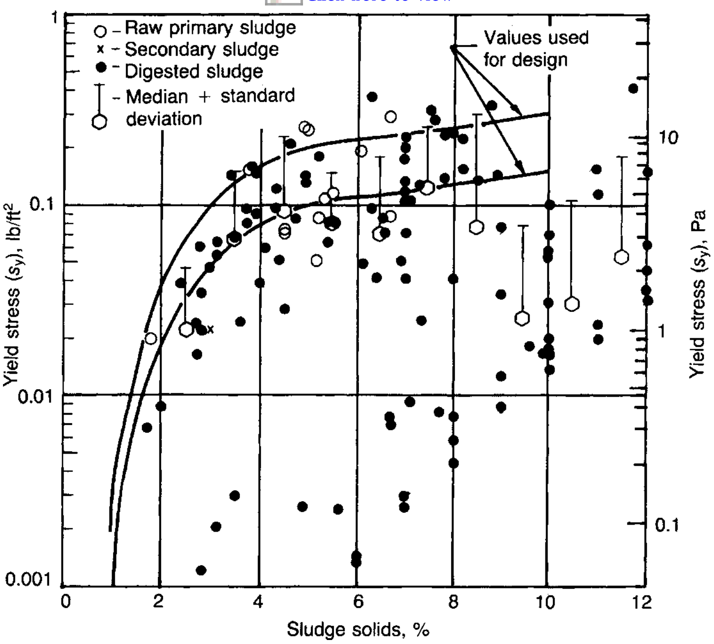


Figure 19-2. Yield stress versus sludge solids.

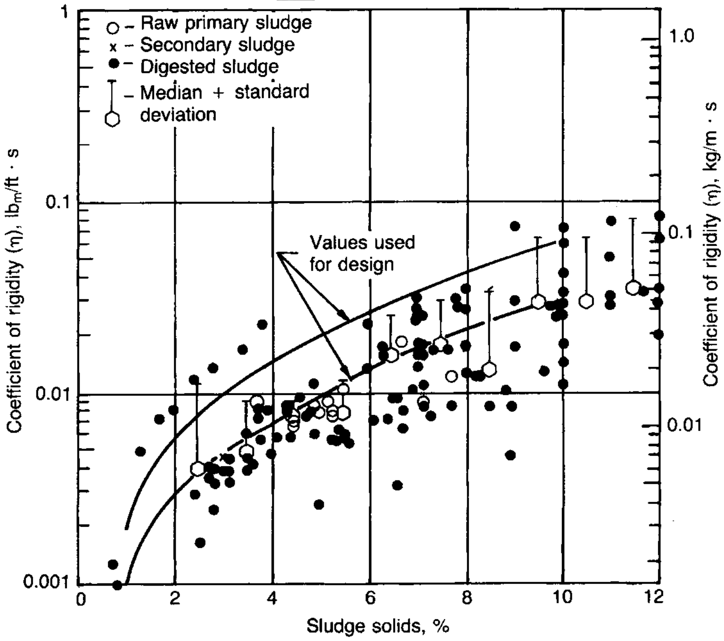


Figure 19-3. Coefficient of rigidity versus sludge solids.

where H is the headloss in meters, L is the length of pipe in meters, s_y is the yield stress in Pascals, ρ is the density in kilograms per cubic meter, D is pipe diameter in meters, η is the coefficient of rigidity in kilograms per meter-second, v is mean velocity in meters per second, and g is $9.81\,\text{m/s}^2$.

In U.S. customary units,

$$\frac{H}{L} = \frac{16s_y}{3\gamma D} + \frac{32\eta v}{g\gamma D^2} \tag{19-1b}$$

where H is the headloss in feet, L is the length of pipe in feet, s_y is the yield stress in pounds per square foot, γ is specific weight in pounds mass per cubic foot, D is the inside pipe diameter in feet, η is the coefficient of rigidity in pounds mass per foot-second (Figure 19-3), v is mean velocity in feet per second, and g is $32.2\,\text{ft/s}^2$. Although Equation 19-1 correlates well with field data, an approximation is included that could sometimes cause errors exceeding 10%. The approximation can be eliminated by solving the Buckingham equation [15] by successive approximations—a tedious task without a computer or a programmable pocket calculator. The development of site-specific data and a comparison with published information are recommended, and if the pipeline is longer than approximately 1 km (0.6 mi), site-specific data are vital. It is acknowledged that specific data are an ideal that is not likely to blanket all conditions to be actually experienced. However, it is far better to collect specific data for tempering both the designer's personal experience and the published information.

It is possible to define an apparent viscosity, μ. However, the viscosity of Bingham plastics is not a characteristic of the fluid alone, as it is with Newtonian fluids. Instead, viscosity depends on the relationship shown in Equation 19-2. The dynamic viscosity in SI units is

$$\mu = \frac{Ds_y}{6v} + \eta \tag{19-2a}$$

where μ is the coefficient of viscosity in Pascal-seconds and other terms are as previously defined.

In U.S. customary units,

$$\mu = \frac{g_c D s_y}{6v} + \eta \tag{19-2b}$$

where μ is the coefficient of dynamic viscosity in pounds mass per foot-second and g_c is $32.2\,\text{lb}_m \cdot \text{ft/lb} \cdot \text{s}^2$. The other terms are as previously defined.

The dynamic viscosity of Newtonian fluids can be measured with a viscometer. For sludge, however, the dynamic viscosity depends on the pipe diameter and velocity of flow, so it cannot be measured directly with a viscometer.

Detailed examples of calculations using this development are presented by the U.S. EPA [2].

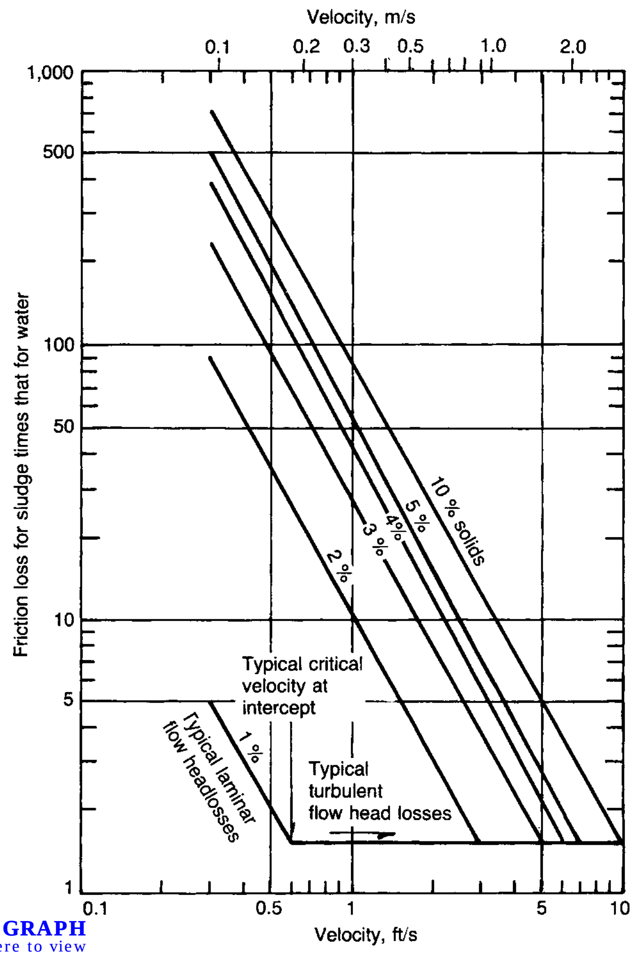
Data for Different Sludge Types

There is now a growing database for different types of sludge for estimating sludge coefficients [16–18]. These data indicate that an increase in the content of unstabilized (undigested) biological solids in a sludge causes an increase in the coefficient of rigidity and, hence, an increase in the dynamic headloss in sludge pipelines. The headlosses may be considerably higher than those indicated for the worst case in Figure 19-4. Considering the variability of the data in Figures 19-2 and 19-3, it is wise to be conservative when designing for unstabilized sludges or sludges containing metal salts, and the values indicated in Figures 19-2 and 19-4 should probably be adjusted upward (tempered with the designer's experience) by an additional 50% or more.

Laminar-Turbulent Transition

Two general approaches have been developed for the transition from laminar to turbulent flow. The first approach was developed by Babbitt and Caldwell [3, 4, 6] and many authors since then, including Mulbarger et al. [13, 14] and Carthew et al. [18]. Equations 19-3 and 19-4 present upper and lower critical velocities based on a Reynolds number between 2000 and 3000. The lower critical velocity in SI units is

$$v_{lc} = \frac{1000\eta + 1000\sqrt{\eta^2 + s_y \rho D^2/3000}}{D_p} \qquad (19\text{-}3a)$$



LIVE GRAPH
Click here to view

Figure 19-4. Recommended design curves. Headloss prediction for worst-case design.

and the upper critical velocity is

$$v_{uc} = \frac{1500\eta + 1500\sqrt{\eta^2 + s_y \rho D^2/4500}}{D_\rho} \qquad (19\text{-}4a)$$

where both critical velocities are in meters per second.

In U.S. customary units the lower critical velocity is

$$v_{lc} = \frac{1000\eta + 1000\sqrt{\eta^2 + s_y \gamma g D^2/3000}}{D\gamma} \qquad (19\text{-}3b)$$

and the upper critical velocity is

$$v_{uc} = \frac{1500\eta + 1500\sqrt{\eta^2 + s_y \gamma g D^2/4500}}{D\gamma} \qquad (19\text{-}4b)$$

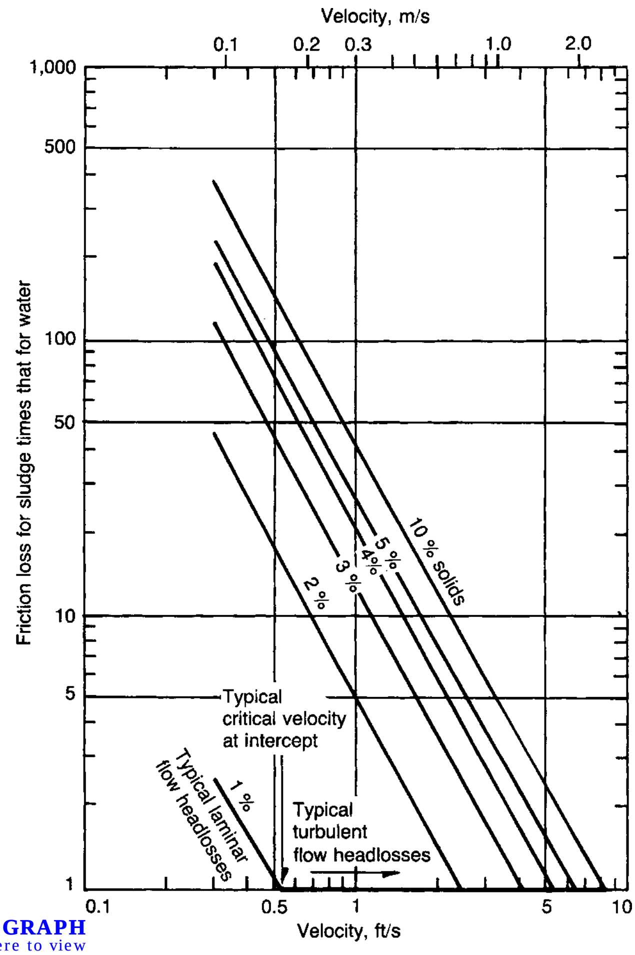
where both critical velocities are in feet per second. Equations 19-2, 19-3, and 19-4 have been used by several authors and work fairly well for most sludges.

Hanks and Dadia [19] criticized the use of the Reynolds number only and suggested the addition of the Hedstrom number for a more rigorous determination of the transition turbulent flow of a Bingham plastic. Although somewhat complex, the approach requires only the yield stress, coefficient of rigidity, density, and pipe diameter. The Hedstrom number is defined in SI units as

$$\boldsymbol{H} = \frac{D^2 s_y \rho}{\eta^2} \qquad (19\text{-}5a)$$

where $\boldsymbol{H}$ is the dimensionless Hedstrom number and other terms are as defined above. In U.S. customary units,

$$\boldsymbol{H} = \frac{D^2 s_y g_c \gamma}{\eta^2} \qquad (19\text{-}5b)$$



LIVE GRAPH
Click here to view

Figure 19-5. Recommended design curves. Headloss prediction for routine operation.

where H is the dimensionless Hedstrom number and other terms are as defined above. A special Reynolds number is also calculated (see Figure 19-6) and the plotted position of H and R on Figure 19-6 indicates laminar versus turbulent flow.

For sludge pipelines, the Hedstrom number is usually about 10^5 to 10^6. In this range, Figure 19-6 implies critical velocities not too different from those of Equations 19-3 and 19-4. If it is important to know whether flow will be laminar or turbulent (as in a sludge heat exchanger, because laminar flow would virtually prevent heat transfer), check both methods. Allow a generous safety factor because a turbulent zone near the pipe wall can coexist with a plug flow zone near the center of the pipe [20].

Headloss in Turbulent Flow

It is often practical to provide sufficient velocity to produce turbulent flow, especially for sludge with a low solids content. Under turbulent conditions, headloss vanes with velocity raised to an exponent of 1.7 to 2.0 [18]. Pipe roughness can be highly significant, and if so, procedures such as Hazen–Williams (Equation 3-9) may be used for sludge. In a smooth pipe, turbulent flow of sludge produces a headloss that is similar to or somewhat higher than the headloss with water. Major references include papers by Caldwell and Babbitt [4, 6] and Hanks and Dadia [19] and encyclopedia articles by Hanks [20] and Darby [21]. The test pipe data of Carthew et al. [18] cover turbulent as well as laminar flow, and the EPA manual [2] provides example calculations.

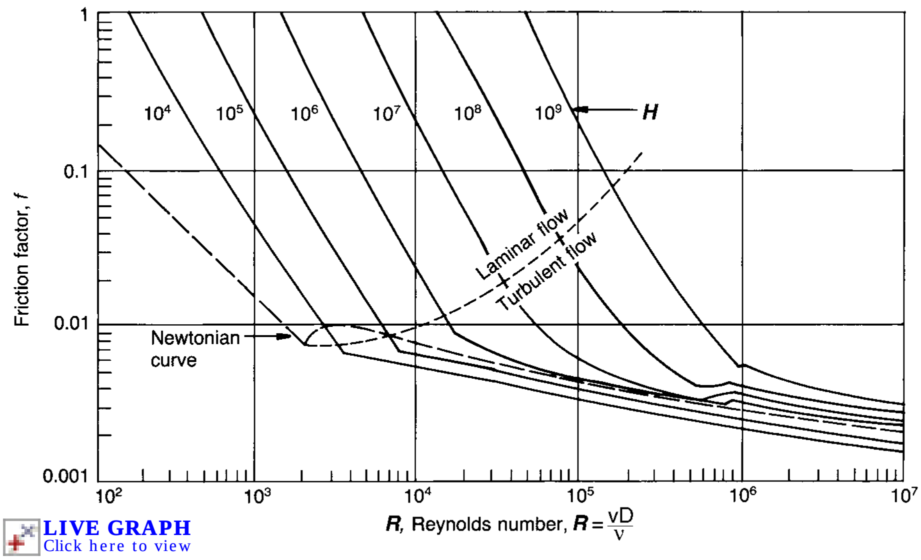
Some engineers select velocities to avoid the laminar-turbulent transition. The laminar-turbulent transition of Newtonian fluids causes unstable, unpredictable headlosses. Sludge, however, is non-Newtonian, and data [18] indicate only a change in trend at the critical velocity—not the abrupt jump that occurs in the headloss of Newtonian fluids.

There are a number of procedures and equations available for the design of either water or sludge pipelines. In general, it is just as important for a designer to understand the practical significance of pipe roughness and interior pipe conditions as it is to select the most appropriate design equation or procedure [22]. The approach of Hanks and Dadia [19] is probably the most accurate method to predict headloss under turbulent conditions. To use this method calculate the Hedstrom number from Equation 19-5 and the Reynolds number as presented in Figure 19-6 and read the friction factor from Figure 19-6. Losses in SI units are computed from

$$\Delta p = \frac{2f\rho L v^2}{D} \qquad (19\text{-}6a)$$

where Δp is differential pressure in Pascals and f is found from Figure 19-6. In U.S. customary units,

$$\Delta p = \frac{2f\gamma L v^2}{g_c D} \qquad (19\text{-}6b)$$



LIVE GRAPH
Click here to view

Figure 19-6. Theoretical friction factor for sludge analyzed as a Bingham plastic. After EPA [2].

where Δp is differential pressure in pounds per square foot and *f*, again, is found from Figure 19-6. All other terms are defined above.

As with water under turbulent conditions, the friction loss in sludge flow is approximately proportional to velocity squared but modified by a multiplier found by using Figure 19-4 or 19-5.

Mulbarger et al. [13, 14] suggested a 50% allowance for worst-case design for headloss under turbulent flow conditions. The 50% allowance is equivalent to using a Hazen–Williams *C* of about 110 instead of 140. The difference between sludge and water increases with an increase of solids under both laminar and turbulent flow as shown by recent experiments [18]. If the two Bingham parameters (s_y and η) can be determined with accuracy, the procedure of Hanks and Dadia [19] may be a more exact method. It is certainly wise to allow for an increase in pipe roughness due to moderate deposits of grit and grease. The above safety factor, however, may not suffice under some conditions, especially at higher solids concentrations. Wherever more accuracy is needed (as in all but short pipelines), the graph of Figure 19-6 should be used.

Thixotropy

Sludges sometimes exhibit characteristics of thixotropy; at other times they display characteristics similar to those of a Bingham plastic. In thixotropic behavior, flow resistance depends on the time at rest, so high pressure is needed to start the fluid moving after it has been at rest. Because some studies have shown that sludge is markedly thixotropic, it is good practice to assume that thixotropy may occur and, hence, raise the friction loss in suction piping. Suction piping should therefore be as short as practical. After passing through a pump, thixotropic effects are unlikely to be important, except when restarting a pipeline that has been shut down while full of concentrated sludge.

Simplified Headloss Calculations

Most of the above information has been available for many years but has not been widely used. A recent advance has been the compilation of Figures 19-4 and 19-5. The use of these figures is demonstrated in Example 19-1, Section 9-3, in which sludge is considered to be a Bingham plastic with values of s_y and η, as shown in Figures 19-2 and 19-3. The factors obtained from Figures 19-4 and 19-5 are compared to losses for water using the Hazen–Williams equation with a *C* value of 140. Choose velocity and percent solids, compute the friction headloss for water, and multiply the headloss by the factors from Figure 19-4 (worst case) or Figure 19-5 (routine design). These figures cover laminar and turbulent flow conditions, both of which are widely used. At lower velocities, sludge friction decreases only slowly as velocity drops. With water, however, friction drops sharply as velocity drops. Hence, the sludge/water headloss ratio increases as velocity decreases. Above a critical velocity, sludge flows more like water and friction varies as velocity to an exponent of about 1.85. Figure 19-4 includes a factor of 1.5 in this region. Headloss also varies with solids content, especially at the lower velocities.

Consider the following when using Figures 19-4 and 19-5:

- Figures 19-4 and 19-5, supplemented with data published since 1980, are far more accurate than popular references published before 1970. These figures were proposed by Mulbarger [13, 14] to represent the general case(s) based on further examination of data derived in his earlier work [13].
- *Figures 19-4 and 19-5 must be used correctly*. The water headloss is computed as a basis for the sludge headloss. Most engineers are comfortable with the approach of (1) calculating headlosses for water for a selected pipe roughness and then (2) multiplying the headloss by a selected sludge factor. The factors in Figures 19-4 and 19-5 are based on an H–W *C* factor of 140 for water. Do not use a different *C* factor even if you would do so for water. Minor losses should never be computed as equivalent pipe lengths but should be computed separately. Minor loss coefficients may be taken as roughly the same for sludge as for water.
- Some sludges may exceed the "design worst-case" curves of Figure 19-4. For undigested activated sludge, the design worst-case should be raised by a factor of 1.5 until more data are available. A similar factor may apply to primary and secondary sludges that contain aluminum or iron salts.
- The charts should be used with great caution for pump suction piping in the calculation of available net positive suction head (NPSHA). Even if NPSHA appears to be adequate, long suction pipelines represent poor practice and should not be allowed because of possible thixotropy.
- Lower velocities and larger pipe may or may not reduce total friction headlosses. Calculate headlosses for at least two pipe sizes and evaluate capital, operating, and maintenance (pipe cleaning) costs before deciding the pipe size (see Example 19-1).
- Figures 19-4 and 19-5 are inadequate for highly accurate results. Accuracy is especially important when the pumped distance is large or when signifi-

cant variations in sludge are expected. A strongly suggested guideline is to conduct a specific testing program whenever (1) the pipeline is more than 1.6 km (1 mi) long, (2) the sludge is expected to be at or over 7% solids, (3) a fluctuation in fluid characteristics of the sludge is likely to occur, or (4) calculations based on Figure 19-4 indicate a friction head greater than 15 m (50 ft).

- Do not assume that resistances obtained by testing will never be exceeded. Design the facility to allow for diluting the sludge to ensure an adequate installation.

19-2. Types of Pumps

Depending on the application, sludge pumps can be centrifugal, vortex, combined screw-centrifugal, air lift, or positive-displacement pumps such as plunger (or piston), progressive cavity, diaphragm, rotary lobe, and high-pressure piston. These pumps are described in Chapter 11 and compared in Table 25-10. In general, centrifugal pumps are most suitable for pumping large volumes of sludge at low (2 or 3%) concentrations, whereas higher concentrations and intermittent pumping warrant the use of a positive-displacement pump.

Centrifugal Pumps

The preferred choice of centrifugal pumps for sludge applications is the nonclog pump because of the need for larger passages through vanes to prevent obstructions. In general, pumps for moving sludge fall in the low end of the mixed-flow range of specific speeds. Sludge pumping applications for which centrifugal pumps are suitable are limited to relatively low (<1%) concentrations if pumping raw primary sludge, and moderate (<5%) concentrations if pumping digested and return sludge. Sludge pumps must be sturdier with larger, more reliable bearings, shafts, seals, and other internal components. Reliability is preferred over pumping efficiency.

One of the major problems in selecting a centrifugal pump for sludge is the difficulty in obtaining the proper size for both flow and head conditions. (An example of the range of design considerations is presented in Example 19-1.) Due to the wide range of expected flow conditions, variable-speed (V/S) control is usually used. The discharge valve should not be throttled in an attempt to control flow because throttling promotes increased valve wear, plugging of the pipeline, and wasted energy. Velocity criteria in the sludge piping system are often compromised when the pump capacity is reduced and the head is raised by throttling a discharge valve.

The impeller can be specified to include one or more vanes and to be enclosed, semi-enclosed, or open. Efficiencies tend to be higher for enclosed impellers and lower for open impellers. Closed impellers are best suited for large return-activated sludge pumps or raw waste pumps with suction and discharge pipes of 200 mm (8 in.) or larger. Open or recessed impellers are better for smaller-volume pumping applications involving stringy material and for suction and discharge piping sizes of 100 to 150 mm (4 to 6 in.). Centrifugal pumps are not recommended for very small volumes of sludge with suction and discharge piping less than 100 mm (4 in.). Note that many designers object to pipe smaller than 150 mm (6 in.) and refuse to use pipe smaller than 100 mm (4 in.) for sludge in any circumstances.

Vortex Pumps

The impeller of a vortex pump (also called a "recessed impeller pump") develops a vortex in the incoming fluid so that most of the solids never touch the impeller. The openings through the pump are large because the impeller is almost entirely out of the liquid flow path, so the plugging often encountered with nonclog pumps is eliminated. (Despite the name, nonclog pumps often clog.) Hence, the size of solids is limited mostly by the size of the suction and discharge nozzles. Vortex pumps have the advantages of less maintenance, greater reliability, and greater ease of parts replacement. But the sizing of vortex pumps is critical because they have very flat system head curves—a major complication when pumping thicker sludges (because of the highly variable pressure drop). Internal fluid recirculation is a concern, the efficiency is very low (only 45% or less), and in some types the impellers cannot be trimmed. In summary, the vortex pump is a desirable selection when the sludge pumping system requirements are such that reliability is paramount in the pumping of sludge with a low (say, 4% or less) concentration of solids. Vortex pumps can pump up to about 6% solids, but a variable-speed drive and a reliable flowmeter are required to allow process control at higher solids contents. Rugged mechanical construction also becomes critical as varying process conditions may require the pump to operate away from its best efficiency point (BEP). A positive means of moving sludge through the suction piping into the pump must be available if pumping is to be reliable.

Horizontal mounting is recommended, because vertical installations can result in noise and/or cavitation.

Restrictions are often built into the discharge passages of the pump to steepen the flat system head curves, so do not fail to specify the size of sphere that must pass entirely through the suction, discharge, and internal passages of the pump.

Combined Screw-Centrifugal Pumps

Recognizing the inherent problems in pumping sludge, pump manufacturers have periodically offered innovative designs. One type (such as the Chicago Pump Scru-Peller®), used for half a century, combines a two-flight screw conveyor with an open-type, two-blade centrifugal nonclog impeller. It was designed and applied particularly for primary sludge pumping. The sharp stellite edges of the screw conveyor and impeller and the stellite shear bars in the screw housing cut or shred stringy materials.

The advantages of the Scru-Peller® pump are its ability to (1) cope with stringy materials, (2) produce a steady discharge, (3) develop heads up to 50 m (160 ft) at 160 m^3/h (700 gal/min), and (4) minimize clogging effects and associated downtime and maintenance. The disadvantages are primarily higher first cost and operating costs as compared with costs for a centrifugal nonclog pump, but less as compared with costs for a vortex pump. The pump is best applied for intermittent pumping instead of continuous duty.

A pump that has often been used to replace an inefficient vortex pump is the combined screw-centrifugal impeller pump (such as the WEMCO Hidrostal® in Figure 11-36). This type of pump has been applied to sludges for about 16 years in the United States. The design was developed to pump live fish without damage for canneries. The impeller has two sections (screw and centrifugal) that combine the clog-free features of a vortex pump with the gentle action of a screw pump at the higher efficiencies of a centrifugal pump. The large open channels in the fluid path provide pumping without the abrupt changes in direction that capture stringy materials. It is claimed that the corkscrew action of the screw section helps to start thicker sludges moving. The advantages of this pump include (1) higher (up to 80%) efficiencies, (2) nonclog operation, (3) gentle fluid action without high turbulence, (4) a steep head-capacity curve, (5) low NPSH requirements, and (6) a nonoverloading power curve. The disadvantage is that for a given flow rate, it is larger and more expensive than a centrifugal pump.

Both of these pump types may be useful when the combination of steady discharge with clog-free operation is desired. The pumping of primary sludge and sludges with stringy and fibrous materials are said to be good applications for both of these pumps.

Air Lift Pumps

Air lift pumps are sometimes used for the thin sludges produced at oxidation ditches or other small, extended aeration-activated sludge plants. Because the control of pumping rate is limited for a single pump, they are often installed in multiples. Air lift pumps are used for activated sludge return where the lift is low—typically less than 2 m (6 ft)—and the pipeline is short [23]. The pump itself is almost indestructible and, except for the splashing and the daily need for cleaning, requires virtually no maintenance. The air blowers, however, do require maintenance. The pump has several disadvantages: (1) it is difficult to regulate flow (so process control is also difficult); (2) the pumpage changes erratically with small variations in the air delivered; and (3) the efficiency is low (typically less than 30%). The best way to regulate discharge is to maintain the maximum air pressure entering the pump and close (or open) the air exhaust ports at the bottom of the pump column. Flushing connections are recommended.

Plunger Pumps

The plunger (or "piston") pump has long been a workhorse and is one of the most commonly used types for pumping sludge. The piston is driven by an exposed drive crank whose eccentricity is adjustable and, thus, varies the stroke length to suit the desired output. The discharge from a plunger pump is therefore a series of interrupted sinusoidal pulses, so the "pulsed" flow (at the peak of the sine curve) is greater than the average flow. The pulsing effect is reduced with multiple piston (duplex or triplex) pumps and/or with air chambers. Design calculations must be based on the peak of the pulsing flow. The approximate flow variations as a percentage of the average flow are shown in Table 19-1. Because the tabular values are based on the assumption of a rigid machine and an incompressible fluid, the observed effects may be somewhat less. Nevertheless, discuss this aspect of pump performance with the pump manufacturer. Air chambers should be specified on the suction side when the suction head is more than 4.6 m (15 ft), and they should always be furnished on the discharge side. Air chambers, or pulsation dampeners, induce a more uniform flow. The volume of an air chamber or dampener for simplex pumps should be six to eight times the displacement of one plunger

Table 19-1. Approximate Flow Variations as a Percentage of the Average Flow in Plunger Pumps[a]

Pump configuration	Number of plungers	Flow variations as a plus or minus percentage of average flow
Simplex	1	320
Duplex	2	160
Triplex	3	125
Two duplexes	4	133

[a]After Henshaw [24].

per stroke (V_s); for duplex and triplex pumps, the volume should be three to four times V_s [25]. The kinematics of reciprocating pumps produce periodic variations in flow, velocity, and head conditions, so design the system to prevent the coincidence of harmonics with the fundamental harmonics of the suction or discharge piping [26].

The advantages of plunger pumps are (1) superior reliability, (2) ease and simplicity of maintenance and repair, and (3) low cost of replacement parts. But the most important advantage is their constant capacity when the head varies or when the head cannot be reliably estimated. The pump motor must, of course, be large enough for the maximum pressure conditions. A procedure for calculating the head for a plunger pump is given in Example 19-2 in Section 9-3.

The disadvantages of plunger pumps include: (1) pulsating flow, which tends to induce "rat-holing" in the sludge sump, (2) noise and vibration, (3) filth and smell, (4) frequent replacement of the balls in the ball check chambers, and (5) the reluctance of some operators to maintain or repair them. The piston frequently tends to tilt during the downstroke, thus causing rapid wear. Piston guides are recommended to reduce this tendency.

Plunger pumps should be designed and operated at the maximum discharge stroke setting for the best operation. Two in-line ball check valves on both sides of the plunger pump are also recommended (1) for better backflow protection if back siphoning from the discharge to the suction tank is possible and (2) to make the operation of the pump smoother if the suction head is high. The use of two ball check valves along with other design features should be discussed in detail with the pump manufacturer. Ball check valve seats should be stainless steel or bronze and should be replaceable without disturbing the valve chamber piping.

If a pulsating discharge rate is acceptable, positive-displacement reciprocating pumps are recommended as the primary means of sludge pumping due to their reliability in pumping the thicker and heavier sludges.

Progressing Cavity Pumps

Progressing cavity pumps (see Figure 11-6) are advantageous if a smooth, steady, predictable discharge is required. The pump produces a relatively constant output of flow because as one cavity reduces in size, the opposing cavity is increasing at the same rate. Flow capability is a function of rotor speed, rotor diameter, rotor eccentricity, and helix pitch. Pressure capability is a function of the number of rotor/stator stages [2]. These pumps are available with a wide variety of flow and head ratings and are very useful when pumping thicker sludges if they do not contain grit, debris, or abrasives.

Because a progressing cavity pump acts as a check valve when inactive due to the minimal clearances between the stator and rotor, check valves may be eliminated, but isolation valves must always be provided. If static back pressure is high, however, the pump might rotate backward unless either a check valve or antireverse gearing is used. The size of solids that can pass through a progressive cavity pump is limited to less than 45 mm (1¾ in.) in diameter, so the sludge must not contain large solids. Unless sludges are relatively free from grit and other abrasives, wear is accelerated, which causes costly and frequent maintenance. Operating speed is limited to about 300 rev/min or less to minimize wear. However, the pump clogs at very low speeds. A general recommendation is to provide an adequate velocity through the system piping while (at the same time) pumping at 200 rev/min or slightly more. Pump speed should be discussed in depth with the pump manufacturer.

Progressing cavity pumps have a reputation for a high rate of wear and the requirement of more maintenance labor than any other widely used type (see also rotary lobe pumps in this section). Wear of the metal rotor on the elastomeric stator increases clearances and reduces flow and pressure, so rotors and stators must be replaced frequently—typically once a year and sometimes more often. Replacement parts are expensive and difficult to install. Ample room is needed to slide internal pump parts out of the casing, so the pump requires significantly more floor space than other pumps. A recommended installation is shown in Figure 19-7.

Diaphragm Pumps

Diaphragm pumps, especially the spring-assisted, air-operated type (such as the Dorr–Oliver type ODS), have been successfully applied to low head sludge pumping service for at least 20 years. The spring acts

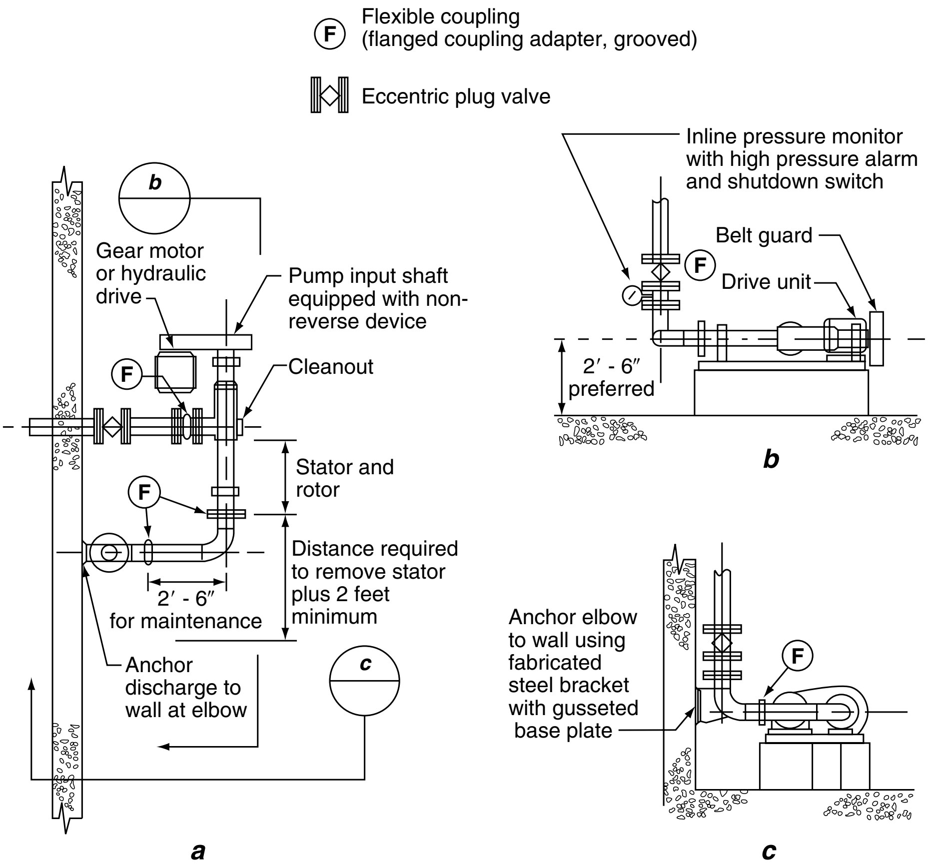


Figure 19-7. Installation recommendations for progressing cavity pumps. (a) Plan; (b) Section b; and (c) Section c.

on the diaphragm to draw sludge into the pumping chamber, so the pump is capable of a modest suction lift although high suction lift cannot be sustained reliably [27]. Compressed air forces the sludge out of the pumping chamber. Check valves on the inlet and discharge connections prevent backflow. Stroke adjustment and timer control of the air supply solenoid permits a wide range of pumping rates. Nonassist and pneumatic-assist types are available. The pump can be furnished with ball-type check valves in either singles or pairs with quick opening covers (recommended). Capacities range up to 34 m^3/h (150 gal/min) as water (and 60% or more of such values as sludge) at 350 kPa (50 $lb/in.^2$) gauge. Compressed air requirements are significant, however, and the air exhaust can be quite noisy. Diaphragm pumps have a good reputation, problems are few, and they are suitable for small treatment plants.

The disadvantages include: (1) pulsating flow, (2) vibration, (3) requirement for compressed air and consequent inefficiency, and (4) high cost per unit of flow. The type with a single valve for suction and discharge is not suitable for stringy materials, but valves in tandem pairs can be obtained or the pump can be preceded by a grinder.

Membranes typically last about two years if the air supply and the solenoid air valve are properly adjusted. Cracks in the membrane can be readily detected, so the membrane can be replaced before it ruptures entirely. A ruptured membrane allows the air lines to fill with sludge to the muffler, but the muffler liner quickly clogs so that little sludge can spill. Cleaning the air lines is a chore; membranes should be replaced well before cracks develop. The O&M manual should contain a warning about the danger of flooding the station with sludge if a muffler and liner are not in place. Better protection is afforded by mounting the muffler high enough to prevent any possible spill.

Another diaphragm pump option consists of a low-pressure pump using dual free or unbalanced diaphragms driven by an eccentric shaft. The features of the pump include a mechanical drive, self-priming

characteristics, ability to operate at speeds reduced to a "snore" condition, the elimination of check valves and moving seals, and an unusually low number of moving parts. The pump is valveless and glandless and is able to operate dry for a short time without damage. It has been available since the mid-1980s and has been successfully applied in over 2000 sludge pumping applications. The pump can reportedly handle up to 8.5% solids [28]. This type of diaphragm pump is limited to a TDH of 30.5 m (100 ft) (or less for the largest model). Maximum solids-handling capacity is 25 mm (1 in.). The pump tends to retain its performance over its operating life because disc and seal wear is low. Be cautious, however, in specifying this pump because, when overpressurized, it can break internally without warning. Applications should be reviewed with the manufacturer in detail.

Rotary Lobe Pumps

The rotary lobe pump has been applied to wastewater sludges only since 1970 but with promising results. Its advantages are: (1) positive displacement with little agitation or shear, (2) high efficiency, (3) self-priming, (4) compactness, (5) good accessibility for maintenance, and (6) sizes up to 0.12 m^3/s (2000 gal/ min). The disadvantages are: (1) close clearances that are subject to wear and abrasion (which makes sludges containing much grit, abrasives, debris, and rags unsuitable), (2) pulsing flow, although the "twisted lobe" pump of Figure 11-5 produces uniform flow. Rotary lobe pumps are so compact that it would be difficult to replace an unsatisfactory unit with another type because of space limitations.

Difficulties have been experienced with a number of rotary lobe pumps (installed between 1980 and 1988) in which rapid wear of lining materials and sometimes catastrophic loss of liners have occurred, according to a survey conducted in late 1988 [29]. Some operators reported rapid wear of end plates and a resulting loss of volumetric capacity. The causes were sorption of water by the liners and failure of the adhesive. The problems have since been corrected [30].

High-Pressure Piston Pumps

Belt conveyors and screw conveyors are usually used to move dewatered sludge to truck loading points or incinerators. An alternative recently developed is the use of specially designed piston pumps to force dewatered sludge through pipes.

The basic pump design is derived from concrete pumps. A high-pressure piston pump typically has two pumping cylinders of small diameter and long stroke. Each cylinder is driven by the pressure of hydraulic oil in an adjacent drive cylinder. Hydraulic oil is also used to power-operate the valves, because ordinary check valves are not satisfactory in plastic fluids with high yield stress. At the pump intake, a short screw conveyor may be used to force sludge into the pump. The installation is mechanically complex and requires considerable floor space, but other conveying systems have the same drawbacks.

Very high pressure drops occur, although data are sparse. In one unpublished test of a sludge containing 36% solids, pressure drops of about 3000 kPa (400 lb/in.^{2}) occurred after thixotropic breakdown in a pipe only 61 m (201 ft) long [31]. The pressure needed to achieve thixotropic breakdown after a day of rest was 4000 kPa (600 lb/in.^{2}). The pipe diameter was 130 mm (5 in.). In spite of the drawbacks, a pumping system may sometimes be more cost effective than a conveyor belt.

Pump Selection

For sludge service, maintenance-free operation with pumps that rarely clog is preferable to the operation of more efficient units in which these advantages are compromised. Should the process design require a centrifugal pump, however, it is good design practice to add a standby positive-displacement pump to ensure that thicker sludge can be moved.

19-3. Pumping System Design

The most important criterion for design is to have sludge pumps capable of transferring the design quantity of solids throughout the range of expected solids concentrations within the required time interval. In most treatment plant sludge pumping, the consistency of the sludge changes during pumping. At first, the most concentrated sludge at the bottom of the basin is pumped. After most of that sludge is removed, a more dilute sludge with characteristics ranging from semisolids to those essentially of water is pumped. This variability in the fluid characteristics causes the pump to operate at different points along its design curve. Designers must take this variability into account and size the pump and motor for all expected service conditions. If the pump motor is not sized correctly, the pump may become overstressed or the motor damaged.

It is desirable to pump as uniformly as practical, but with small systems it is better to pick larger pumps and pipelines than are needed and pump

intermittently with timers. The pump should operate long enough (approximately 10 to 15 min) to heat the motor to the operating temperature. The pump cycle time (subject to review with the manufacturers of the pump, motor, and motor starter) should be at least 15 min. The velocity of pumping is important. The friction in starting the sludge movement is much greater than in keeping the sludge moving once it is started. In pumping thick sludges, it may be desirable to flush the line with a more dilute concentration of sludge before the pump is turned off.

Calculation of System Curves

Consider (1) the worst possible combination of variability of solids concentrations, and (2) discharge to the several possible outlets. If several pumps are to discharge to a common pipe, each pumping route must be considered for sizing the piping and determining the dynamic headlosses. Two typical sludge pumping system schematics for an activated sludge plant are shown in Figures 19-8 and 19-9. Note the various pumping routes available in both figures. Furthermore, the static head may change substantially during pumping. Consequently, four system curves are essential for analysis.

- *Maximum system curve:* The pressures that occur with maximum solids concentration. The system operates along this curve when the sludge is most concentrated and when it is being pumped through the piping system route that results in the maximum static and dynamic loss. This route usually has the greatest length and the smallest pipe diameter, but differences in the static lifts and heads between alternative routes must also be considered.
- *Average system curve:* The pressures that occur at average operating solids concentration.
- *Minimum system curve:* The pressures that occur at minimum solids concentration.
- *Water system curve:* The lowest pressures that can occur when water without solids is pumped through the critical route of the piping system. The critical route is the one with minimum static and dynamic loss, which is usually the pipe with the shortest length or the largest diameter, but differences in static lifts between alternative routes must also be considered.

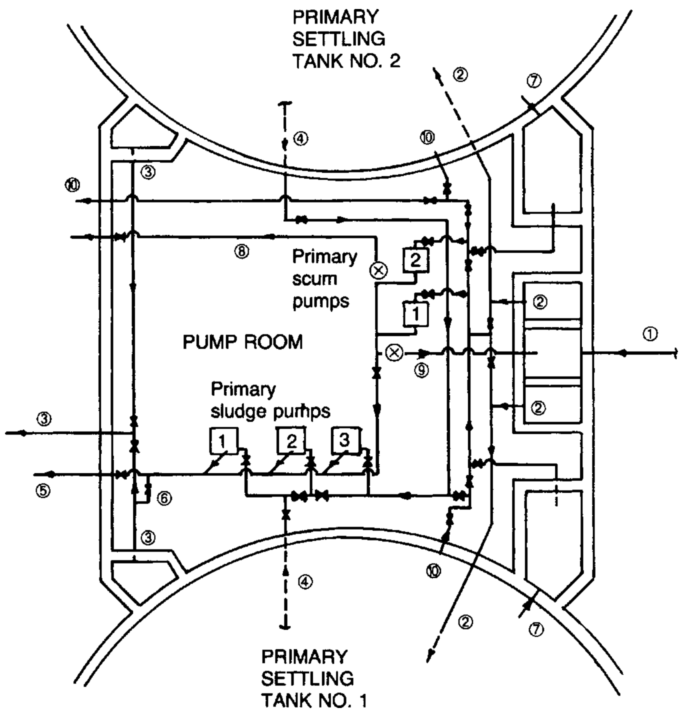


Figure 19-8. Primary sludge and scum piping. 1, raw wastewater from pumping station; 2, raw wastewater to settling tanks; 3, settled effluent to aeration tanks; 4, sludge drawoff; 5, sludge to sludge thickener (normal route) or to digesters (alternate route); 6, sludge bypass to aeration tanks; 7, scum removal; 8, scum pumped to truck for disposal; 9, scum semiliquid to influent well; 10, tank drains to plant sewer; ⊗, air-operated valves. Courtesy of Stearns & Wheeler Engineers and Scientists and J. Kenneth Fraser & Associates.

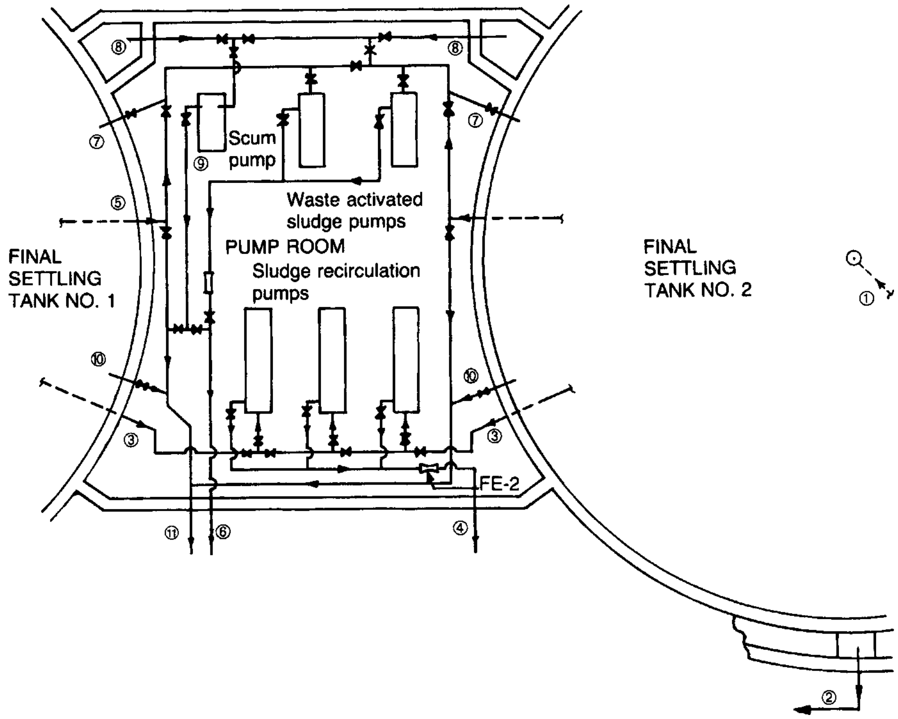


Figure 19-9. Return and waste activated sludge piping. 1, influent from aeration tanks; 2, settled effluent to chlorine contact tanks; 3, recirculation sludge drawoff; 4, recirculation sludge to aeration tanks; 5, waste activated sludge (WAS) drawoff; 6, WAS to thickener via control building; 7, sidewall drawoff; 8 scum drawoff; 9, scum to pumping station via plant sewer; 10, tank drain; 11, drain to plant sewer. Courtesy of Stearns & Wheeler Engineers and Scientists, and J. Kenneth Fraser & Associates.

In the design of centrifugal pumps, the motor power required is maximum for the water system curve because the flow rate is maximum. However, in the design of positive displacement pumps, the maximum system curve is the most critical because the pump must deliver its unvarying flow rate with the most concentrated sludge.

Design Procedure for Centrifugal Pumps

A design procedure for selecting a centrifugal (or vortex) sludge pump and motor [14] is outlined below for the plant shown in Figure 19-8. For illustrative purposes, it is assumed that only one pump discharges at a time and that each outside duty pump is dedicated to a separate settling tank. The middle pump is the common standby and is set up to take over the duty of either pump 1 or pump 3. Were several pumps to operate simultaneously, the total flow pumped through the system must be considered in determining total head. Note also that the scum pump is a positive-displacement pump that can also serve as a backup to the centrifugal pumps.

1. Plot the maximum, minimum, average, and water system curves (see Figure 19-10 in Example 19-1).
2. Examine the curves in Step 1, choose a candidate pump, superimpose a plot of the pump H-Q curves, and locate the operating points on each of the system curves.
3. If a constant-speed (C/S) pump is to be used, select the smallest impeller size that intersects the three sludge system curves beyond the required flow. For vortex pumps with a choice of impeller sizes, use the largest impeller diameter and be sure the pump has a positive suction head (including allowances for headlosses at design flow rates).
4. If a variable-speed (V/S) pumping unit is to be used, select the maximum pump speed required for the three sludge system curves.
5. If a C/S pump is to be used, the intersection of the pump H-Q curve for the impeller found in Step 3 with the water system curve determines the power required. Either calculate the power or find the smallest horsepower-designation curve beyond this intersection. A C/S pump installation makes

flow difficult to control. A V/S pump should ordinarily be used.

6. If a V/S drive is to be used, consult the pump manufacturer to determine the maximum speed of the drive. The water power required is determined by the intersection of the maximum speed curve and the water system curve. Add corrections for the efficiency of the pump and the net efficiency of the driver to convert water power to the required power of the motor.
7. The manufacturer's pump performance curves must not be extrapolated because cavitation or vibration problems could occur at flows higher than those covered by the manufacturer's curves. If the minimum system curve extends into the region beyond that covered by the pump curves, select a larger pump. If no suitable centrifugal pump can be found, consult the manufacturer. Always verify the selection of sludge pumps with several manufacturers.

Example 19-1
Design of a Vortex Pump System for Sludge

Problem: In the process design of a municipal activated sludge plant, a maximum of 13,600 kg/d (30,000 lb/d) on a dry basis of primary sludge solids (with no industrial component) must be transferred. The layout consists of two duty pumps and a standby pump, as shown in Figure 19-8. Note that there are several piping discharge routes, but for the purpose of this example, consider a system of 107 m (350 ft) of 150-mm (6-in.) piping and a static head of 3.2 m (10.5 ft) at average design conditions.

For this example, (1) plot the system curves, (2) select pumps for the V/S drives, (3) select the driver, and (4) check the computed headloss by a different method.

Solution: For problems of this kind, consider:

- Several discharge piping routing options with varying pipe sizes, lengths, and static heads
- Varying static heads on any or all piping routes due to operating conditions
- The likely design feature of different-sized suction and discharge piping
- The varying mass and concentration of solids that must be transferred
- Variations in flow rates over the design life from initial operating conditions to final design conditions. (Consider ranges in solids concentration at both initial and final design conditions.)

Determine pump capacity. The required pumping capacities to transfer 13,600 kg (30,000 lb) of dry sludge solids per day are found from the following expressions:

SI Units

V_s = volume of sludge, m^3/d
M_s = mass of dry solids, kg/d
ρ = density of water, $1000\,kg/m^3$
S_s = specific gravity of sludge
P_s = percentage of solids/100

U.S. Customary Units

V_s = volume of sludge, gal/d
M_s = mass of dry solids, lb/d
γ = specific weight of water, $62.4\,lb/ft^3$
S_s = specific gravity of sludge
P_s = percentage of solids/100

Solve the expression for cubic meters per day (or gallons per day) for a reasonable range of solids (10, 5, 3, and 1%). For the purpose of this example, assume the specific gravity of the sludge is 1.00.

10% solids:

$$V_s = \frac{13{,}600}{1000(1.0)(0.1)} = 136\,m^3/d \qquad V_s = \frac{30{,}000 \times 7.48}{62.4(1.0)(0.1)} = 36{,}000\,gal/d$$

5% solids:

$$V_s = \frac{13{,}600}{1000(1.0)(0.05)} = 272\,m^3/d \qquad V_s = \frac{30{,}000 \times 7.48}{62.4(1.0)(0.05)} = 72{,}000\,gal/d$$

3% solids:

$$V_s = \frac{13{,}600}{1000(1.0)(0.03)} = 453\,m^3/d \qquad V_s = \frac{30{,}000 \times 7.48}{62.4(1.0)(0.03)} = 120{,}000\,gal/d$$

1% solids:

$$V_s = \frac{13{,}600}{1000(1.0)(0.01)} = 1360\,\mathrm{m^3/d} \qquad V_s = \frac{30{,}000 \times 7.48}{62.4(1.0)(0.01)} = 360{,}000\,\mathrm{gal/d}$$

Estimate two duty pumps, each dedicated to a separate settling tank (including the third pump as a common standby) and each operating 6 h/d and at different times. The discharge rate required of each pump is

10% solids:

$$\left(\frac{136\,\mathrm{m^3/d}}{2\text{ pumps}}\right)\left(\frac{1\,\mathrm{d}}{6\,\mathrm{h}}\right) = 11.3\,\mathrm{m^3/h} \qquad \left(\frac{36{,}000\,\mathrm{gal/d}}{2\text{ pumps}}\right)\left(\frac{1\,\mathrm{d} \times 1\,\mathrm{h}}{6\,\mathrm{h} \times 60\,\mathrm{min}}\right) = 50\,\mathrm{gal/min}$$

5% solids:

$$\left(\frac{272\,\mathrm{m^3/d}}{2\text{ pumps}}\right)\left(\frac{1\,\mathrm{d}}{6\,\mathrm{h}}\right) = 22.7\,\mathrm{m^3/h} \qquad \left(\frac{72{,}000\,\mathrm{gal/d}}{2\text{ pumps}}\right)\left(\frac{1\,\mathrm{d} \times 1\,\mathrm{h}}{6\,\mathrm{h} \times 60\,\mathrm{min}}\right) = 100\,\mathrm{gal/min}$$

3% solids:

$$\left(\frac{453\,\mathrm{m^3/d}}{2\text{ pumps}}\right)\left(\frac{1\,\mathrm{d}}{6\,\mathrm{h}}\right) = 37.8\,\mathrm{m^3/h} \qquad \left(\frac{120{,}000\,\mathrm{gal/d}}{2\text{ pumps}}\right)\left(\frac{1\,\mathrm{d} \times 1\,\mathrm{h}}{6\,\mathrm{h} \times 60\,\mathrm{min}}\right) = 167\,\mathrm{gal/min}$$

1% solids (essentially water):

$$\left(\frac{1360\,\mathrm{m^3/d}}{2\text{ pumps}}\right)\left(\frac{1\,\mathrm{d}}{6\,\mathrm{h}}\right) = 113\,\mathrm{m^3/h} \qquad \left(\frac{360{,}000\,\mathrm{gal/d}}{2\text{ pumps}}\right)\left(\frac{1\,\mathrm{d} \times 1\,\mathrm{h}}{6\,\mathrm{h} \times 60\,\mathrm{min}}\right) = 500\,\mathrm{gal/min}$$

Fluid velocities at pump capacities. Find the velocity in 150-mm (6-in.) pipe at the pump discharge rates above for maximum (10%), average (5%), low (3%), and minimum (1%) solids concentrations:

		Velocity (m/s)			Velocity (ft/s)	
Sludge concentration	Discharge (m^3/h)	150-mm pipe	100-mm pipe	Discharge (gal/min)	6-in. pipe	4-in. pipe
10%	11.3	0.18	0.40	50	0.6	1.3
5%	22.7	0.35	0.80	100	1.1	2.6
3%	37.8	0.59	1.34	167	1.9	4.3
1%	113	1.77	4.02	501	5.7	12.9

The velocities for 150-mm (6-in.) pipe are somewhat low for sludge service, so tabulations for 100-mm (4-in.) pipe are added.

(1) Plot system curves. Plot system curves for 10, 5, 3, and 1% solids flowing in 150-mm (6-in.) and 100-mm (4-in.) pipe by determining headlosses on the basis of Hazen–Williams $C = 140$. See Figures 19-10 and 19-11. Then multiply the headlosses by the friction factors from Figure 19-4 (for worst case) or from Figure 19-5 (for routine operation). Headloss predictions for worst-case design are recommended for the maximum system curve. Headloss predictions for routine operations are recommended for all other system curves. The calculation for one point on the 5% solids, routine curve for 150-mm (6-in.) pipe is given below to illustrate the procedure.

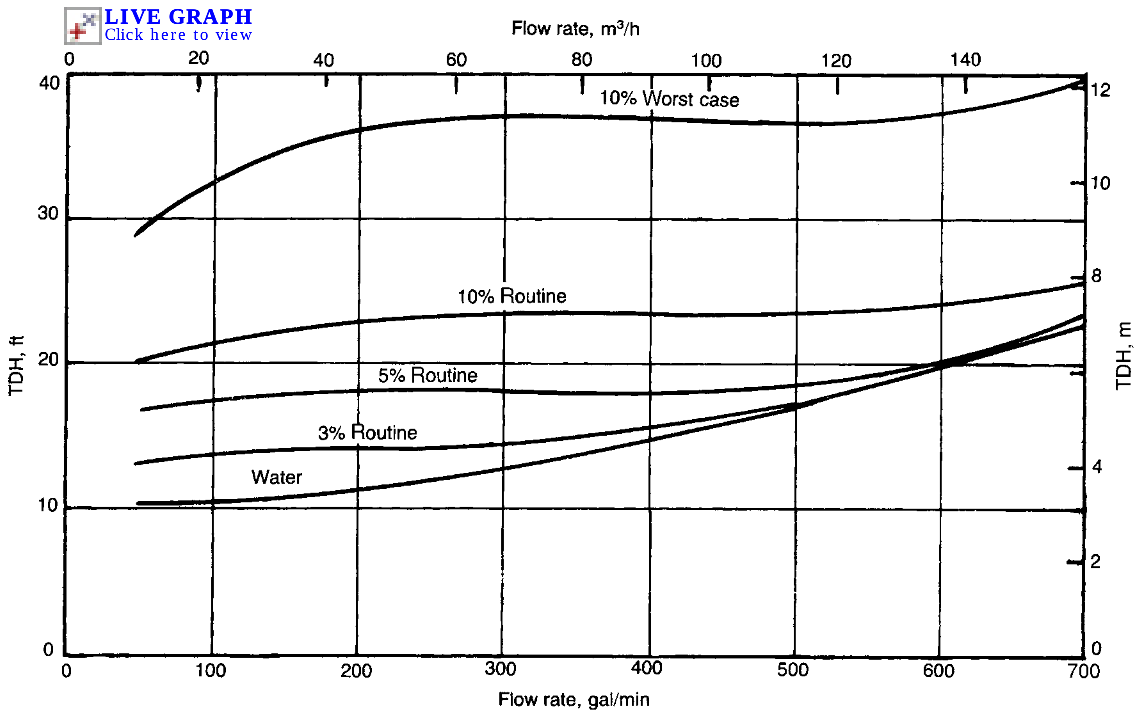


Figure 19-10. System H-Q curves for sludge in a 150-mm (6-in.) pipe.

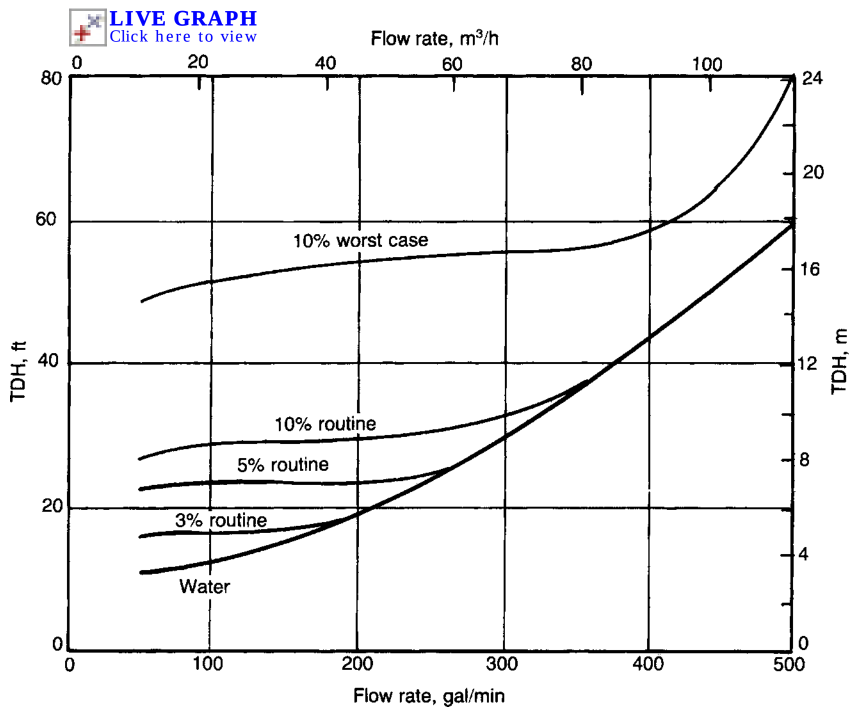


Figure 19-11. System H-Q curves for sludge in a 100-mm (4-in.) pipe.

Item	**SI Units**	**U.S. Customary Units**
Velocity	0.35 m/s	1.14 ft/s
Friction head, $C = 140$	0.91 m/1000 m	0.91 ft/1000 ft
Friction factor (Figure 19-5)	21	21
Friction loss	$0.91 \times 21 = 19.1\,\text{m}/1000\,\text{m}$	$0.91 \times 21 = 19.1\,\text{ft}/1000\,\text{ft}$
Pipe length	107 m	350 ft
Friction head	$(107/1000)19.1 = 2.0\,\text{m}$	$(350/1000)19.1 = 6.7\,\text{ft}$
Static head	3.2 m	10.5 ft
Velocity head	0.01 m	0.02 ft
TDH	5.21 m	17.22 ft

The curves of TDH for 150-mm (6-in.) pipe are shown in Figure 19-10 and those for the smaller pipe, in Figure 19-11. In comparing the curves of the two figures, the following are evident:

- In both sets of system curves the headlosses at high flow rates approach those of water. Note that some conservative designers would always provide some multiple of headloss above that of water. However, in this example such high flow rates are beyond those normally encountered.
- The 10% solids *worst-case* curve has a 50% safety factor over the *routine operations* curve. The safety factor approximates a deterioration of pipe from $C = 140$ to $C = 110$. If the designer believes that the pipe will deteriorate more than this or prefers an additional allowance for headlosses, then the system curves should be modified accordingly. Note that flexibility in pump drive speed may allow for future adjustments to higher head conditions.

Based on a review of the system curves, 150-mm (6-in.) piping is selected because of (1) lower TDH and horsepower requirements, (2) better size selection for maintenance and cleaning, (3) lower operating costs, and (4) future operational flexibility.

(2) Pump selection. A vortex pump is a good choice for this application. A 100-mm (4-in.) Model C WEMCO vortex pump is selected. Performance curves for the pump combined with the developed system head curves for 150-mm (6-in.) piping are shown in Figure 19-12.

A review of Figure 19-12 indicates:

- Speeds of about 400 to 800 rev/min are adequate for a wide range of operating conditions.
- A speed of 710 rev/min is required to pump 10% solids at worst-case conditions with low velocity. A speed of 800 rev/min gives better velocity.
- A speed of about 640 is required to pump 1% solids.

Solids, %	Speed, rev/min	Flow, m^3/h	TDH, m	Flow, gal/min	TDH, ft
10[a]	710	11.3	8.81	50	28.9
10[b]	600	11.3	6.13	50	20.1
5[b]	570	22.7	5.67	100	18.6
3[b]	525	37.9	4.27	167	14.0
1[b]	640	113.3	5.27	500	17.3

[a]Worst-case.

[b]Routine operation.

(3) Select the driver. Following the pump selection, a decision can now be made to limit the drive speed to desirable values. The motor horsepower is determined by the intersection of the water system curve with the maximum drive output speed. In this example, if the maximum drive output speed is chosen to be 900 rev/min, the motor horsepower must be 20. Select the best practical motor drive and pump speed combination based on costs, sizes, and equipment compatibility, and adjust the pump flow rates and times of operation to meet process transfer requirements.

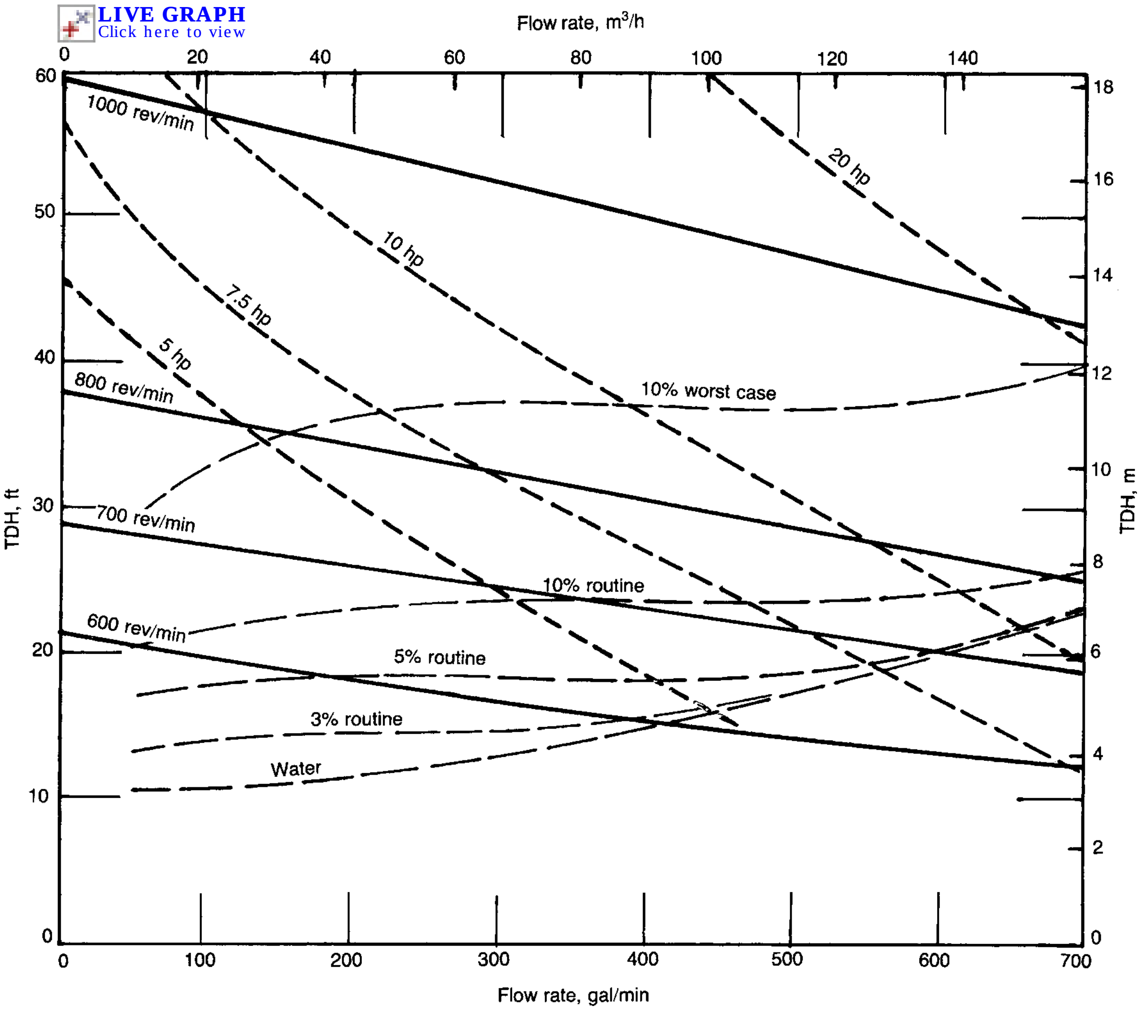


Figure 19-12. Performance curves for a 100-mm (4-in.) vortex pump and system curves for a 150-mm (6-in.) pipe. After WEMCO.

Possibilities for drives include:

- Constant speed at 900 rev/min, with the time period of pumping adjusted accordingly.
- Variable speed at 900 to 450 rev/min to allow either a slip drive or an AFD (see Section 15-11) with appropriate running time adjustments. (One method is the use of time clocks.)
- Variable speed at 900 to 450 rev/min, a turndown ratio of at least 2:1 to allow a variable-ratio belt drive to be used.

Without more knowledge of the owner's circumstances, there is no answer better than the one given here.

(4) Compare headloss by a different method. Always cross-check both the design method and the actual calculations when designing sludge systems, because an error can have disastrous consequences. For example, use Equations 19-3 and 19-4 to confirm that the flow will be laminar, then use Equation 19-1 to check the headloss calculated in Part (1). Consider a sludge with a specific gravity of 1.02, containing 5% solids, flowing at a rate of 22.7 m^3/h (100 gal/min) with a velocity of 0.35 m/s (1.14 ft/s) in a 150-mm (6-in.) pipe.

SI Units

Equation 19-3a

$$\nu_{lc} = \frac{1000_{\eta} + 1000\sqrt{\eta^2 + s_y \rho D^2/3000}}{D\rho}$$

$\eta = 0.015\,\text{kg/s} \cdot \text{m}$
$s_y = 4.8\,\text{Pa} = 4.8\,\text{N/m}^2$
$\rho = 1020\,\text{kg/m}^3$(specific gravity $= 1.02$)
$D = 0.15\,\text{m}$
$g = 9.81\,\text{m/s}^2$

$$v_{lc} = 1000 \times 0.015/(0.152 \times 1020) + 1000 \times \sqrt{\frac{(0.015)^2 + (4.8)(1020)(0.152)^2/3000}{(0.152)(1020)}}$$

$v_{lc} = 1.35\,\text{m/s}$

Equation 19-4a

$$v_{uc} = 1500 \times 0.015/(0.152 \times 1020) + 1500 \times \sqrt{\frac{(0.015)^2 + (4.8)(1020)(0.152)^2/4500}{(0.152)(1020)}}$$

$v_{uc} = 1.69\,\text{m/s}$

U.S. Customary Units

Equation 19-3b

$$\nu_{lc} = \frac{1000_{\eta} + 1000\sqrt{\eta^2 + s_y \gamma g D^2/3000}}{D_{\gamma}}$$

$\eta = 0.010\,\text{lb/s} \cdot \text{ft}$
$s_y = 0.10\,\text{lb/ft}^2$
$\gamma = 63.65\,\text{lb/ft}^3$(specific gravity $= 1.02$)
$D = 0.5\,\text{ft}$
$g = 32.2\,\text{ft/s}^2$

$$v_{lc} = 1000 \times 0.010/(0.5 \times 63.65) + 1000 \times \sqrt{\frac{(0.010)^2 + (0.10)(63.65)(32.2)(0.5)^2/3000}{(0.5)(63.65)}}$$

$v_{lc} = 4.43\,\text{ft/s}$

Equation 19-4b

$$v_{uc} = 1500 \times 0.010/(0.5 \times 63.65) + 1500 \times \frac{\sqrt{(0.010)^2 + (0.10)(63.65)(32.2)(0.5)^2/4500}}{(0.5)(63.65)}$$

$v_{uc} = 5.52\,\text{ft/s}$

Both lower and upper critical velocities are well above the actual velocity, so the Bingham equation for laminar flow is applicable.

SI Units

Equation 19-1a

$$\frac{H}{L} = \frac{16 s_y}{3D\rho g} + \frac{32\eta v}{\rho g D^2}$$

$$\frac{H}{L} = \frac{16(4.8)}{3(0.152)(1020)(9.81)} + \frac{32(0.015)(0.35)}{(1020)(9.81)(0.152)^2}$$

$$\frac{H}{L} = 0.0176 = 17.6\,\text{m}/1000\,\text{m}$$

From Part (1) the friction slope is

$$\frac{H}{L} = 19.1 \text{ (good check)}$$

U.S. Customary Units

Equation 19-lb

$$\frac{H}{L} = \frac{16 s_y}{3D\gamma} + \frac{32\eta v}{\gamma g D^2}$$

$$\frac{H}{L} = \frac{16(0.10)}{3(0.5)(63.65)} + \frac{32(0.010)(1.14)}{(63.65)(32.2)(0.5)^2}$$

$$\frac{H}{L} = 0.0175 = 17.5\,\text{ft}/1000\,\text{ft}$$

$$\frac{H}{L} = 19.1 \text{ (good check)}$$

Critique of Example 19-1

In the pump layout for the primary sludge pumps shown in Figure 19-8, there are two duty pumps with a common, identical standby pump (the one in the middle). This design allows a duty pump to be dedicated to each settling tank, allows separate process control for each tank, and gives the operator much more flexibility. The sludge flow characteristics are better with one pump for each tank than with one pump to serve two tanks, because there are no imbalances in friction head from each tank's suction piping to cause different sludge levels in the tanks.

The use of 150-mm (6-in.) piping requires confirmation by the manufacturer for applying the pump below the speeds shown on the catalog curve. This pump (4-in. Model C WEMCO) has been applied successfully for this particular example on the basis of eight years of operating experience at a speed of 300 to 350 rev/min [32]. Some engineers might favor the 100-mm (4-in.) piping because of better scouring due to greater fluid velocities, but objects such as toothbrushes or popsicle sticks can clog such small pipe unless grinders are used, so other engineers invariably avoid 100-mm (4-in.) pipe for sludge service. Note that operational velocities of 0.3 to 0.6 m/s (1 to 2 ft/s) occur in the 150-mm (6-in.) pipe at the expected solids concentrations of 3 to 5%. Such velocities are satisfactory in sludge pipelines if the grit content is not excessive. However, a variable-speed drive gives the operators the capability of pumping at greater velocities to clear the pipe. Adequate flushing water combined with the use of the plunger pump (see Figure 19-8 and Example 19-2) provide additional assurance for clearing obstructions in the 150-mm (6-in.) pipe. Hot water is helpful in flushing grease accumulations and is a desirable maintenance feature.

It is unlikely that primary sludge concentrations would reach 10%, although 7 or 8% may be attainable. However, the data for parameters s_y and η for undigested sludge are not extensive, so the use of parameters for 10% sludge is justifiably conservative.

If a more concentrated sludge at lower volumetric throughput is desirable, a different type of pump would probably be required. Options include: (1) a plunger pump, (2) a progressive cavity pump (if preceded by screening and grit removal), and (3) the combined screw-centrifugal impeller pump (such as a WEMCO Hidrostal®). An advantage of a positive-displacement-type pump is that process control measurements need not depend on flowmeters. Flowmeters may be unreliable and inaccurate in measuring sludge at low velocities.

The selection of a pump operating speed range of 450 to 900 rev/min allows operators to cope with varying sludge properties. Low pump speeds result in significantly less wear and reduced maintenance costs but require sturdier construction.

Design Procedure for Positive-Displacement Pumps

The design procedure for positive-displacement pumps is as follows:

1. Determine the maximum design pumping capacity required at the minimum sludge solids concentration.
2. Determine which combinations of simplex, duplex, and triplex pumps are feasible using the capacities found for each type of pump in a manufacturer's catalog. (Capacities vary from manufacturer to manufacturer.)
3. Plot the sludge system head curve for the maximum percentage of dry solids sludge concentration.
4. Calculate the actual pulsing flow for each of the pumps under consideration by using the percentages shown in Table 19-1.
5. From the system head curve, read the total head corresponding to the maximum pulsed flow for each type of pumping combination. Each type of pump specified must meet this total head requirement, which represents the most critical condition against which the plunger pump must operate.
6. Study manufacturers' publications to determine the motor horsepower required. Consult with the manufacturer to confirm the selection.
7. Select the type of pump most applicable with consideration for the following factors:

   - Because the single-cylinder, simplex pumps are the least balanced of the pumps, this type exerts the maximum loads on pump bearings and gears, and causes the highest friction in the pipe. These considerations are of the greatest concern at high heads.
   - When design conditions approach 690 kPa (100 lb/in.^{2}), it is preferable to select the smallest possible bore and multiple cylinders.

Example 19-2
Design of a Plunger Pump Installation

Problem: Design a plunger pump backup for the vortex pumps of Example 19-1. Assume 100-mm (4-in.) piping.
Solution: As a backup, the plunger pump would be used only when a pipeline is clogged or if all three vortex pumps were being repaired, so concern is limited to concentrations of solids higher than about 3% and, hence, to flows of approximately 11.3 to 37.7 m^3/h (approximately 50 to 167 gal/min).

Pump: Find the pump scheme that approximates the design flow. The capacities per plunger per number of cylinders for one brand of pump are as follows.

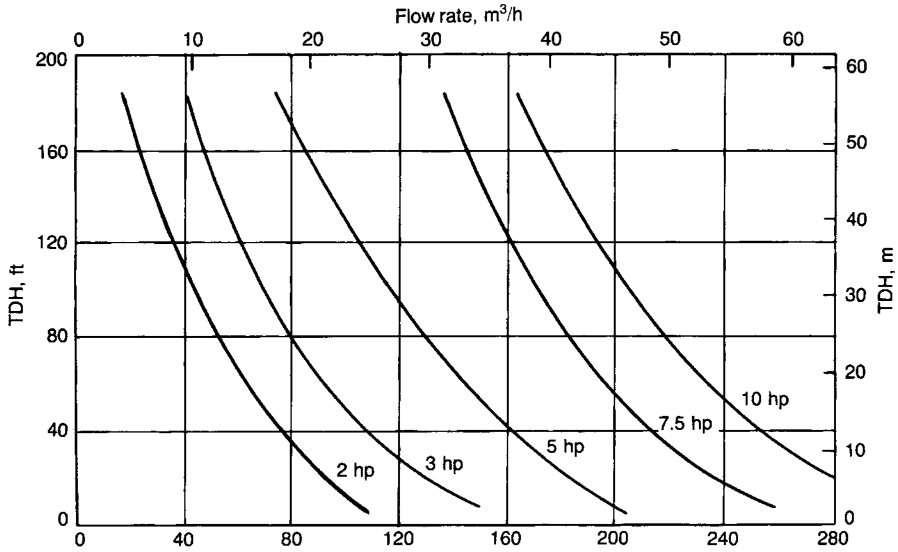
Pump scheme	Average rated capacity for various cylinder diameters			
	225 mm	275 mm	9 in.	11 in.
1 simplex	19.3 m^3/h	28.4 m^3/h	85 gal/min	125 gal/min
2 simplexes	38.6 m^3/h	56.8 m^3/h	170 gal/min	250 gal/min
1 duplex	34.1 m^3/h	49.9 m^3/h	150 gal/min	220 gal/min
1 triplex	52.2 m^3/h	68.1 m^3/h	230 gal/min	300 gal/min

One duplex pump with the larger cylinder size is adequate.

Variable-speed drive. Variable speed gives better control over pumping conditions than does constant speed. At high sludge concentrations, the high flow rates delivered by a constant-speed pump may result in "keyholing" in which the sludge cannot enter the suction pipe at high rates. Power is based on the head at pulse flow rates. The ratio of pulse to average flow rate for a duplex pump is 1.60 (see Table 19-1). Find the head corresponding to the pulse flow rate from the curves in Figure 19-11.

Curve in Figure 19-10	SI Units			U.S. Customary Units		
	Average flow, m^3/h	Pulse flow, m^3/h	Pulse TDH, m	Average flow, gal/min	Pulse flow, gal/min	Pulse TDH, ft
10% worst case	11.3	18.2	14.3	50	80	47
5%	22.6	36.3	7.3	100	160	24
3%	37.7	60.6	7.6	167	267	25

In Figure 19-13, find the intersection of pulse TDH and average flow. For 3% sludge, the shaft output is about 5 hp (by interpolation). The speeds should vary about 167:50 or 3:1, and the maximum speed is 50 rev/min. A variable-ratio belt drive (see Table 15-2 in Section 15-11) is the recommended choice. Consult with the manufacturer to determine the efficiency and, thus, the size of motor required. Note that some manufacturers' curves are based on a "nonpulsating" flow and corresponding head due to probable inclusion of air chambers. Be cautious; use experience and judgment to interpret the basis of manufacturers' curves and design accordingly.



LIVE GRAPH
Click here to view

Figure 19-13. Performance curves for a duplex, heavy-duty, dual-valve, 275-mm (11-in.) plunger pump with a maximum speed of 50 rev/min and a 3-m (10-ft) suction lift (for pulse TDH and average flow rate). After Komline–Sanderson Engineering Corp.

Critique of Example 19-2

There are many other choices that might be made, because the operating conditions for a backup positive-displacement pump are a matter of judgment. For example, smaller pumps can be operated for longer times. Typically, the primary objective (pumping scum) determines the sizing of the plunger pump, and then the size is double-checked against the pump's standby function.

19-4. Piping System Design

Due to the nature of sludge, the requirements of sludge piping systems differ in many ways from those of water piping systems. The following discussion relates specifically to sludge as an addendum to the general design practices for water and wastewater presented elsewhere in this text.

Pipe Materials

The most common pipe materials for sludge piping systems are ductile iron and steel; regional preferences are toward ductile iron in the eastern United States and steel in the western United States. The choice depends primarily on costs and availability because the performance of either, when properly protected from corrosion, is satisfactory.

Designers differ on the minimum allowable size of pipe. Some limit sludge piping to a minimum of 150 mm (6 in.), and some require no less than 200 mm (8 in.). Some do not object to 100 mm (4 in.) with glass or cement-mortar lining. Pipe used in sludge piping systems should be lined to resist corrosion and/or abrasion or to resist grease buildup. Grease is a significant concern with scum and primary sludges, and a smooth surface in piping for digested sludge inhibits the formation of struvite crystals [2]. The most common lining for ductile iron and steel pipe is cement mortar, but an even smoother lining, such as glass, is desirable. The economics and long-term reliability of glass-lined pipe should be evaluated carefully. Any piping system used for sludge service should have take-down couplings at not greater than 10-ft intervals to facilitate cleaning operations. It may be possible to achieve the desired goal of reducing maintenance on sludge pipelines to the same level that is projected for glass-lined pipe by judiciously selecting the allowable velocities, eliminating bends and pipe constrictions or "bottlenecks," and providing adequate flushing and cleaning facilities in pipes lined with cement mortar. The use of polyethylene-lined ductile iron pipe is a viable, economic alternative to glass-lined pipe. If the grit content of the sludge is low, some plastic pipes exhibit good resistance to wear and maintain clean internal surfaces.

Valves

The discussion in Chapter 5 should be modified by the following general comments and tempered with the designer's experience. Note that any valve should be both accessible and operable from the pump room floor. If pigs or other cleaning devices must pass through an open valve, select a ball, cone, or eccentric plug valve, preferably with a full, round port to favor clog-free operation.

Eccentric Plug Valves

One of the most common types of valves, the non-lubricated, eccentric plug valve, is usually specified to have a port area of at least 80% of the full pipe area to ensure clog-free operation. These valves can be obtained with a full, round port opening, which is advantageous if the line is to be rodded or pigged. However, some pigs can pass through square ports.

Typically, valves smaller than 200 mm (8 in.) are equipped with 50-mm (2-in.) nuts for wrench operation, whereas larger valves have worm gears and hand wheels. Synthetic rubber covering is usually specified for eccentric plug valves to avoid wear of and damage to the seating surface. Eccentric plug valves that provide tight shut-off with pressure in either direction (a distinct advantage in sludge pipelines with multiple flow routings) are offered by at least one major manufacturer.

Ball Valves

Consider full-bore ball valves instead of plug valves on applications where a stoppage could require a major process tank (such as a digestion tank) to be dewatered to service the valve. In general, three-way valves are not used routinely because they usually have smaller openings and increase the likelihood of clogging. Also, if the valve fails both piping routes are out of service until the problem is corrected. Ball valves are sometimes preferred on corrosive service because the relatively small body size and weight mean less of an expensive alloy. Some ball valves are commonly made of thermoplastic with tension seats [33].

Check Valves

Check valves add to the design problem on sludge systems if they are installed on a vertical pipe because solids pack on the discharge side. Install check valves (if used at all) on horizontal pipes.

Ball Check Valves

Ball check valves are used in (and recommended for) plunger and diaphragm pumps (see also Section 5-4). The balls, made of lead-impregnated synthetic rubber, are contained in a chamber with a quick access door to facilitate the frequent replacement required.

Stringy materials tend to prevent proper seating of the balls, so plunger pumps are commonly supplied with a pair of check valves in tandem on both suction and discharge sides of the pump (see Figure 11-31). Diaphragm pumps can also be obtained with pairs of ball check valves. The reliability of a plunger or diaphragm pump can be improved by installing a grinder or macerator ahead of the pump. If grinders are placed too far upstream, they are likely to be less effective because separated and ground stringy material tends to become reconstituted.

Pinch Valves

Pinch valves are also widely used in sludge service, especially as relief valves where there is a need to protect a positive-displacement pump (or other downstream equipment) from an inadvertent valve closure in the discharge line. The system consists of an air-loaded pinch valve at a pipe tee that discharges fluid back to the pump suction source. A pinch valve system is relatively maintenance-free, but it does require an air source and a pressure regulator.

Cone Valves

Cone valves are designed for throttling control and have been used successfully on water and raw sewage lines to regulate valve closures for elimination of water hammer effects. Cone valves also replace both the pump discharge check and isolation valves. None of these aspects, however, routinely applies to sludge service. Apparently, cone valves are not used in sludge systems and one manufacturer's representative stated that the valve is not intended for such service [34].

Gate Valves

Gate valves are seldom used today in sludge service because of their tendency to collect debris and solids as compared to the preferable plug valve [27].

Pump Seal Systems

Most sludge pumps are specified with seal water systems rather than mechanical seals due to the nature of the pump service. The seal water flow requirement should be reviewed with the pump manufacturer. Seal water is taken from a water system but never from the pumped fluid discharge. Potable water must be separated completely from seal water by an air gap in the seal water tank or an acceptable backflow prevention system.

Seal water lines should have a pressure-reducing valve on the seal water header supplying seal water to a series of pumps. The valve prevents the seal water from entering the pump casing at an excessive pressure that could dislodge the packing and accelerate packing and shaft sleeve wear. Seal water leaving the pump should be conveyed by piping to the floor drain system and never be allowed to overflow the pump base or pump room floor. Pump bases should be curbed with drain outlets. A typical water seal system schematic is shown in Figure 19-14 and a suggested pump drain system is shown in Figure 19-15.

Flowmeters

The nature of sludge makes it necessary to use flowmeters that do not obstruct the flow. Sludge metering systems are expensive, often inaccurate, and always require preventive and corrective maintenance. Unless positive-displacement pumps are used, meters are nevertheless necessary for measuring the flow of sludges. It is also desirable to measure solids content continuously so that the operators can determine the mass of (dry) solids transferred to track process control. If some means of sampling solids concentrations is not included, the flow data will only track volumes of fluids and not mass of (dry) solids.

Venturi, magnetic, ultrasonic, or in-line density flowmeters can be used. The first three are typically preferable to the fourth due to pipeline obstruction considerations. One advantage of a positive-displacement pump is that the pump itself can be used as a flowmeter if the number of revolutions or strokes over a specific time period is recorded.

A concern about meters in sludge pipelines is pipeline cleaning. Flowmeters should be installed in the piping system with a bypass for routine maintenance, calibration, and (if needed) a water purging system (see Figure 20-12). A pipe spool piece should be available to replace the meter when it is removed for servicing or when the line must be pigged.

The relative accuracy of each type of sludge meter is debatable. Many engineers believe the Venturi system is the best overall choice when accuracy, maintenance, and long-term reliability are considered. Others prefer magnetic or ultrasonic meters.

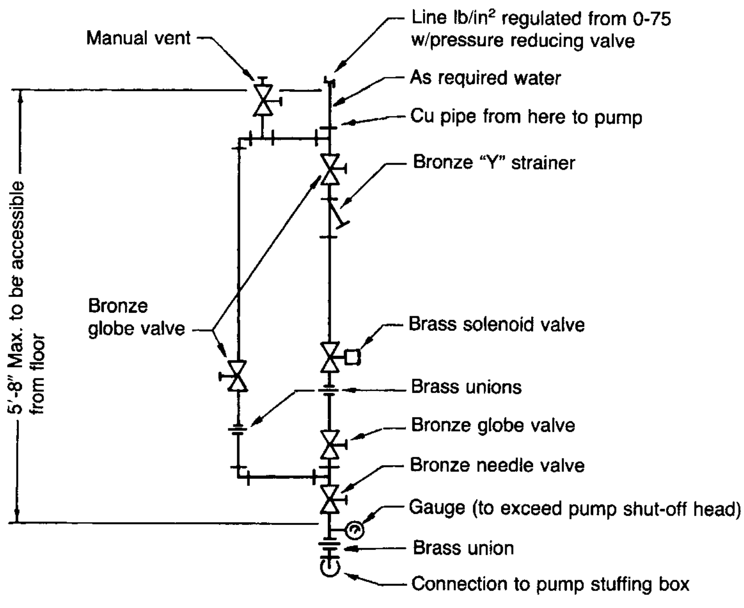


Figure 19-14. A typical water seal connection. Courtesy of Stearns & Wheeler Engineers and Scientists and J. Kenneth Fraser & Associates.

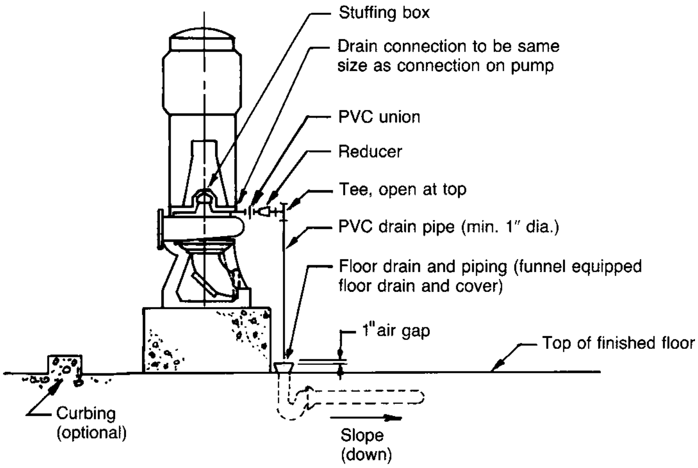


Figure 19-15. Pump drain system. This arrangement is also applicable to horizontal pumps. Courtesy of Stearns & Wheeler Engineers and Scientists and J. Kenneth Fraser & Associates.

Grinders

Grinders or macerators are sometimes provided in sludge piping systems for shredding solids, rags, and other debris to reduce wear on downstream pumps and sludge handling or processing equipment.

When an upstream sludge grinder is used to protect the pump, add an interlock between the sludge grinder and the pump to keep the grinder operating whenever the pump is running. However, a manual bypass should be installed around the grinder so it can be repaired.

Pipe Flushing and Draining

Flushing sludge piping systems is a common maintenance chore in water and wastewater treatment plants. Flushing requires an adequate velocity and pressure of water to dislodge solids, move obstructions, and generally clean the pipe. Flushing connections should be liberally provided to allow convenient operations. Flushing water should be available to the piping system on both the suction and discharge sides of sludge pumps.

Pipe Cleaning Stations

A well-designed system for pipe cleaning allows the piping system to be maintained in mint condition and reduces the pumping power requirement. Two systems used for cleaning pipe are (1) rodding out each straight section of piping, and (2) cleaning the entire pipe with pipe-cleaning pigs.

Rodding

Rodding requires specifying cleanouts on all elbows, tees, and crosses within the pumping station. Require pipe-sized cleanouts for diameters up to 100 mm (4 in.) and cleanouts one-half the pipe size on larger pipe at bosses on all bends or other angled fittings. Additionally (or alternatively), bends can be replaced by crosses with removable blind flanges for cleanout or rodding. All cleanout plugs should be fitted with 25-mm (1-in.) or larger plug valves with hose adapters. The sludge piping can then first be drained or cleared by high-pressure water in preparation for rodding. Cleanouts for rodding must be located no farther than 15 m (50 ft) apart unless special rodding tools are available.

Pigging

The main advantage of cleaning pipe with pigs is that the pipe length is practically unlimited. Cleaning pigs range from soft swabs to flexible, abrasive pigs to rigid steel scraper pigs. Block ice or bagged ice cubes can also be used; with this method, retrieval is unnecessary and permanent jamming cannot occur. Flexible pipe-cleaning pigs can clean several pipe sizes, but are limited in turning around pipe crosses. Transmitter-cleaning pigs enable the maintenance crew to locate underground pipe failures accurately.

Cleaning the piping with pigs requires provisions for the insertion and retrieval of the pigs. The pigs are

inserted in an isolated length of pipe one size larger than the piping to be cleaned and located as close to the primary pumps as possible. The pig launcher is fitted with a pressure gauge, a drain, a high-pressure water fitting, and an easily removed blind flange or mechanical coupling. The pump discharge and the pig launcher pressure gauges should be easily read from one point. Ideally, this control point is protected from a pipe rupture and includes other controls as necessary. The pressure gauges last longer if they are filled with glycerin and protected by an isolation diaphragm and a snubber (glycerine-filled capillary tubes) to keep sludge out of the gauge and attenuate sharp pressure spikes. Pig launching and retrieval stations are shown in Figures 4-22 and 4-23. The details of cleaning pipe with pigs are discussed by Playford in the *Proceedings* [14, pp. 878–879].

The ease of cleaning pipes with pigs and the increasing cost of power warrants the installation of pipe-cleaning, launching, and receiving stations on most pipe systems and networks.

Safety Factors for Sizing

Sludge piping systems should be as short as possible with a minimum number of bends. Bends (unless replaced by tees or crosses) should be the long-radius type to minimize the headloss. In addition, the headloss contributed by check valves, flowmeters, and other piping accessories should be considered critically in the analysis of the system. If at all possible, omit check valves and fittings that would cause plugging of solids or add unnecessary headloss.

In each sludge pumping system, at least two pumps sized at full capacity should be provided so that at least one full-capacity standby pump is available. It may well be appropriate to use a positive-displacement standby pump because of its reliability and ability to move heavy sludge.

19-5. Long-Distance Pumping

Pumping wastewater sludges through pipelines 1.6 km (1 mi) long (or longer) requires greater attention to issues that are not as critical in pipelines of shorter length. Small variations in the unit dynamic headloss are magnified by the length of piping in a long pipeline. Dynamic headlosses are significantly affected by:

- Concentration of the sludge
- Type of sludge (raw, secondary, or digested)
- Variability of blends if a mixture of sludge is pumped
- Additives used in the wastewater or sludge treatment process (polymers or metal salts).

Other design and operating considerations for long sludge pipelines include:

- Periodic venting of gases released from solution or generated by the digestion process
- Correct sizing of the pipeline to allow for large flow rate variation over the system lifetime
- Sludge storage or a backup pipeline to enhance reliability and allow for pipeline maintenance
- Selection of pumps that can perform satisfactorily over a wide range of discharge pressures as concentrations and sludge blends change.

The designer of a long sludge pipeline must give serious consideration to measuring the characteristics of the range of sludge blends, solids concentrations, chemical additives, and sludge conditions (including temperature) likely to be encountered. The need for site-specific sludge testing is highlighted by the variability of the published data [13, 18] on sludge characteristics—particularly when major investments are to be made in pumping transport and storage systems. Extensive testing is even more necessary when undigested secondary sludges are to be pumped. These complex problems are covered in more detail by Carthew et al. [18].

The use of specially designed diaphragm pumps capable of large flow rates at pressures up to 2400 kPa (350 lb/in.^{2}) has been successful. The application of such pumps is unusual, but several advantages include (1) low maintenance, (2) high reliability, (3) excellent protection against overpressuring the pipeline, (4) clean, quiet operation, and (5) separation of sludge from most of the pump. Diaphragm pumps not specifically manufactured for sludge or slurry service do not have these advantages.

19-6. References

1. Metcalf, L., and H. P. Eddy, *American Sewage Practices. Volume III: Disposal of Sewage*, 3rd ed., McGraw-Hill, New York (1935).
2. EPA Process Design Manual for Sludge Treatment and Disposal, EPA 625/1-79-011, U.S. Environmental Protection Agency, Municipal Environmental Research Laboratory, Office of Research and Development, Cincinnati, OH, (September 1979), pp. 14-4–14-17, 14-29–14-37.
3. Babbitt, H. E., and D. H. Caldwell, "Laminar flow of sludges in pipes with special reference to sewage

sludge," University of Illinois Engineering Experiment Station, *Bulletin Series, No. 319* (1939).
4. Caldwell, D. H., and H. E. Babbitt, "The flow of muds, sludges, and suspensions in circular pipe," *Trans. of the American Institute of Chemical Engineers,* 37, 237 (April 1941).
5. Wolfs, J. R., "Factors affecting sludge force mains," *Sewage and Industrial Wastes,* 22, 1 (January 1950).
6. Babbitt, H. E., and D. H. Caldwell, "Turbulent flow of sludges in pipes," University of Illinois Engineering Experiment Station, *Bulletin Series, No. 323* (1940).
7. Brisbin, S. G., "Flow of concentrated raw sewage sludges in pipes," *Journal of the Sanitary Engineering Division American Society of Civil Engineers,* 83(SA3), 1274, (June 1957).
8. Chou, T. L., "Flow of concentrated raw sewage sludges in pipes," *Journal of the Sanitary Engineering Division American Society of Civil Engineers,* 84(SA1), 1557 (February 1958).
9. Sparr, A. E., "Pumping sludge long distances," *Journal of the Water Pollution Control Federation,* 43, 1702 (August 1971).
10. Dick, R. L., and B. B. Ewing, "The rheology of activated sludge," *Journal of the Water Pollution Control Federation,* 39, 543 (1967).
11. Bourke, J. D., "Sludge handling characteristics in piped systems," *Proceedings of the Northern Regional Conference of the California Water Pollution Control Association,* Monterey, CA (October 19–20, 1973).
12. Vesilind, P. A., *Treatment and Disposal of Wastewater Sludges,* 2nd ed., Chapter 4, Ann Arbor Science Publishers, Ann Arbor, MI (1979).
13. Mulbarger, M. C., et al., "Pipeline friction losses for wastewater sludges," *Journal of the Water Pollution Control Federation,* 53, 8, 1303–1313 (August 1981).
14. Sanks, R. L., C. W. Reh, and A. Amirtharajah (Eds.), *Conference Proceedings, Pumping Station Design for the Practicing Engineer,* Montana State University, Bozeman, MT (1981), pp. 854–899. Available through interlibrary loan.
15. Govier, G. W., and K. Aziz, *The Flow of Complex Mixtures in Pipes,* Van Nostrand Reinhold, New York (1972).
16. Cain, C. B., et al., "Design of 90-mgd wastewater reclamation plant," *Journal of the Environmental Engineering Division American Society of Civil Engineers,* 107(EEI), 29 (1981).
17. Rimkus, R. R., and R. W. Heil, "The rheology of plastic sewage sludge," in L. K. Cecil, Ed., *Second National Conference on Complete Water Reuse, American Institute of Chemical Engineers,* Chicago, IL (May 4–8, 1975).
18. Carthew, G. A., C. A. Goehring, and J. E. van Teylingen, "Development of dynamic headloss criteria for raw sludge pumping," *Journal of the Water Pollution Control Federation,* 55, 472–483 (May 1983).
19. Hanks, R. W., and B. H. Dadia, "Theoretical analysis of the turbulent flow of non-Newtonian slurries in pipes," *Journal of the American Institute of Chemical Engineers,* 17, 554 (May 1971).
20. Hanks, R. W., "Principles of slurry pipeline hydraulics," in N. P. Cheremisinoff (Ed.), *Encyclopedia of Fluid Mechanics, Vol. 5, Slurry Flow Technology,* Gulf Publishing, Houston (1986).
21. Darby, R., "Laminar and turbulent pipe flows of non-Newtonian fluids," in N. P. Cheremisinoff (Ed.), *Encyclopedia of Fluid Mechanics, Vol. 7, Rheology and Non-Newtonian Flows,* Gulf Publishing, Houston (1988).
22. *Pipeline Design for Water and Wastewater,* American Society of Civil Engineers, Report of the Task Committee on Engineering Practice in the Design of Pipelines, New York (1975).
23. Karassik, I. J., W. C. Krutzsch, W. H. Fraser, and J. P. Messina, *Pump Handbook,* 2nd ed., McGraw-Hill, New York (1986), Chapters 2 and 9.
24. Henshaw, T. L., *Reciprocating Pumps,* Van Nostrand Reinhold, New York (1987), Chapter 6.
25. Baumeister, T. (Ed.), *Standard Handbook for Mechanical Engineers,* 8th ed., McGraw-Hill, New York (1978), Chapter 14.
26. Wylie, E. B., and V. L. Streeter, *Fluid Transients,* McGraw-Hill, New York (1978), Chapter 14.
27. Water Pollution Control Federation and the American Society of Civil Engineers, *Wastewater Treatment Plant Design,* WPCF Manual of Practice No. 8. Lancaster Press, Lancaster, PA (1977).
28. Burrage, K., President, Penn Valley Pump Co. Private communication (September 1997).
29. Jones, G. M., Vice President, Brown and Caldwell Consultants. Private communication (February 1989).
30. Svaras, J. C., Application Engineer, Alfa Laval Pumps, Inc. Private communication (September 1997).
31. Caballero, R. C., County Sanitation District of Los Angeles. Private communication (undated; tested March 1983).
32. Cunningham, J., Chief Operator, Clinton Wastewater Treatment Plant. Private communication (January 1989).
33. O'Keefe, W., "Valves for Corrosives and Slurries," *Power,* Special Report (May 1986), pp. 51–51.
34. Allis-Chalmers Valve Division. Private communication (January 1989).

Chapter 20

Instrumentation and Control Devices

ROBERT S. BENFELL
ALAN VAUSE

CONTRIBUTORS
Russell H. Babcock
Mead Bradner
Harold D. Gilman
Mayo Gottliebson
Paul C. Leach
John Leak
Gary Skipper
Earle C. Smith
James Taube
Horton Wasserman

The purpose of this chapter is to introduce the basic instrumentation and control devices that are needed and used in pumping stations. The information presented should be of value to (1) designers, who must plan for and specify equipment, (2) project engineers, who must decide how a pumping station will operate and what kind of equipment will be needed, and (3) owners, who must maintain the equipment selected. The types of instruments and their principles of operation are discussed in detail by Lipták [1]. Specific control devices and their advantages can be found in manufacturers' literature. However, what is needed (and presented in this chapter) is a careful analysis of the *relative* merits of instruments specifically for use in pumping stations.

20-1. Reliability

Instruments for pumping stations should be selected to provide long life, low maintenance, and high reliability in damp, corrosive environments. These requirements are not necessarily unique to pumping stations, because many industrial applications have similar requirements. But industrial users are frequently less concerned with extended life, and most such users maintain highly skilled, full-time maintenance staffs to service sophisticated equipment. Pumping station operators often have limited sophisticated maintenance skills and usually rely on outside maintenance services. Furthermore, most pumping stations are designed to operate for 20 years or more without major reconstruction. Accordingly, the simplest instrument that will perform the required function should be selected for pumping stations, and premium materials, such as stainless steel, should be used wherever possible to provide maximum resistance to environmental conditions.

Because many pumping stations are operated unattended and the potential property damage is significant if a serious malfunction occurs, instruments should be selected for their inherent reliability.

Back-up systems should be provided, and the consequences of component failure must be carefully considered.

Instrument design is a highly specialized field that requires knowledge of both instrumentation equipment and fluid mechanics. Any engineer who designs instrumentation and control systems must carefully consider the application limitations of the systems to be specified and should seek the services of qualified specialists if unusual and/or complex systems are required.

20-2. Instrument Selection

There are many instruments and instrument systems on the market that were designed specifically for water and wastewater facilities. But only some of this equipment was designed with serious consideration given to the requirements of such facilities. Unfortunately, much of it was designed with low cost as the primary consideration to compete in a market in which much equipment (1) is bought from the "low bidder" and (2) is probably inferior to equipment that was designed primarily for the industrial marketplace. Every instrument or functional instrument system must be evaluated solely on the basis of its suitability for the application. Instruments and instrument systems designed for the water and wastewater marketplace are not inherently better or worse than instruments and instrument systems designed for industrial applications for service in pumping stations.

Environmental Conditions

The environmental conditions to which the various instruments in a pumping station are subjected vary considerably from station to station and from location to location within a pumping station. Most pumping stations have areas of high humidity, many have areas of extreme temperature, and most wastewater pumping stations have areas that are corrosive or classified as hazardous by the NEC [2]. Instruments that are resistant to these conditions are generally available. However, instrument life, reliability, and/or maintainability are adversely affected by service under such conditions. Effects of corrosive gasses and high humidity can be reduced or eliminated by adequate ventilation or by removing as much of the instrumentation equipment as possible to more hospitable areas of the station. Electrical instruments in hazardous atmospheres can be put into explosion-proof cases, but pneumatic instruments and intrinsically safe electronic instruments are easier to maintain.

Reliability and Maintainability

In many instruments, the "motion-balance" principle is used for operation. These instruments have springs, racks and pinions, and bearings that are all subject to wear. A simplified motion-balance Bourdon-tube mechanism used for pressure measurement is shown in Figure 20-1. An increase in pressure within the Bourdon tube straightens it slightly, which lifts the linear voltage differential transformer (LVDT) core. The primary coil is excited with a high-frequency alternating current, and the secondary coils are connected in series opposition so that the two voltages produced are of opposite phase. As the core moves up, the magnetic coupling between the lower secondary coil and the primary coil decreases and the magnetic coupling between the upper secondary coil and

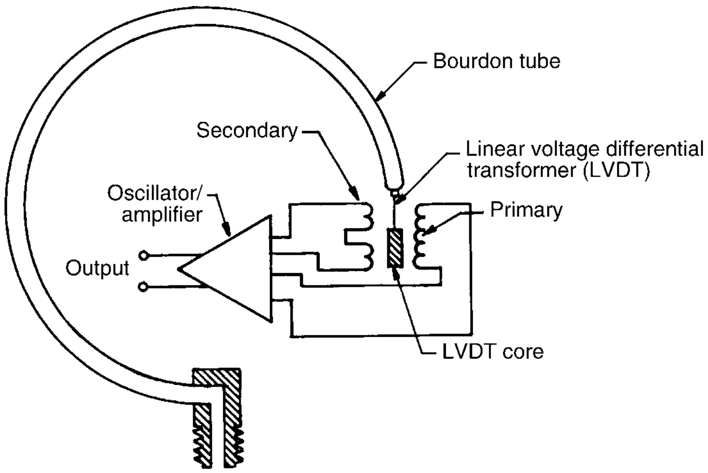


Figure 20-1. Motion-balance mechanism.

the primary coil increases. Suitable phase-sensitive electronics are used to produce a direct current output proportional to core position.

"Force-balance" instruments reduce the amount of movement and, therefore, reduce wear. A simplified force-balance mechanism used for pressure measurement is shown in Figure 20-2. An increase in pressure within the housing displaces the diaphragm to the left, which rotates the force beam clockwise about the fulcrum and unbalances the null detector. The amplifier detects this unbalance and increases its output, which tends to rotate the force beam counterclockwise until balance is again established. The current passed through the solenoid force motor is also the output current, which is typically 4 to 20 mA. By designing the solenoid for a linear response and using a high gain null detector and amplifier, the motion required to produce a signal is reduced to a very small value. The only parts actually strained are the diaphragm and the seal at the fulcrum. The stiffness of these parts is made insignificant, which results in a very stable pressure-measuring device.

Many modern instruments utilize solid-state circuitry with no moving parts. Ordinarily, force-balance and solid-state instruments are more reliable and require less maintenance than motion-balance instruments. However, motion-balance instrument mechanisms are more easily understood by maintenance personnel familiar with mechanical equipment. Consider who is going to maintain the instruments before selecting exotic modern equipment that may require highly skilled specialists for routine servicing.

Utilities

Many instruments require either electrical power or instrument air for their operation, and such facilities constitute a major cost of instrumentation systems. Electric power must be (1) conditioned to remove surges, which may interfere with proper instrument operation or even damage sensitive electronic circuitry, (2) converted to the required voltage levels, and (3) regulated to ensure stable operation. In many pumping stations, provision must be made to ensure that electric power for instruments is maintained during service power outages.

Pneumatic instruments require clean, dry air except for bubbler systems connected to pressure switches. If the instrument air supply is not of adequate quality, instrument reliability suffers and maintenance requirements increase very significantly. Because the cost of purchasing and maintaining adequate instrument air equipment is significant, electronic instrument systems are most often selected for modern systems. However, pneumatic instruments have significant

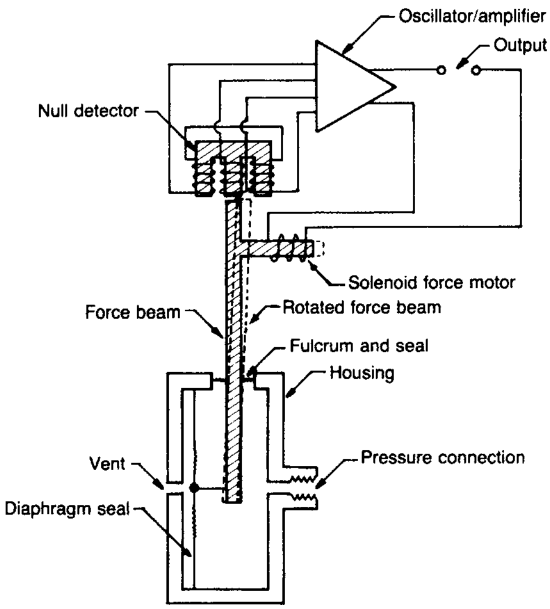


Figure 20-2. Force-balance mechanism.

advantages in high temperature, corrosive, and hazardous locations. Also, level measurements are more reliably made with pneumatic bubbler systems than with electronic systems. In many pumping stations, particularly large wastewater stations, pneumatic instruments can be used advantageously if the maintenance workers are familiar with this type of equipment.

It is true that explosion-proof cases are available for field instruments, but there is no way to service or calibrate them without opening the case, which makes them "nonexplosion-proof." Intrinsically safe circuitry sounds easy but is difficult to accomplish. In warm climates, pumping stations are often outdoors and sunlight can heat the cases to temperatures that electronic devices cannot withstand. Chlorine and hydrogen sulfide can quickly destroy electronic circuitry.

The choice of electrical or pneumatic equipment may depend on the skills of the service personnel. A skilled technician can disassemble, clean, and reassemble pneumatic equipment quickly (often in much less than an hour) in the field. Electronic equipment usually has to be removed and returned to the factory.

Accuracy

Accuracy requirements for instrument systems are generally not severe. The exception to this rule is "custody transfer" flow measurements (situations in which money passes from one entity to another based on flow quantities) that occur in water or wastewater applications. Instrument systems usually consist of a number of interdependent elements, each of which contributes to the total measurement error—errors that are always greater than the ones published in manufacturers' literature. A major source of error is the piping configuration at the point of measurement. It seldom resembles the piping in the hydraulics laboratory that was originally used to establish instrument performance.

The error of many instruments is stated as a percentage of span. Most instruments ordinarily operate at a point considerably below the maximum calibrated range. If high accuracy (such as in custody transfer) is required, extreme care must be taken to be sure that the instruments are properly applied and installed and that unnecessary error is not introduced by less than optimum configurations of multiple instruments in a system. Where instrument systems are used for custody transfer, it must be recognized that a certain amount of error is inevitable, and contracts must be written to recognize and accept the error.

Package Systems

Package systems are available to perform many instrumentation functions commonly used in pumping stations. Common examples include pump sequence control and chlorination control. The obvious advantage of a package is that it can be applied by a designer who is not skilled in instrumentation system design.

The following are disadvantages of package systems:

- It may be impossible to find one that does exactly what is desired, which may lead to compromises in control strategy.
- Packages may be difficult or even impossible to modify if the control strategy needs to be changed.
- Nonstandard signals between elements are frequently used, so all replacement parts must be obtained from the original vendor, who may discontinue the model or even go out of business.
- Packages tend to be designed so that only the original vendor can service them.
- Many are designed without adequate consideration of the environment in a pumping station or the need for reliability and long life.

This is not to say that packages should never be used. They frequently provide cost-effective solutions, particularly in relatively simple applications. But take the same care in selecting packages as in selecting components for custom-designed instrument systems.

Measurement Types

There are two basic types of measurement: analog and discrete. Analog measurements produce a signal or readout proportional to the value of a certain parameter, such as flow or level. Discrete measurements produce a true/false indication, such as valve "open/closed" or pump "running/stopped."

Signal Transmission

Many instruments provide only a local indication of a measurement. A pressure gauge is a common example. But, frequently, signal transmission to another location is required. Wherever possible, use standard signal transmission systems, such as 4 to 20 mA for electrical systems and 20 to 100 kPa (3 to 15 $\mathrm{lb/in.}^2$) for pneumatic systems. These systems allow unrestricted interconnection of instruments of

different manufacturers. Many nonstandard systems have been used over the years in water and wastewater facilities. Some are well established and conversion devices are readily available to incorporate such signals into standard systems, but such conversion devices (1) add more equipment to maintain, (2) reduce reliability, and (3) increase system error.

Variables Measured

Variables or parameters frequently measured or sensed are listed in Table 20-1. Not all of them would be used in any but very large stations. The purpose of a pumping station is to pump—not to collect data. If there is not a well-defined use and need for data, do not collect them. If a device is not essential, omit it.

20-3. Level Measurements

Level is one of the most common measurements made in pumping stations. Instrument types frequently used for this measurement are summarized in Table 20-2 more or less in order of popularity.

Floats

A float is a buoyant body that rides on the surface of a liquid and includes a method of monitoring its float elevation. A float may be suspended from a rod or a cable that mechanically links it to an elevation monitoring device, or the float may contain a magnet and simply slide up and down on a rod containing reed switches that are magnetically actuated. Another type of float, sometimes called a "tethered float," consists of a spheroidal, hollow vessel with a switch inside. When lifted (or buoyed), the float changes position and actuates the switch. Tethered floats (Figure 20-3) are frequently used in wastewater applications with some degree of success. Floats have these disadvantages:

- They are difficult to adjust.
- Access to the wet well is required for servicing.
- The electric cable flexes and wears out, so the float must be replaced periodically (2 to 3 years).

Float devices, which are available in a wide variety of materials, are simple and inexpensive. They are not well suited to analog transmission applications because of the complications involved in converting the motion produced to a transmission signal, but

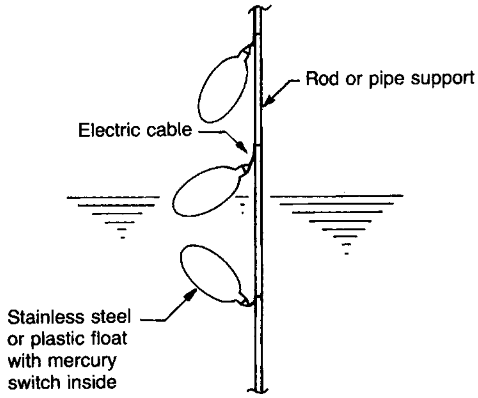


Figure 20-3. Tethered floats.

they are good for discrete applications and local indication applications. Because the presence of floating debris in wastewater may foul the floats, use only large floats and call for periodic washdown in the O&M manual in such service.

Although popular, tethered floats have sometimes given trouble and they do not carry a UL label for use in Class I, Division 1 hazardous areas such as wastewater wet wells. The hazard can be circumvented by using very low energy, intrinsically safe control circuitry but, because this adds another level of design complication, some engineers avoid them.

Displacers

Displacers differ from floats in that the buoyant body has a density greater than that of the liquid being monitored. The displacer, shown in Figure 20-4, is a cylindrical body whose height is at least equal to the maximum level span to be monitored. As the displacer is immersed, it loses apparent weight by the weight of the liquid displaced (in accordance with Archimedes' principle), and this apparent weight loss stretches the spring slightly and actuates the switch used to indicate liquid level and start or stop pumps. Built-in limit stops prevent overstretching of the spring. Displacers do not produce any significant mechanical motion, but rather work on the force-balance principle. They are very suitable for both analog and discrete measurements, but do not produce a direct readout of level. The useful measurement range is limited by the practical cylinder length—less than 10 ft. Displacers are better than floats in wastewater applications because the liquid rises and falls on a smooth cylinder producing a washing action, although fouling is still possible.

Table 20-1. Parameters Frequently Monitored

Parameter	Station type: Clean water	Station type: Wastewater	Reasons
Liquid level			
Wet well		X	Pump control Maintenance dispatch
Suction storage	X		Pump control Maintenance dispatch
Distribution storage	X		Pump control Maintenance dispatch
Dry well		X	Maintenance dispatch
Pressures			
Distribution system	X		Pump control
Pump suction	X	X	Troubleshooting
Pump discharge	X	X	Troubleshooting
Flow			
Influent or effluent	X	X	Pump control Troubleshooting Billing Chemical feed control
Pump status			
Running	X	X	System operation
Speed		X	System operation
Available	X	X	System operation
Maintenance dispatch			
Primed	X	X	Pump control
Dry suction		X	Equipment protection
Bearing temperature	X	X	Equipment protection
Motor temperature	X	X	Equipment protection
Operation time	X	X	Maintenance dispatch
Time start/stop	X	X	Troubleshooting Litigation defense
Start/stop sequence fail	X	X	Maintenance dispatch
Vibration	X	X	Equipment protection Maintenance dispatch
Water in motor	X	X	Maintenance dispatch
Valve and gate positions	X	X	System operation
Chemical feed systems			
Storage weight or level	X	X	Inventory Maintenance dispatch
Feed rate	X	X	System operation
Chlorine residual	X	X	Feed rate control
Feed pumps running	X	X	Maintenance dispatch
Utilities and environment			
Voltage	X	X	Maintenance dispatch Alternate source control
Amperage	X	X	Troubleshooting Expansion planning
Wattage	X	X	Power cost control
Power factor	X	X	Power cost control
Temperature	X	X	Equipment protection Maintenance dispatch
Explosive atmosphere		X	Personnel protection Equipment protection
Chlorine gas leak	X	X	Personnel protection Maintenance dispatch Public protection

Table 20-1. Continued

Parameter	Station type: Clean water	Station type: Wastewater	Reasons
Sulfur dioxide gas leak	X		Personnel protection Maintenance dispatch Public protection
Hydrogen sulfide gas		X	Personnel protection
Ventilation equipment status	X	X	Equipment protection
Fire	X	X	Equipment protection Maintenance dispatch Fire brigade dispatch
Unauthorized intrusion	X	X	Equipment protection Public protection
Air pressure	X	X	Maintenance dispatch
Fuel level	X	X	Inventory
Control power voltage	X	X	Maintenance dispatch
Battery chargers operating	X	X	Maintenance dispatch
Sample pumps running		X	Maintenance dispatch
Engine systems			
Engine running	X	X	System operation
Engine available	X	X	System operation
Engine malfunction	X	X	Maintenance dispatch

Admittance Probes

The admittance system is a modern improved version of the older capacitance probe. It consists of a slender, usually Teflon®-coated rod that extends into the monitored liquid. The rod is excited with a low-power, high-frequency voltage. An electrical current flows between the rod and a ground reference, which may be either the fluid itself (if it is a conductor) or a ground plane on the wall of the tank (if the fluid is essentially a nonconductor). The current flow varies as the fluid level changes and provides a measurement of the tank level. The system is totally solid state, which produces neither motion nor mechanical force. Solid-state systems are most useful where a transmitted signal is required. They do not produce a direct mechanical readout, but they can be fitted with a local electronic indicator to display the level.

Ultrasonic Measurement

The ultrasonic level measurement system consists of a transducer (suspended above the fluid) and associated

Table 20-2. Level Measuring Elements[a]

Element	Clean water	Wastewater	Discrete	Analog	Comments
Air bubbler	E	E	E	E	High cost, general application
Float	E	P	E	F	Low cost
Ultrasonic	E	E	E	E	High cost, general application when instrument air is not available
Immersible head transmitter	E	F	NA	E	Replaces the diaphragm box
Differential pressure	E	G	NA	E	High cost, limited application
Conductivity	E	F	E	NA	Low cost
Admittance probe	G	G	E	G	Moderate cost
Displacer	E	F	E	G	Limited range
Diaphragm box	G	F	G	G	High cost, obsolete
Microwave	E	E	E	E	High cost. Use instead of ultrasonic for difficult conditions (see text)

[a]E, excellent; G, good; F, fair; P, poor; NA, not applicable

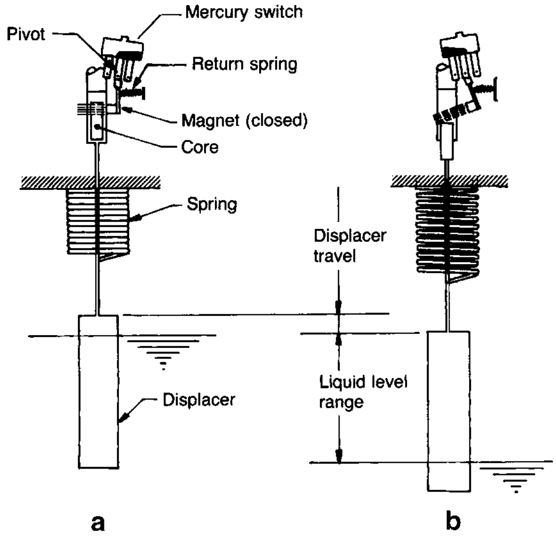


Figure 20-4. Displacement-type liquid level control. (a) Mercury switch closed; (b) mercury switch open.

electronics. The transducer periodically emits a pulse of sonic energy toward the liquid, and the pulse is reflected by the liquid surface back to the transducer. The transit time of this pulse is approximately proportional to the distance between the transducer and the liquid surface. The speed of the sonic pulse is affected by the density of the gas above the liquid and, hence, by both barometric pressure and temperature. Good-quality ultrasonic level systems are fully compensated and provide very accurate level measurement. But many systems are either uncompensated or only temperature compensated and cannot measure with accuracy. Readings may also be affected by foam, which causes either a false reflection or a loss of reflection. Discontinuities in the vessel walls may also cause false reflections.

The better quality ultrasonic level systems have microprocessors that reject spurious reflections. Nevertheless, when applying ultrasonic level measuring devices, check the angle of the emitted sonic pulse to ensure that the space within the transmission cone is clear of obstructions. Manufacturers rate ultrasonic level transducers for operation from −40 to +90°C (−40 to +200°F), but field installations have not been satisfactory below freezing or in windy locations. The ultrasonic system is totally solid-state and does not provide a direct mechanical indication of level.

Modern microprocessor-based ultrasonic level systems are accurate and reliable. They have supplanted bubbler systems in many facilities where maintenance of instrument air supply equipment has become uneconomical or onerous.

Microwave

The microwave level measurement system is similar to the ultrasonic except that a microwave frequency pulse is emitted toward the liquid surface. Microwave pulses penetrate foam, and the speed of microwaves is affected very little by the density of the gas above the liquid or by very low or high temperatures. Thus, although considerably more expensive than the ultrasonic system, the microwave system operates reliably under much more difficult conditions.

Immersible Head Transmitter

Immersible electronic head transmitters are designed to be completely immersed in the liquid being monitored. A cable containing both the signal leads and a pressure reference tube connects the transmitter to a junction box above the vessel or channel. The pressure element is typically the strain gauge type described under pressure transmitter. Immersible head transmitters were originally designed for oil field borehole use. They are economical, rugged, and reliable. They are cylindrical in shape, and some users drop them into the liquid through a pipe such that the sensing diaphragm, which forms the bottom end to the transmitter, is flush with the bottom of the pipe. This convenient installation protects the transmitter from gross solids present in wastewater applications. These units have been used as a complete replacement for air bubblers in clean water applications. However, the typical size (25 mm or 1 in.) is

not suitable for narrow spans (less than 700 mm or 2.3 ft). With 75-mm (3-in.) diaphragms, they can be used for controlling V/S pumps where narrow spans are required. With reasonable frequency of cleaning, they are suitable for wastewater applications if the enclosing pipe is extended below LWL to protect the transmitter from scum.

Conductance Probes

Conductance probes utilize the ability of virtually all liquids to conduct electricity. When the probe is immersed, a small electric current flows between probes or between a probe and the tank ground, and this current is detected by a sensitive relay. Conductance probes are suitable only for discrete measurements. They are used primarily in clean water applications. In wastewater applications, grease deposits and the necessity of using low electrical energy levels to prevent ignition of hazardous atmospheres generally rule out these devices.

Air Bubblers

The bubbler is probably the most common liquid level measurement device used in wastewater applications. It consists of a dip tube into which a small amount of air is metered. The back pressure produced is a measure of the liquid level above the tube outlet. Any pressure-measuring device can then be used to produce either an analog or a discrete signal, and a pressure gauge can be used for a direct readout. The bubbler works well with any liquid, including sludge and scum. Its advantages also include safety in explosive atmospheres and longevity in corrosive environments. The perceived disadvantages are the complexities, continual maintenance, problems with leaks, and high cost. But if the advice in this section is followed, air bubbler systems are easily maintained, leaks are easily fixed, and the system is the most reliable of all. Some bubbler systems have been replaced by tethered floats (sometimes not the best of choices), by ultrasonic measurement systems, or by immersible head transmitters. Consider the latter two if the bubbler is the only device requiring instrument air.

A complete air bubbler assembly for level measurement is shown in Figure 20-5. Instrument air is the most vulnerable of all systems, so do not use a cheap installation. Oil-free air (which cannot be obtained with oil-pumped and filtered air) must be supplied by oil-less compressors. Air is most commonly supplied by dual-redundant, piston-type compressors derated from 550 to 690 kPa (80 to 100 lb/in.^{2}) to 400 kPa (60 lb/in.^{2}) discharging to an oversized receiver so that the compressor does not cycle frequently. Use refrigerated air dryers and specify a hot gas bypass to prevent freezing. Provide for air purging of the dip tube at no less than 200 kPa (30 lb/in.^{2}). Neither compressor should have to operate more than 5 min/h to produce 28 L/h (1 ft^3/h) of air.

The pressure regulating valve, shown in Figure 20-5, reduces supply pressure to a level low enough to prevent possible damage to the pressure element if the dip tube plugs but high enough to ensure purge flow at maximum liquid level. If high accuracy is required, a constant-flow regulator is needed to prevent variations in purge flow from introducing dynamic headloss variations. The regulator is set to maintain a constant differential pressure, usually 20 kPa (3 lb/in.^{2}), across the needle valve so that the valve becomes a constant-

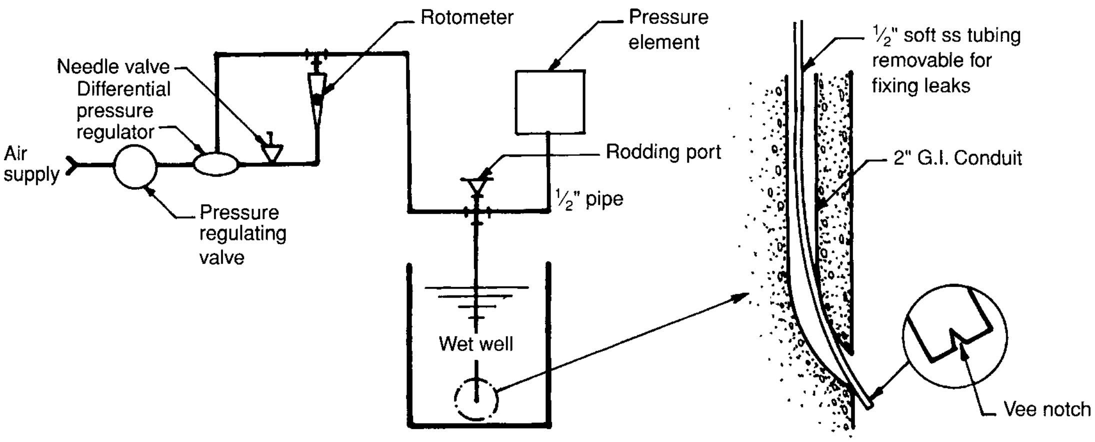


Figure 20-5. Air bubbler assembly.

flow regulator. The flow rate is measured by the rotameter. Usually, the differential pressure regulator, needle valve, and flow indicator are furnished as a factory-assembled unit. Air flow is typically adjusted to about $0.025\,\text{m}^3/\text{h}$ ($1.0\,\text{ft}^3/\text{h}$), but the flow rates are sometimes a third as much.

Dip tubes should always be installed so that they can be easily rodded out if they become plugged. The bottom of the dip tube is notched to keep bubble size as small as possible to maintain the highest accuracy. Some engineers (and owners) prefer straight dip tubes (for easy rodding) made of 13-mm ($\frac{1}{2}$ -in.) Schedule 40 PVC or stainless-steel (ss) pipe attached to the wet-well wall with pillow blocks. Others prefer to install soft stainless steel tubing in 50-mm (2-in.) conduit of harder stainless steel or galvanized steel encased in the concrete. The conduit must be bent so that the dip tube can enter the pool, but the bends must have a long radius to make it easy to insert the dip tube. Rodding must be done with a flexible rod (preferably furnished by the contractor). In either installation, make all joints, especially the rodding port, easily accessible for finding and fixing leaks. Joints should *never* be encased in concrete.

Pressure instruments used with bubbler systems should be installed adjacent to the bubble pipe (dip tube) to keep tubing lengths as short as possible because long tubing runs (e.g., 30 m or 100 ft) can add significant time delays to the measurement, increase the possibility of leaks that disable a bubbler system, and, in closed-loop control systems, cause instability. One way to eliminate these problems is to mount the purge panel and pressure instruments on a wall or handrail adjacent to the bubble pipe to keep tubing runs to less than about 3 m (10 ft).

High accuracy is not usually required for pump control—especially if the range of wet well levels is 1 m (3 ft) or more—so the high cost can be somewhat reduced by simplifying the system. The differential pressure regulator is often omitted even though this causes the purge flow to vary with liquid level, and small errors result from changes in system dynamic pressure losses and changes in bubble size. If an analog signal is required, a bubbler with an electronic change-of-pressure transmitter is satisfactory. Providing a different (or even an additional) system for back-up is good practice, so use a float switch for the high-level alarm.

Captive Air Systems

Essentially, the captive air system consists of a long tube that traps air. The pressure in the tube is a reflection of the depth of the liquid over the end of the tube. It is similar to a bubbler, but it does not require such an elaborate air supply system. On the other hand, it does not have the self-cleaning and purging aspects of a bubbler. Some utilities prefer it because of its simplicity.

Diaphragm Boxes

A diaphragm box consists of a closed box with an elastomer diaphragm forming one wall. The system is obsolete and has been supplanted by immersible electronic pressure elements, which may be installed in the same fashion and are much more accurate.

Differential Pressure Elements

Differential pressure units may be flange-mounted to a tank to expose the pressure element directly to the fluid. This approach is practical only when (1) freezing does not occur or can be prevented, and (2) access to the side of the tank is readily available. If the tank is not at atmospheric pressure, a second connection must be made above the maximum fluid level. The pressure exerted on the pressure element is then a direct indication of the fluid head above the connection point. This system, which is simple and very accurate, is the favored level measuring system in industrial applications, but it does not work if the specific gravity of the fluid in the tank is unpredictable. An isolating valve can be placed between the pressure unit and the tank to facilitate instrument maintenance without emptying the tank, but this modification entails a length of piping between the pressure element and the tank that is subject to clogging in dirty water applications.

Install a water connection for convenient flushing of any pipe used between the vessel and the diaphragm. Flushing can done be at regular intervals or a continuous small flow can be used to isolate the pressure cell from contaminated fluid.

20-4. Pressure Measurements

Pressure is a basic measurement that is made not only for itself but also for converting many other measurements into readable parameters. For example, many level and flow measurement systems produce a pressure that can infer a value for another parameter. Pressure is, in fact, almost always measured as a differential pressure. Atmospheric pressure is nor-

Table 20-3. Pressure Measuring Elements[a]

Element	Clean water	Wastewater	Discrete	Analog	Comments
Bourdon	E	G	E	G	Low cost, general application
Diaphragm	E	G	E	G	Low cost, low pressure
Bellows	G	P	E	G	Special applications
Force balance	E	E	NA	E	Becoming obsolete
Strain gauge	E	E	NA	E	Moderate cost, general application
Capacitance	E	E	NA	E	Moderate cost, general application
Manometers	E	NA	G	E	High cost, special application

[a]E, excellent; G, good; P, poor; NA, not applicable

mally the reference, but virtually all pressure-measuring elements can be arranged for connection of an independent reference pressure. The various types of pressure elements are summarized in Table 20-3.

Pressure gauges should be installed on both the suction and discharge spools (short pipe section) of every pump (see Figure 20-6). Instead of fixed gauges, however, pressure taps with short, 12-mm (½-in.) pipes terminating in shut-off cocks and quick-connects allow more accurate portable gauges to be used. Portable gauges are advantageous because only two are required and they can be kept in calibrated condition. Suction and discharge pressures are valuable checks on the condition of the station because they can be used to diagnose such troubles as blockages, deposits in the pipe, and worn impellers. In conjunction with the manufacturer's pump curve, pressure measurements can be used to measure flow with an error of no more than about 7% (refer to Section 3-9).

An alternative to the installation of a pressure gauge on the discharge of every pump is a single pressure gauge installed on the manifold; this is advantageous because only one gauge is needed and the flow and pressure are likely to be more stable than at a gauge near the pump. The disadvantages are (1) the need to shut off all of the pumps except the one under observation and (2) the loss of head between pump and gauge.

Bourdon Tubes

The Bourdon tube is the most common of pressure-measuring elements. It is used in virtually every pressure gauge and in many pressure switches. It consists of a semicircular length of flattened tubing. When pressure is applied to the inside of the tube it tends to straighten, and the mechanical motion can be coupled to indicators, switch mechanisms, and transmitter

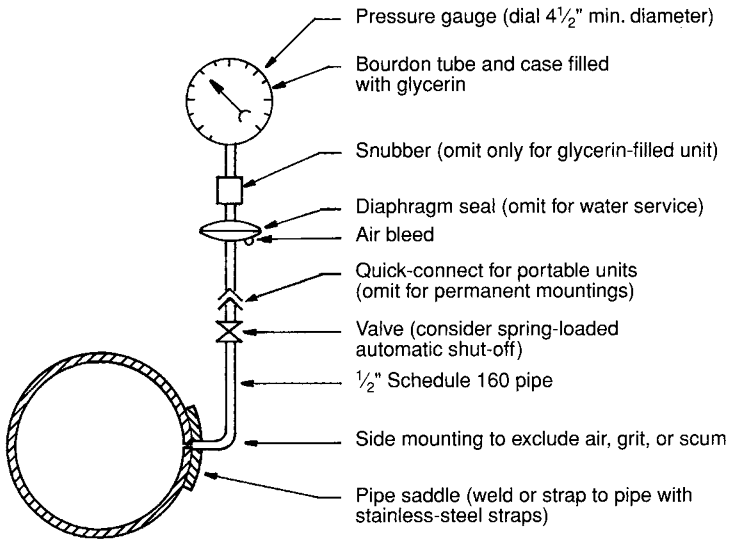


Figure 20-6. Proper Bourdon pressure gauge assembly.

mechanisms. It is a motion-balance device. The tube can be fabricated from copper-based and stainless-steel alloys, which offer moderate corrosion resistance. Wastewater and other dirty waters must be excluded from the Bourdon tube by a diaphragm seal, as shown in Figure 20-6. All of the tubing (including the Bourdon tube) above the diaphragm seal is filled with glycerin. The case is also filled with glycerin, thereby immersing the rack and pinion gears both to prevent corrosion and to dampen rapid pressure fluctuations. The snubber (a micropore filter) can be omitted, particularly if the shut-off cock is normally closed. Bourdon tube pressure elements are suitable for moderate to high pressures—generally more than 100 kPa (15 lb/in.^{2})—although low-pressure gauges are available.

For hard-to-gauge liquids such as sludge, one form of Bourdon gauge installation consists of an annular, flexible, impermeable tube or liner that is recessed so that its inner wall is flush with the pipe. The tube is connected to the Bourdon gauge, and both are filled with a sensing fluid. Various fluids and tube materials are available. The advantages are a protected gauge, responsive pressure readings, and protection from clogging. The disadvantages are cost (perhaps five times the cost of a Bourdon gauge), a small amount of hysteresis, and some loss of accuracy.

Diaphragms

Diaphragm pressure elements are used for lower pressure spans than those of Bourdon tubes. A variety of materials can be used to provide corrosion resistance, and mechanical configurations are available that permit mounting the diaphragm flush with a vessel wall to permit use with solids-bearing fluids. Housings can be designed to protect the diaphragm from high overpressures. Large sizes can be made to provide sufficient sensitivity for measuring very low pressure spans. The limited amount of mechanical motion is sufficient for direct actuation of precision switches and transmission mechanisms. Diaphragms are also coupled to Bourdon tubes by liquid-filled capillary tubes to produce a common type of chemical seal.

Bellows

A bellows is a stack of diaphragms. Bellows elements can measure very low pressure spans and, at the same time, produce large amounts of mechanical motion for operating indicators and switch and transmission elements. Bellows elements are not available in corrosion-resistant materials, their configuration is unsuitable for solids-bearing fluids, and they cannot be protected against overpressure.

Force-Balance Transmitters

The force-balance mechanism is frequently coupled to a diaphragm to produce a very rugged pressure transmitter (see Figure 20-2). Mechanical power available from the transmitter pneumatics or electronics is used in a null-balance circuit to oppose movement of the diaphragm. As a result, mechanical motion and, hence, wear is limited to a very small amount. Force-balance transmitters can be built in very low pressure spans and can withstand high surges and overpressures without damage or alteration of the calibration. The pneumatic force-balance mechanism is the standard industrial pressure transmitter, but the electronic versions have become obsolete.

Strain Gauge and Capacitance Transmitters

These transmitters have replaced the electronic force-balance transmitter. Modern electronic circuitry permits the mechanical deformation of a diaphragm element to be limited to a very small amount and still provide sufficient movement to produce a usable output. These transmitters are extremely rugged and, in addition, provide error values of less than 0.25% of span.

Manometers

A manometer is simply a U-tube (see Figure 3-4), which contains an indicator fluid (such as mercury, air, on carbon tetrachloride) of known density. If a pressure difference exists between the two ends of the U-tube, the indicator fluid elevation in the two legs is different, and this difference is a measure of the pressure. If any but the lowest pressures are to be measured, either the U-tube must be very tall or the density of the indicator fluid must be great. Mercury has been used frequently, but it is a very poisonous substance that must *never* be used for potable water and should be avoided for any application.

Manometers are useful for direct readout of very low pressures. Mechanisms for the conversion of indicator fluid position to useful motion for switch and transmitter operation are complex and expensive. Manometers are very accurate, however, and are

sometimes used in conjunction with air bubblers to monitor reservoir levels where very small changes in level represent large quantities of water.

20-5. Flow Measurements in Pipes

Flow is an easy measurement to make in clean water applications, but very difficult to make in wastewater applications. Most flow elements convert flow to some other variable that is more readily measured, such as pressure, level, or an electrical signal. The accuracy of flow measurement is frequently of considerable concern, and sustained accuracy is difficult to achieve without giving great care to the installation and the continued maintenance of the instruments (see ASME MFC-3M [3]). Instrument types frequently used for flow measurement are summarized in Table 20-4 and discussed by Miller [4]. The relative costs for some of the more common types of flowmeters, including the 4- to 20-mA output but excluding installation, are shown in Figure 20-7.

The accuracy of flowmeters is (for most types) significantly affected by piping configurations in the vicinity of the flow element. Because designing the approach conditions to match factory or laboratory flow test facilities is ordinarily impossible, the manufacturers' claims of accuracy are rarely realized in practice. The effects of nonstandard approach conditions on orifice and Venturi meters have been thoroughly explored and illustrated graphically by Starrett et al. [6]. Investigations of field installations show that errors greater than 10% are the rule in most installations, and errors from 50 to 200% are not uncommon [7]. *In situ verification of calibration is essential for confidence and accuracy*. Calibrations with the ‘‘meter provers'' (often used in the process industries) is impractical for the flows typically encountered in pumping stations. Adequately sized tankage associated with a pumping station can often be isolated and used for an approximate volumetric calibration, but much more accurate and reliable results can be obtained with tracer techniques (see Section 3-9). Pipeline traverses with either an inserted magnetic probe or a pitot tube should be considered only as a last resort. Accurate traverses are difficult to accomplish in the presence of swirling and/or pulsating flow conditions caused by flow profile disturbances.

Orifice Plates

An orifice plate is a simple, flat restriction (in a pipeline) that produces a differential pressure propor-

Table 20-4. Flow Measuring Elements

Primary element[a]	Clean water[b]	Waste water[b]	Secondary element[a]	Comments
Orifice plate	G	X	Pressure	High headless, range 8:1, error−1–2%
Venturi meter[d]	G-E	F	Pressure	High cost, range 8:1, error−0.25–1%
Flow tube	G	F	Pressure	Medium cost, range 5:1, error−1–2%
Elbow meter	F	P	Pressure	Cheap, range 4:1, error−2%[e]
Magnetic	E	E	Electronic	High cost, accurate, range 10:1, error 0.5–1%
ADFM Velocity Profiler™	X	E	Electronic	High cost, accurate, range 20:1, error 2%
Doppler	P	G	Electronic	Low cost, no intrusion into pipe range 10:1, error−2−20%
Transit-time ultrasonic	E	F	Electronic	Special applications, error−1%.
Propeller	E	X	Mechanical	Low cost, accurate, range 10:1, error−2%
Turbine	E	X	Mechanical	Accurate, range 100:1, error ±1%
Weir	E	X	Level	High headless, standard open channel device, fair accuracy, range 20:1, error−2%
Flume	G	G	Level	High cost, fair accuracy, range 20:1, error−4%
Vortex	G	X	Pressure change frequency	Range 20:1, insensitive to fluid properties, error−1%

[a]See Chapter 2 for definition.

[b]E, excellent; G, good; F, fair; P, poor; X, not to be used.

[c]Typical error to expect under ‘‘best'' field conditions with premium quality instruments and frequent, careful recalibration. Expect double the stated error if recalibration is infrequent.

[d]Venturi meters are considered to be the standard flow measuring device for custody transfer. Accuracy given is confined to suitable Reynold's numbers.

[e]Only if accurately calibrated in situ.

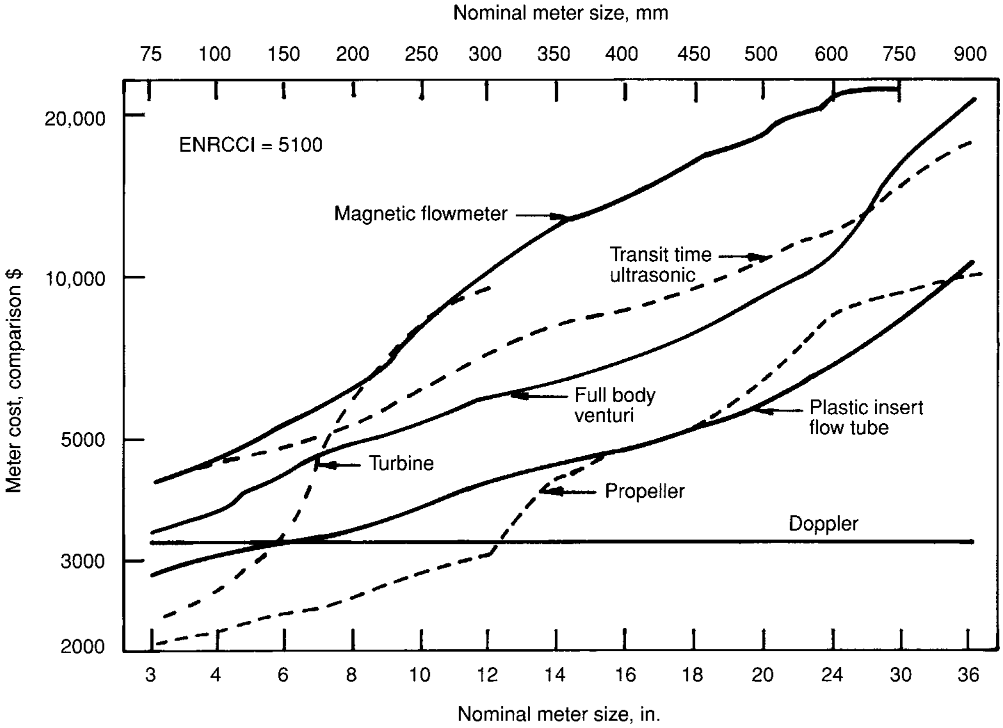


Figure 20-7. Relative 1993 costs for flowmeters with 4 to 20 mA outputs but no processor nor installation costs. Courtesy of WaterWorld Review [5].

tional to the square of the flow velocity in accordance with Bernoulli's principle (see Figure 20-8). Small reductions in flow cause a great reduction in the differential pressure, so orifice plates should not be used for flows that vary over a range greater than 8:1. Because the abrupt restriction does not pass solids, this instrument is unsuitable for wastewater, and head recovery downstream is only moderate so there is a relatively high permanent headloss. Orifice plates require a fully developed flow profile to produce accurate measurements, and, depending on pipe and orifice size, a straight pipe 6 to 45 diameters long is required upstream. The straight pipe requirement downstream is 5 diameters. The orifice plate system becomes unpredictable for Reynolds numbers below 10,000.

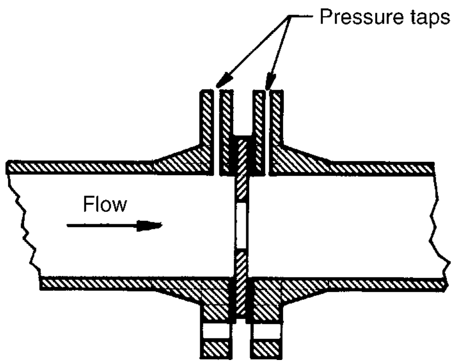


Figure 20-8. Orifice meter.

Venturi Tube

The Venturi tube (shown in Figure 20-9) also produces a differential pressure proportional to the square of the velocity. With its smooth approach and recovery cones, the Venturi tube works well in clean water applications and reasonably well in wastewater applications. Head recovery is excellent. The approach cone tends to correct aberrations in flow profile caused by poor approach conditions. Some error is introduced by irregular flow profiles, but the error is much less than with orifice plates. Standard Venturi tubes are acceptable as custody transfer flow-metering devices and are frequently used in clean water applications as revenue meters.

The principal disadvantage of a standard Venturi meter is the high cost for large sizes and long laying

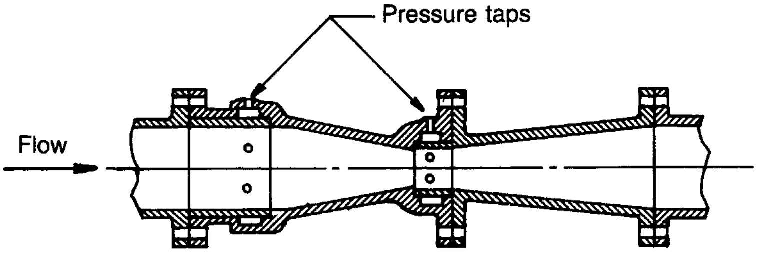


Figure 20-9. Venturi meter.

length (see Starrett et al. [6]). A number of proprietary designs (see, for example, Figure 20-10) have been developed to shorten the length and reduce the cost. The performance of these proprietary designs is generally excellent, but the volume of laboratory data available to substantiate their performance may not be as great, and their use as custody transfer meters may not be fully acceptable.

When Venturi tubes are used in dirty water or wastewater applications, solids must be excluded from the differential pressure measurement connections. The most common method is to install a water purge system, but even this is sometimes troublesome with raw wastewater. Diaphragms, flush-mounted in the throat wall, may also be used, but some difficulty is encountered in maintaining measurement accuracy with such arrangements.

Elbow Meter

A 90-degree bend can be converted to an elbow meter by installing pressure taps on the inside and outside of the bend at either 22.5 degrees or 45 degrees [4] and measuring the differential pressure. Pressures at taps installed at angles greater than 45 degrees are erratic. Because the differential pressure is very low, an air manometer (Figure 3-4) is probably the best type of measuring device. The low cost and ease of installation (or the use of an existing elbow) make it a useful flowmeter for field testing in stations not otherwise equipped with a flowmeter.

Magnetic Flowmeters

The magnetic flow tube (shown in Figure 20-11) is an open cylinder that produces practically no headloss and readily passes solids. It functions with any conductive liquid, such as water or wastewater. It consists of a flow tube that develops a magnetic field across the flow profile. When a conductor passes through a magnetic field, an electrical potential proportional to the conductor's velocity (in accordance with Faraday's law) is produced. With proper design and modern electronics, very accurate flow measurements (which are virtually immune to distorted flow profiles) can be obtained.

The principal difficulty is with wastewater containing grease, which can coat the electrodes and interfere with the potential measurement. This problem has been overcome somewhat by high-quality electronic circuitry that can measure the potential across very poor conductors. Nevertheless, magnetic flowmeters for wastewater applications should be installed to facilitate cleaning. A piping configuration that allows

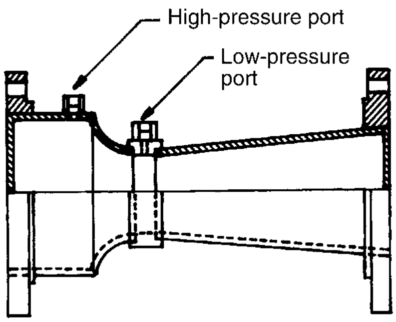


Figure 20-10. A typical proprietary flow tube.

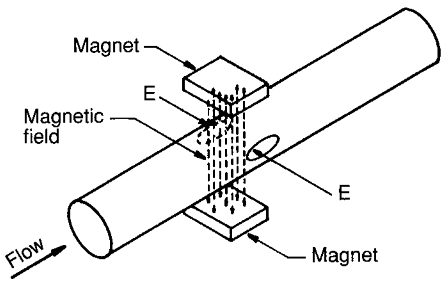


Figure 20-11. Magnetic flowmeter.

the electrodes to be cleaned without removing the flow tube from the pipe is shown in Figure 20-12. Alternatively, the meter can be installed in the straight pipe with a Dresser® coupling on one side and a Victaulic® coupling on the other side (or with Victaulic® couplings on both sides) to allow the meter to be removed while the flow goes around the installation in a bypass pipe. Another cleaning method sometimes used consists of pig launching and removal stations. Use only soft pigs for magnetic flow tubes. Always arrange the magnetic flow tube so that (1) no stress is placed on the flow tube, and (2) mating flanges can be separated to permit easy removal of the tube.

The magnetic flow tube liner must be both nonconductive and nonmagnetic. Polyurethane, Teflon®, and ceramic are the most commonly used liners. Polyurethane provides good abrasion resistance and is the standard liner material. Teflon® is used for meters in wastewater sludge service for its superior resistance to grease, but it has poor abrasion resistance and must never be used if the sludge contains much grit. Ceramic liners are available for meters up to about 150 mm (6 in.) and are resistant to both grease buildup and abrasion. For meters larger than 150 mm (6 in.), used with liquids that contain significant amounts of grit, use polyurethane and limit the velocity to 3 m/s (10 ft/s). The grit content of raw wastewater is usually low enough to allow a peak flow velocity of 4.5 m/s (15 ft/s) and the grease content is usually low enough to allow a minimum velocity of 0.3 m/s (1 ft/s), so the usable range is 15:1. With sludge, the fluid velocity range is about 1.0 to 3.0 m/s (3 to 10 ft/s) unless grit is effectively removed before entering the meter. Always provide a means for easily cleaning the electrodes. Failure to observe these limitations has frequently resulted in unsatisfactory service in wastewater applications. Nonetheless, the magnetic flowmeter remains the most satisfactory wastewater flowmeter. In clean water flow applications, the magnetic flowmeter can function from 0.03 to 6.0 m/s (0.1 to 20 ft/s), but the electronics generally limit the usable range to about 20:1.

Probable errors in magnetic flowmeters are typically stated as 0.5% of flow for liquid velocities greater than 1.5 m/s (5 ft/s) and 0.0075 m/s (0.025 ft/s) for fluid velocities less than 1.5 m/s. Most vendors can furnish meters with half of these errors, but typically there is extra cost for custom calibration. The magnetic flowmeter is one of the most accurate meter types available. The initial cost may seem high, but the simplicity, reliability, and low maintenance makes its life-cycle cost competitive with other meter types up to about 400 mm (16 in.) in diameter. For applications other than sludge and larger than 400 mm, consider the transit-time ultrasonic meter.

Sonic Meters

There are two types of sonic flowmeters: (1) Doppler meters and (2) transit-time meters. Other than their common use of ultrasonic pressure waves, the operating principles of the two are completely different, and their applications are different. The Doppler meter is useful for measuring flows in fluids containing at least 100 ppm of suspended solids or air bubbles for which the transit-time meter is useless. Unlike the Doppler meter, the transit-time meter is very accurate.

Doppler Meter

The Doppler meter (see Figure 20-13) consists of a pair of piezoelectric transducers, which are usually contained in a common case and mounted on the outside of the pipeline. One transducer injects an ultrasonic pressure wave at frequency f_1 into the pipeline, and this pressure wave is reflected by solids, gas bubbles, or other discontinuities in the liquid

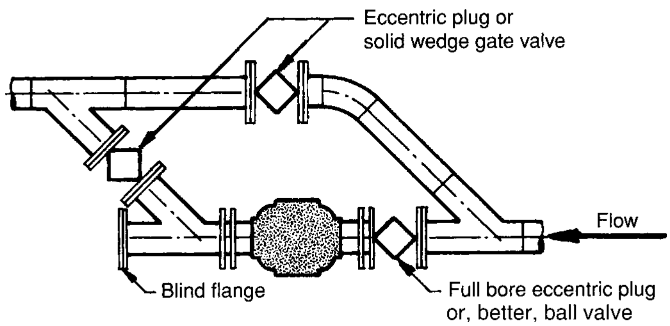


Figure 20-12. Piping plan to allow cleaning a magnetic meter in situ. With permission of The Foxboro Company.

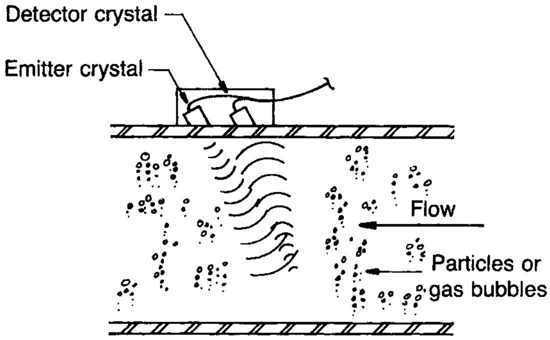


Figure 20-13. Doppler flowmeter.

back to second transducer. The reflected frequency, f_2, varies by an amount proportional to the reflecting surface velocity. According to the principle of Christian Doppler,

$$f_a = f_1\left(\frac{v}{v - s}\right) \tag{20-1}$$

where s is the velocity component axial to the normal pressure wave propagation and v is the velocity of sound in the fluid. The practical execution of this principle is fraught with difficulties. The sound pressure waves must be precisely controlled, and the reflecting particles must be moving coaxially with the pipe at a speed representative of the flow (which varies from wall to center).

There are many error sources in Doppler flowmeters: (1) the reflecting surfaces may not be moving at the same speed as the average velocity of the liquid; (2) they may not be moving parallel to the main liquid flow; (3) the frequency shift is affected by the speed of sound in the liquid, which is a function of temperature, pressure, and density and cannot be fully compensated; and (4) the angle of injection of the pressure wave is difficult to control accurately. In summary, the Doppler meter is inaccurate. Errors as great as 20% occur regularly in practical installations. Doppler meters are used only in dirty water applications because clean water does not contain reliable reflective surfaces.

Doppler flowmeters do have the outstanding advantage that there is nothing to foul or corrode. Users consider them to be maintenance-free devices. They are frequently used on sludge pipelines where accuracy is not particularly important and the flow is very difficult to measure at all with other devices. The maximum velocity is limited only by what can be practically pumped through a pipeline, and the minimum velocity is limited only by the resolution of the frequency measurement circuitry—about 0.3 m/s (1 ft/s) for most units. In sludge service, however, the minimum velocity must be enough to prevent a buildup on the pipeline walls, which would change the effective pipe diameter.

The Doppler flowmeter is also available as a moderately priced, nonintrusive flow switch. Accuracy is not critical in this application. The Doppler flow switch is particularly useful as a pump discharge flow detector, where limit switches cannot be used on a discharge check valve. If the Doppler flow switch is located immediately downstream of a pump, reliable signals are produced even with clean water, because the bubbles from the pump impeller reflect the sonic pulse.

Velocity Averaging Doppler Meter

An advanced form of Doppler signal processing can be employed to measure velocity data at multiple points in the flow cross-section, rather than estimating an average flow velocity from a single velocity measurement. One such type of Doppler meter is the ADFM® Velocity Profiler. It incorporates sophisticated technology to correct most of the errors inherent in other Doppler flowmeters. Excellent accuracy has been demonstrated in both open channels and pressure pipes even with highly distorted flow profiles and at velocities less than 0.3 m/s (1 ft/s) and more than 1.2 m/s (4 ft/s). As with any Doppler meter, some suspended solids and/or gas bubbles must be present in the flow to provide reliable reflection of the ultrasonic pulses. The manufacturer's claims regarding approach conditions are reasonably sound, but there have been instances where upstream variations in turbulence have resulted in poor or inaccurate performance.

The transducer emits five acoustic beams—one for measuring depth and four for measuring velocity profiles. The four velocity-measuring beams are grouped into two upstream–downstream pairs. Due to this beam geometry, only two of the velocity-measuring beams can be seen in each of the views of Figure 20-14. Echoes from the four inclined velocity-measuring beams are back-scattered from particles suspended in the flow. Because the particles move with respect to the transducer, these echoes are Doppler shifted in frequency.

The ADFM divides the return signal from each beam into multiple discrete intervals, known as bins, which correspond to different depths in the fluid. One velocity profile is calculated from the horizontal velocity components measured in each depth cell of the first upstream-downstream beam pair. The second beam pair on the opposite side of the transducer produces a second velocity profile. Velocity data from these two profiles are converted into flow rate by means of an algorithm that describes flow velocities over the entire cross-sectional area of the flow. The result is an accurate determination of flow rate with no need for calibration in situ. The ADFM® also adapts to changing hydraulic conditions, as these changes are directly manifested as a change in the real-time measurement of the velocity distribution. In tests at the Bureau of Reclamation laboratory of a flume 1.2 m (4 ft) wide and another 3.6 m (12 ft) wide, the profiler errors were 1.7% and 1.0% respectively.

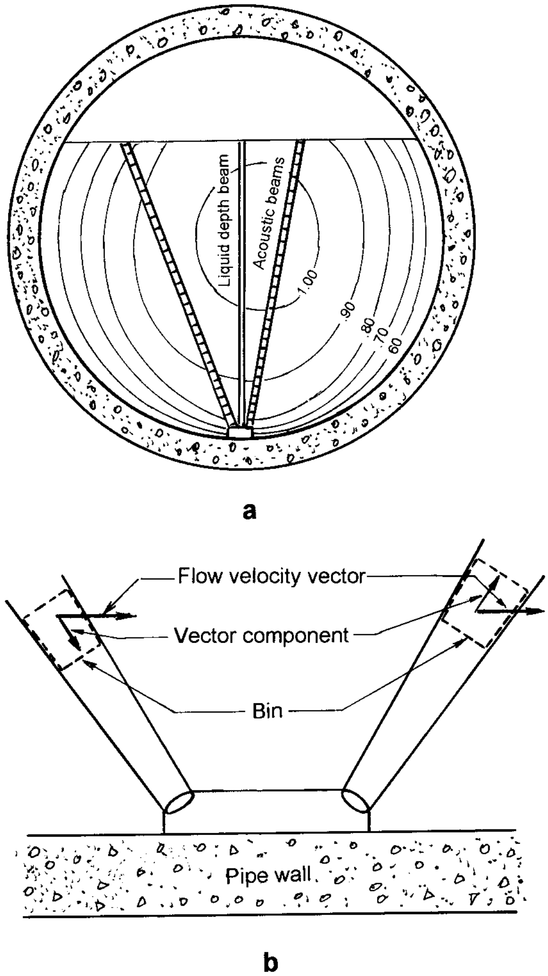


Figure 20-14. ADFM Velocity Profiler. (a) Cross-section with velocity contours; (b) transducer [8].

Transit-Time Ultrasonic Flowmeters

The transit-time ultrasonic flowmeter (see Figure 20-15) consists of at least one pair of piezo-electric transducers mounted on opposite sides of the pipeline and offset from each other by 30 to 45 degrees. Additional transducer pairs are often used to provide the ability to average the flow profile. An ultrasonic pressure pulse is injected by one transducer and the period of time for it to reach the second transducer is measured. The process is then reversed. The difference in time required for the pulse to propagate upstream and downstream through the fluid is proportional to the average fluid velocity between the transducers. This metering system is capable of very high accuracy, and when multiple transducer pairs are used, it can average out irregularities in the flow profile. The process of reversing the signal direction cancels errors that would be caused by variations in the speed of sound through the fluid. This meter does not perform well in dirty water or wastewater applications. The same solids, gas bubbles, and other fluid discontinuities that make the Doppler meter work diffract the pressure pulse so that it arrives at the receiving transducer as a burst of noise, not as a single pulse. High-quality electronics can deal with the noise to a certain extent, but they increase the cost of the meter substantially.

The transit-time ultrasonic flowmeter is especially useful on very large pipes where cost becomes less of a factor. The maximum velocity is limited only by what can be pumped through a given size pipe, and the minimum velocity is limited by the resolution of the timing circuits—0.3 m/s (1 ft/s).

Propeller Meters

As its name implies, a propeller meter contains a propeller installed coaxially in the center of the pipe, as shown in Figure 20-16. Flow turns the propeller at a speed proportional to the liquid velocity. It is a low-cost device used primarily in clean water applications, where it gives excellent service. Depending on the manufacturer and the meter size, the mechanical nature of the equipment limits the usable flow range to about 0.15 to 3.0 m/s (0.5 to 10 ft/s). Its only disadvantage (for clean water service) is the need to replace bearings from time to time.

Turbine Meters

Turbine flowmeters have many blades on a rotor whose speed is a linear function of fluid flow velocity within an error of about ±0.5% over a very wide flow range of

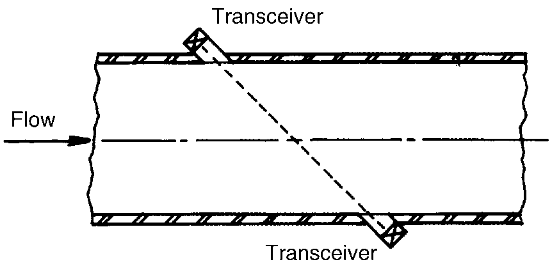


Figure 20-15. Transit-time ultrasonic flowmeter.

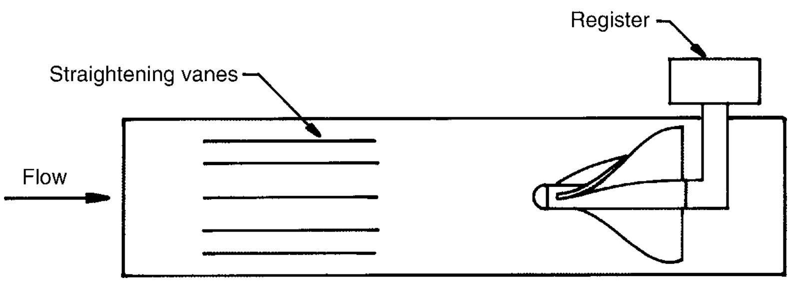


Figure 20-16. Propeller meter.

10:1 or even 20:1. The blades are vulnerable to particulate matter, so the meter is used only for clean water.

Vortex Flowmeters

In vortex flowmeters, a transverse, flat bar (called a "bluff body") across the pipe splits the flow into two paths and causes vortices to shed alternately from the sides of the bar at a frequency that is (1) linearly proportional to velocity, and (2) independent of fluid density over flow ranges up to 30:1. The meters can be used for liquids, gases, and cryogenic fluids. There are no moving parts, and the construction allows sand in dirty water to pass without obstruction. The shedding rate for vortices is linear above a Reynolds number of 10,000. The meters are available in sizes from 38 to 400 mm (1.5 to 16 in.), and their use in chemical and industrial fields is increasing because of their accuracy (which is slightly greater than the accuracy of a Venturi meter), range, and insensitivity to fluid properties. These excellent characteristics are no less applicable to the measurement of water.

Summary

For clean water applications, the propeller meter is a good choice for all sizes of pipes. For wastewater, the Venturi meter is widely used even though keeping the pressure ports clean is troublesome. Venturi meters for pipes larger than 600 mm (24 in.) become comparatively costly and require straight-approach pipes that may be inconveniently long. Install a freshwater purge system and a pipe configuration that allows the Venturi meter to be removed for cleaning. The magnetic meter is much preferred by many experienced designers because of its superior accuracy and reliability, although maintenance is still a problem. Both Venturi and magnetic meters are heavy in sizes larger than 300 mm (12 in.), so hoisting beams should be installed overhead. Consider the Doppler meter for measuring the flow of sludge. The accuracy of Doppler meters is not good (but usually adequate), maintenance is negligible, and operators like them. Except for raw wastewater service, consider the vortex meter.

20-6. Open Channel Flow Measurement

Weirs and flumes are used for open channel flow measurement. Weirs are applicable only to clean water and treated wastewater because they cannot pass heavy solids. Flumes may be used in both clean and wastewater applications.

Weirs

A weir is simply an obstruction across a channel, and weirs are built in a number of configurations, including rectangular, vee-notch, and trapezoidal. Vee-notch weirs typically have an included angle of 30, 60, or 90 degrees. Weirs are fabricated in the field, usually of concrete or steel, and the care used in their construction is a major limitation on their accuracy. Because the flow downstream from a weir must fall freely away from the crest, weirs introduce substantial headloss into the channel.

Depending on the weir configuration, the flow varies as a power (1.5–2.5) of the upstream level, so the change in level is not great with respect to the flow, and level instruments must be extraordinarily accurate. The usable flow range is typically 20:1, but the accuracy is limited because errors often exceed 5%.

Flumes

Flumes are built in a number of configurations, including Parshall (Figure 20-17), Palmer–Bowlus (Figure 20-18), parabolic, and cutthroat. Except for the Palmer–Bowlus flume (see Section 3-7), they are purely empirical devices and their accuracy is limited by the degree to which the field installation resembles the laboratory model. Approach conditions that produce irregular flow profiles and/or flows greater or less than that specified for the size give poor and unpredictable results. Properly sized and installed, flumes provide reliable measurement of clear water and wastewater—even with gross solids. The head change developed by a flume with a rectangular throat is proportional to the flow to approximately the 0.65 power. Flumes have a wide usable flow range—typically 20:1.

20-7. Chlorine Residual Measurement

Water is often chlorinated in water pumping stations. Standard analyzers in "package" units are available to control the dose or to measure the residual chlorine. Refer to the treatise by White [9] for a complete discussion of chlorination.

20-8. Utility and Environmental Measurements

Utility and environmental measurements include a large variety of miscellaneous measurements that

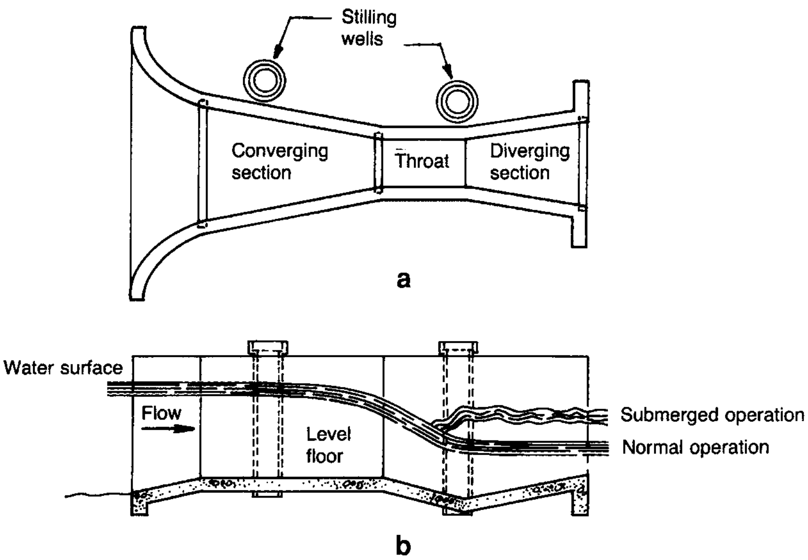


Figure 20-17. Parshall flume. (a) Plan; (b) longitudinal section.

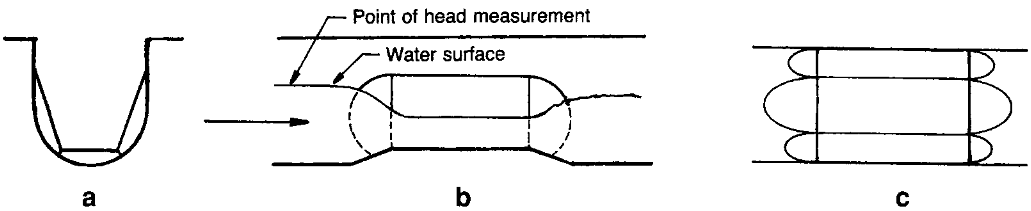


Figure 20-18. Palmer–Bowlus flume. (a) Cross-section; (b) longitudinal section; (c) plan.

provide information about and protection of the pumping station itself. These measurements are not unique to pumping stations. They might be made in any facility containing valuable equipment.

Electrical Parameters

Electrical parameters such as volts, amperes, watts, and power factor are easily measured either with direct-reading instruments or with transducers that convert these readings to standard transmission signals for activating alarms or telemetering to a distant location. Most electric power–monitoring instruments operate on rather simple electromagnetic principles and are very accurate, reliable, and virtually maintenance-free.

Temperature

There are a great many ways to measure temperature. Mercury thermometers should be confined to the laboratory because of the extreme hazard of a mercury spill. In field applications where only a discrete signal is required, the most common device is a closed chamber in which a fixed amount of gas or liquid produces a variation in pressure with changes in temperature.

A bimetal element consists of two metal strips with different thermal expansion coefficients formed into a spiral. As the temperature changes, the spiral tends to wind tighter or unwind. Bimetal elements are commonly used in residential applications both to indicate the temperature and to control on–off devices such as furnaces and air conditioners. These devices are low in cost but do not provide the long-term accuracy of liquid- or gas-filled systems. Where electronic transmission signals are required, coils of platinum, copper, or nickel alloys (which change resistance as the temperature changes) are usually used for temperature transmitters. Nickel alloy is low in cost and commonly used for heating, ventilating, and air conditioning, but it has a nonlinear temperature-resistance coefficient. Platinum has a linear coefficient, is extremely stable,

and is commonly used where accurate, long-term measurement is required.

Environmental Safety Instruments

Methane, hydrogen sulfide, and oxygen depletion monitoring are required in wastewater pumping stations. Instruments for measuring these parameters are costly and require substantial maintenance—particularly when installed in wet wells. Only fixed hydrocarbon monitors are required in wet wells by the NFPA. Hydrogen sulfide detectors are too unstable for permanent installation and these monitors should be carried by the personnel. Oxygen depletion is not generally considered to be a problem in wet wells.

The calibration of any of these instruments should be checked at least monthly. For small facilities, portable instruments carried by maintenance personnel may be a better solution if acceptable to the fire marshal. Portable instruments should be carried by maintenance personnel anyway, because they frequently must enter collections system manholes where permanently installed equipment is never present.

Explosive Atmosphere

Methane (an explosive gas) occurs in underground water sources and forms naturally in wastewater facilities, and fuel vapors sometimes occur due to accidental leakage or intentional dumping into sewers. The most common explosive-gas-monitoring device is the catalytic detector, which consists of two heated elements. One is exposed to the ambient gas and the other is isolated. If the ambient gas is combustible, it burns on the exposed element, which raises its temperature above that of the isolated element. The difference in temperature measures the combustible gas present. Because catalytic elements are easily poisoned by the hydrogen sulfide frequently present in wastewater facilities, detector life is limited. Some manufacturers produce catalytic detectors with improved resistance to hydrogen sulfide poisoning and typically guarantee the detector for one year. In extreme conditions infrared detectors, which are totally immune to poisoning, are available, but in 2005 these cost about $2900 per point—four times as much as the improved catalytic type.

Chlorine Gas Leak Detectors

Chlorine gas leak monitoring is required in either water or wastewater pumping stations if chlorination equipment is present. Permanently installed monitoring equipment is necessary at any chlorination location, because chlorine leaks can present a hazard to the public outside of the station. Where permanently installed monitoring systems are used, audible and visual warning devices must be installed at all entrances to the monitored space.

Hydrogen Sulfide Gas Detectors

Hydrogen sulfide is a ubiquitous, dangerous gas as deadly as hydrogen cyanide. Because it is so common around wastewater works (who has not smelled a faint odor of rotten eggs?), it is often ignored and many people have been killed or have suffered permanent brain damage from this insidious gas. Below ground, sample for hydrogen sulfide continuously. The effects of various concentrations are given in Table 20-5. Detectors are seldom installed permanently in pumping stations because they should be carried at all times by personnel in wastewater facilities when working in any confined space where wastewater is present.

Oxygen Depletion

Oxygen depletion monitors are available from the same companies that manufacture explosive gas and hydrogen sulfide detectors. A common type of unit has a galvanic cell containing two dissimilar electrodes in a basic electrolyte. Oxygen diffusing through the sensor cell face initiates a redox reaction that generates a minute current proportional to the oxygen partial pressure. The typical range is 0 to 25% oxygen concentration.

Table 20-5. Effects of Hydrogen Sulfide

Concentration in air	Effect
3 ppb	Max concentration for electronic systems per ISA
10 ppb	Max concentration for electrical equipment per NEMA
100 ppb	Threshold concentration for smell
1 ppm	Offensive odor of rotten eggs
5 ppm	Deadens olfactory senses
10 ppm	Max 24-h exposure limit per OSHA
15 ppm	Max 8-h exposure limit per OSHA
50 ppm	Max 30-min/d exposure per OSHA
300 ppm	Instant death for sensitive people
500 ppm	Instant death for everyone
46,000 ppm	Lower explosive limit (LEL)

Fire

Severe fire hazards do not normally exist in water and wastewater facilities because they are usually constructed of noncombustible materials and very little of the contents can burn. In very large facilities, the control equipment may present a significant fire hazard, and prompt detection of a fire may reduce the damage and facilitate repair. Because these electrical fires produce little visible smoke or heat, "products-of-combustion" detectors are the most suitable type. In some facilities, the local fire marshal may require a fire alarm system conforming to applicable building codes to protect personnel. The design of such systems is usually outside of the competence of instrumentation engineers, and specialists should be retained.

Unauthorized Intrusion

Intrusion alarm systems are frequently needed in urban areas not only to protect the pumping station, but also to protect the intruders from injury. Most pumping stations have a limited number of possible entry points, and simple door switches are adequate. If there are windows, passive infrared detectors may also be needed. To be effective, intrusion alarms must be transmitted to a continuously staffed facility, such as a police or fire station.

Air Pressure

Where instrument air and/or starting air systems form an essential part of a pumping station, pressure alarm systems are required. Simple pressure switches are adequate for this purpose.

Fuel Level

Where fuel is stored on-site for engine-driven pumping units or emergency generators, the fuel supply must be monitored. A simple staff gauge is frequently used. If remote monitoring is required, some type of level measurement system is used. Automatic monitoring of diesel tanks can be accomplished with bubblers, floats, radio-frequency probes, or ultrasonic units. Buried liquid fuel tanks must be furnished with leak detection systems. Package tank monitoring units, which include the leak detection and level monitoring, are available and recommended.

Propane tanks are more difficult. Pressure measurement, which is often used for monitoring propane tanks, does a better job of measuring tank temperature than liquid level until the tank is almost empty. A differential pressure element connected to the liquid withdrawal nozzle on the bottom of the tank with a compensating connection to the gas withdrawal nozzle on the top of the tank provides a simple, accurate measure of tank level. Such transmitters are readily available with the static pressure rating of 4140 kPa (600 lb/in.^{2}) required for this application. Alternatively, the tank can be mounted on platform scales to give a direct measure of propane quantity.

Control Power Availability

Where possible, control circuitry should be designed so that it is not dependent on a single control power supply. If a single control power supply is used, it should be monitored by an undervoltage relay to provide an alarm if failure occurs.

Battery Charger Operation

Chargers designed for constant-voltage-float charging of stationary batteries are available with built-in alarm circuitry that provides a warning when the charger fails.

Sample Pumps Running

Sample pumps are used to transfer sample flows to chlorine residual analyzers and/or containers for subsequent laboratory analysis. In noncritical applications, a motor-starter auxiliary contact or control relay connected across the motor gives a reasonable indication of pump operation. A running pump, however, is not a sure indication of flow. Because a valve might be closed or the pipe might be blocked, a flow switch is needed in critical applications. Vane switches installed in a pipeline can be used in clean water applications. In wastewater applications, a thermal-dispersion flow switch (which is a heated temperature-sensitive element encased in a smooth cylinder that is inserted into the pipeline) is better. When flow is present, the element temperature stays near the fluid temperature, but when flow stops, the element temperature rises because the heat is not dissipated effectively. This temperature change can be detected by suitable electronic circuitry. The thermal-dispersion flow switch is much more expensive than vane switches, but the system is fairly resistant to fouling.

20-9. Pumping Unit Monitors

Most pumps are driven by constant-speed electric motors of moderate (less than 110 kW or 150 hp) size. Such units require only the standard protective devices provided with any electric motor, such as the replica-type thermal overload protection normally furnished in standard motor controllers. "Replica" means that the thermal overload device, which is located in the starter where it is heated by motor current, duplicates motor heating. It is most common in motors up to 150 kW (200 hp). In larger motors, it is better practice to add temperature monitors in the windings. Temperature monitors should also be used on most adjustable-frequency drives. Always equip engine drives with cooling water temperature monitors and lubrication oil pressure monitors.

Bearing Temperature

Bearing housings of larger (greater than 150 kW or 200 hp) units may be provided with temperature monitors. Frequently, these are filled-system temperature switches and may be specified to be furnished with the pumping unit. Alternatively, resistance temperature detectors (RTDs) or thermocouples (TCs) may be installed in bearing housings and connected to external electronic monitoring equipment. Both the RTDs and the TCs are more accurate and easier to adapt to remote monitoring.

Motor Winding Temperature

Motor winding temperature monitors provide much better overtemperature protection than replica-type thermal overloads. Three types are commonly used: (1) end turn snap switches (Klixons™), (2) positive temperature coefficient (PTC) thermistors installed in the winding slots, and (3) resistance temperature detectors (RTDs) installed in the winding slots. Klixons™ are not adjustable, may produce a winding hot spot where attached, and are not recommended. The PTC thermistors are recommended for motors up to about 750 kW (1000 hp), and RTDs are recommended for larger motors.

Pump Speed

In many applications, a motor-speed meter is an adequate measure of pump speed and can be obtained as an accessory with adjustable-speed drive equipment. Where an actual measure of pump shaft speed is required (either for positive interlocking purposes or because the adjustable-speed drive equipment cannot provide a speed signal), either a belt-driven tachometer or a reluctance pickup device may be used. Belt-driven tachometers are mechanical devices with bearings and brushes that require significant maintenance. The belt cannot be replaced without opening the coupling between the driver and the pump. A reluctance pickup consists of a split gear clamped to the pump shaft and a magnetic pickup coil assembly mounted close to (but not in contact with) the gear. As the gear teeth pass the pickup coil, a small alternating current voltage is generated that can be detected by suitable electronics. These devices are essentially maintenance-free and are very accurate.

Vibration

Vibration monitoring is a specialized field, and most engineers defer to the recommendations of the machinery builder. There are three parameters that may be monitored: (1) acceleration, (2) velocity, and (3) shaft displacement. The choice depends on machinery characteristic dynamics such as speed, mass, and bearing type. In general, low-cost accelerometers utilizing a spring-suspended mass attached to a machine housing give a little protection. Elaborate electronic systems costing thousands of dollars may be able to provide a degree of warning of incipient failure, but nothing substitutes for periodic inspection by a competent operator.

Engine Driver Systems

Engine drivers of all sizes require extensive instrumentation (see Chapter 14). The instrumentation is usually furnished as part of the engine-control equipment. Electric or electronic sensors should be used on the engine itself with signals wired to a separate, vibration-free panel. Only sensors designed specifically for engine service should be mounted on the engine.

20-10. Control Equipment

Control equipment used in pumping stations has advanced from crude, motor-driven cam stacks to the compact, electronic, stored-program, microprocessor controllers used in the new designs. The older systems were readily understood by personnel accustomed to maintaining mechanical devices, but many

operators now can deal with modern control equipment. When there is a failure, either the device must be returned to the vendor or the vendor must come to the site to make repairs. This situation is unacceptable for equipment that must be kept operating. The overall reliability of modern solid-state controllers, however, is much better than that of the older mechanical devices, so modern devices are now frequently applied to pumping stations, even in locations far from skilled mechanics. If solid-state controllers are to be used, it is essential that emergency manual control systems be provided, that spare control devices be stocked by the operator, and that operators have or gain adequate skill levels to make repairs by replacing defective modules.

Motor Controllers

A motor controller is required for every motor in a pumping station. Article 430 of the National Electrical Code [2] establishes the minimum requirements for (1) motor controllers, (2) motor and branch-circuit overload protection, (3) motor branch-circuit, short-circuit, and ground-fault protection, (4) control circuit characteristics, and (5) disconnecting methods. Beyond these code requirements, a great number of variations in motor controllers are established by the characteristics of the power source and the requirements of the driven equipment.

Motor Branch-Circuit, Short-Circuit, and Ground-Fault Protection

Either a fused switch or an automatic circuit breaker is used for branch-circuit, short-circuit, and ground-fault protection. Fuses provide higher short-circuit interrupting capacity than circuit breakers, but they must be replaced after a circuit interruption. A single fuse may blow, which causes damaging single-phase power to be fed to a three-phase motor. Circuit breakers always open all conductors feeding a motor circuit and may be reset following a circuit interruption. Where the available short-circuit current exceeds the interrupting capacity of circuit breakers, fuses may be used in conjunction with the circuit breaker to provide the advantages of both. But this combination system is objectionable in terms of cost (30% added to the cost of the motor and starter), space requirements, and convenience. Although standard fuses are readily available, the special fuses used in conjunction with a circuit breaker may be difficult to obtain in an emergency. The type of short-circuit and ground-fault protection to be used is not a clear-cut decision, and the selection should be made in conjunction with the preferences of the facility operator.

Motor Starters

Motor starters typically provide motor branch-circuit overload protection as well as a means of starting the motor. Both magnetic and solid-state motor starters are available but, in 2005, solid-state units are still very expensive (they add about 50% to the cost of the motor and starter) and are seldom used except in special applications.

Full-Voltage, Nonreversing (FVNR) Magnetic Starters

The FVNR starter is the most commonly used type of magnetic starter. It connects and disconnects an electric motor with the power source in response to a relatively low energy control signal.

Reduced-Voltage, Nonreversing (RVNR) Starters

The RVNR starter must be used where the capacity of the electrical power source is limited in comparison with the size of motor. This is a common limitation in pumping stations located far from other plant facilities and in residential districts. Electric power utilities frequently stipulate that reduced-voltage starters be used under these conditions. Reduced-voltage starters are also helpful in getting a motor started on an emergency generator. The various types of RVNR starters, along with their advantages and disadvantages, are listed in Table 20-6 (see also Section 8-3).

Multispeed Motor Starters

Multispeed motors are sometimes used to provide two or more pumping capacities from a single pumping unit. Multispeed motors may be of the two-speed/one-winding, two-speed/two-winding, or four-speed/two-winding types, depending on the required speed ratios. The type of multispeed starter used depends on the type of motor to be started. The design of the control circuitry used with multispeed starters must ensure that the motor is deenergized for a sufficient time to allow the motor magnetic flux to decay during speed changes. If the time for decay is too short, extremely high transient torques may be generated

Table 20-6. Reduced-Voltage, Nonreversing (RVNR) Motor Starters

Starter type	Characteristics[a]
Autotransformer	Moderate cost, good starting current reduction, uses standard motors
Primary resistor	Moderate cost, poor starting current reduction, uses standard motors, not applicable to high-voltage (over 600 V) motors
Primary reactor	Same characteristics as primary-resistor starter, but may be used with high-voltage motors
Part winding	Low cost, poor starting current reduction, requires a special motor
Wye–delta	High cost, moderate starting current reduction, requires a special motor
Solid state	High cost, excellent starting current reduction, presently not applicable to high-voltage motors

[a]Moderate cost is 100%, low cost is 95%, and high cost is 140%.

when the motor is re-energized. Magnetic flux decay time varies from 0.5 s for small motors to over 1 s for large motors.

For adjustable-speed motor drives, refer to Chapter 15.

20-11. Control Logic

Control logic is required when equipment operates automatically. Although control logic is trivial in many small pumping stations, it can become quite complex in larger pumping stations with multiple pumping units and/or numerous auxiliary devices. For example, some pumping stations have several pumps of different sizes and require control logic to sequence them in a way that matches the pumping demand as closely as possible. Pumping stations discharging into long force mains frequently require power-operated check valves that are opened only after an oncoming pumping unit has been started (and are closed before the pump is stopped) to reduce surges that can occur when a pump suddenly starts or stops.

Control logic can be provided either by electromechanical devices, such as relays and drum programmers, or by solid-state devices, such as programmable logic controllers and microcomputers. Mechanical devices are usually more expensive for all but the simplest control systems, and the reliability of large numbers of mechanical devices in complex control systems is poor because dirt on a single relay contact can cause a malfunction. Solid-state systems have become very reliable if properly designed and installed, but they are more difficult to repair when a malfunction occurs.

Relays

Two types of relays are available: machine tool and miniature plug-in. Machine-tool relays are comparatively large and are capable of operating large motor starters. Because a large armature and contact mass must be moved quickly when a machine-tool relay operates, these relays are subject to mechanical failure in high duty-cycle applications. Miniature plug-in relays have a limited contact rating and are prone to early failure if applied to switching heavy loads. Miniature relays, however, may be quickly and easily replaced. Therefore, miniature plug-in relays should be used for logic level circuits and machine-tool relays should be used for switching large motor starters.

Timers

Most machine-tool timing relays provide time delays by forcing air through an adjustable orifice. Time delays of up to approximately one hour can be reliably obtained in this fashion, but the accuracy is not particularly good. Where longer time delays or precise timing is required, synchronous-motor-driven timers are used. Solid-state timers are also available in either large-frame or miniature plug-in configurations. Solid-state timers can provide high-accuracy timing with delays of many hours.

Drum Programmers

Complex sequential logic requires large numbers of relays. Drum programmers provide a much simpler method of accomplishing such logic. Drum programmers can be actuated by either solenoids or motors. Either type is suitable for low duty-cycle applications, but the motor-actuated type is more reliable in high duty-cycle applications. The cam stacks that are used in many existing pumping stations are a form of motor-driven drum programmer.

Programmable-Logic Controllers

Programmable-logic controllers (PLCs) were originally developed for the automotive and machine tool industries. These units are industrial-grade computers that have been designed to emulate conventional relay logic. Reliability and maintainability have been developed to a very high level. Programming can be accomplished by any competent control electrician, and repairs are facilitated by extensive diagnostic systems and extensive use of plug-in components. PLCs are less expensive and more reliable than discrete components for all but the simplest of logic systems, and they are greatly preferred for pumping station control systems if control logic is needed.

Microprocessor Controllers

Microprocessor controllers are similar to programmable-logic controllers, but (1) they require computer programming skills to alter their logic, and (2) they ordinarily lack diagnostic facilities and plug-in components. These controllers frequently appear in preprogrammed vendor packages.

This type of equipment is useful in applications such as constant-speed, multipump wastewater lift stations and multipump water system booster stations, but it should not be used for overall control of a pumping system unless the facility operator has, or is willing to develop, the requisite maintenance skills. When compared with programmable-logic controllers, the simpler design of a microprocessor controller does provide high performance at low cost.

20-12. Altitude Valves

Altitude valves (see Section 5-5) are sometimes used with water system storage to control pumps indirectly. The pumps are typically started by a time clock. When the storage tank is full the altitude valve closes, and a discharge pressure switch located at the pump detects an increase in system pressure and shuts the pump off. Because no electrical connections are required between the reservoir and the pumping station, this pump control is simple and economical.

The operation of pressure-controlled valves is explained in manufacturers' literature.

20-13. Monitoring and Data Acquisition

Providing information about the operation of pumping station equipment is particularly useful for large, unattended pumping stations. The simplest systems consist of only an event recorder (to record pump operation) or an annunciator with memory (to give visiting operators an indication of malfunctions). Telemetry systems may be added to transmit station status to an attended location for immediate indication of failure. Full supervisory control and data acquisition (SCADA) systems are capable of monitoring pumping station operation and controlling station operation from a distant location.

Annunciators

An annunciator monitors one or more discrete contacts and activates a warning light or alarm on malfunction. The addition of memory causes the annunciator to retain transitory alarms to inform a visiting operator of something gone wrong in the worker's absence. Recording annunciators provide a printed record of each alarm occurrence as well as the time, data, acknowledgment, and return to normal condition.

Recorders

Recorders produce a permanent record of analog or discrete variables on charts, which are frequently retained for many years. Flows and/or water levels are recorded in virtually all but the smallest pumping stations. The major disadvantages of recorders are as follows: (1) using information from recorder charts is tedious; (2) it takes considerable effort to keep pens working and charts replaced; and (3) the maintenance of recorders is particularly poor in many pumping stations. Recorders, therefore, should be sparingly used.

Telemetry

Transmitting data to an attended receiving station makes them more (and immediately) useful as compared with data stored in a remote pumping station. Telemetered alarms provide immediate warning of malfunctions. Centralized recorders are easier to maintain, and trends can be monitored continuously.

Communication channels are a major part of any telemetry system. They may be leased from common carriers (such as local telephone companies), or the pumping station utility may construct its own. Historically, the performance of leased facilities has been less than satisfactory. Data channels require the

utmost reliability—particularly during storms when leased facilities are least reliable. Leased channels are primarily designed for audio (voice) transmissions, which are not seriously impaired by intermittent noise. But noise is a severe problem for data transmissions. Consequently, dedicated communication facilities are frequently constructed even though they are almost always more expensive than leasing.

The various types of communication channels used for telemetry and SCADA systems are listed in Table 20-7. The type of telemetry equipment required depends on the type and amount of data to be transmitted and whether remote control capability is desired. In the simplest system, the data are simply displayed on annunciators, indicators, and recorders. In larger systems, video display terminals and data-logging types are frequently used.

Tone Equipment

Tone equipment is a system for modulating discrete contact signals into the voice-frequency range for transmission over voice-grade communication channels. It permits multiple signals to be transmitted over a single communication channel by using different carrier or center frequencies for different signals. By utilizing multipoint communication channels, this system is suitable for multiple signals from a single station and for multiple signals from different stations.

For best performance, the number of different carrier frequencies used on one channel should be limited to about 10. Two types of tone equipment are available: amplitude modulation (AM) and frequency-shift-keyed (FSK) modulation. Amplitude modulation has very poor resistance to communication channel interference and should never be used. Frequency-shift modulation has some resistance to communication channel interference, but is still generally unsatisfactory. Tone systems are ordinarily capable only of transmitting data in one direction—from the remote station(s) to the central station.

Digitally Encoded Tone Equipment

Tone equipment manufacturers have generally recognized the interference problem, and most now offer digital encoding systems that permit the tone equipment to reject errors. Digital encoding also permits several pieces of information to be transmitted using a single carrier frequency, which substantially increases the data capacity. With multipoint communication channels, one carrier frequency can be assigned to each station connected to the channel and, thus,

Table 20-7. Telemetry Communication Channels

Channel type	Characteristics
Voice-grade telephone	Lowest cost and very good, but reliability may be a problem in storms. Similar to conventional telephone service except that a dedicated circuit is leased; signals must be modulated into voice-frequency range of 300 to 3000 Hz; moderate amounts of data can be transmitted over unlimited distance. Channels can be “conditioned” to improve the performance, and with a high degree of conditioning, large amounts of data are transmittable. Multipoint configurations permit several pumping stations to be connected to the same channel, but telemetry equipment must be able to separate the station signals.
VHF and UHF radio	Best system if a license can be obtained. Voice-grade communication channels functionally similar to voice-grade telephone; VHF radio frequencies are virtually all taken, but UHF frequencies (especially 928 to 952 megahertz) are still available for water systems (although not for wastewater systems) in many areas. Vulnerable to hackers.
Microwave radio	Signals bounce off temperature inversion layer, but by using both high and low antennas (the principle of space diversity) signals go through temperature inversions 99% of the time but at twice the cost. Microwave-radio frequencies are available, but cost and transmission capability are beyond most budgets and needs. Birds may cause spurious signals (but not if the system is properly designed for error rejection). Vulnerable to hackers.
Underground cables	Use pipeline easements for underground cables; such cables may be either buried directly or in conduits. Communication is reliable and of high quality, but initial costs are very high.
Overhead cables	Practical either on leased pole space or on privately owned poles, but, because they are subject to damage by storms and automobiles, reliability is inadequate.
Fiber optics	Impractical; very expensive to repair damage; suited to transmitting huge quantities of data in infinitesimal time periods—completely unnecessary for pumping stations.

permits about 10 stations to be connected to the channel.

20-14. Telemetry

Introduction

Providing information about the operation of pumping station equipment is extremely important for unattended pumping stations. Telemetry means measuring at a distance. Telemetry systems make remote information available at a central facility. For example, one telemetry device, located at a pumping station, interfaces with instruments that measure and control level, pressure, flow, power, etc. The other telemetry device is located at a central monitoring point (such as the main treatment plant which is usually manned 24 hr/d) where events and equipment conditions are monitored by display panels, or instruments such as data loggers, historians, status indicators, chart recorders, and annunciators. In addition, the facility can be provided with the capability for manual control of the remote equipment. On a telemetry system with multiple remote sites, data can be exchanged between remote stations (peer-to-peer), directly from remote to master (slave-master), one to many (point-to-point), or one to one (point-to-point).

A telemetry or supervisory control and data acquisition (SCADA) system, designed properly, can save time and money by eliminating the need for operating and service personnel to visit each site for inspection, data collection/logging or make adjustments whenever the logging or maintenance service schedule (daily, weekly, etc.) demands. Real-time monitoring, system modifications, troubleshooting, increased equipment life, automatic report generating, are just a few of the benefits. Telemetry and SCADA systems are terms often used interchangeably and in today's technology there is technically very little difference between them.

The simplest monitoring system at a small facility may consist of only an event recorder (to record pump operation) or an annunciator with memory (to give visiting operators an indication of malfunctions). A telemetry system may be added to transmit station status to an attended location for immediate indication of failure. SCADA systems are designed to integrate the monitoring of various system operations into a central data display and processing center so that the operating staff can monitor, control, and evaluate the system in real time. SCADA systems can also be programmed to respond to changes in system parameters, either by performing automatic operation or by alerting operators so they can respond manually.

SCADA systems integrate data aquisition systems with data transmission and graphic presentation. SCADA systems today can include security and safety systems and integrate data from cameras, intrusion detectors, etc. as well as laboratory analysis systems. SCADA systems can not only control and monitor processes but can also be used for measuring, forecasting, billing, analyzing, and planning.

The type of telemetry system to be employed will depend on the following:

- System Architecture
- Network Communications
- System Hardware
- Software

Acronyms

The following acronyms are used in this Section:

Acronym	Definition
ACL	Access Control List
ATM	Asynchronous Transfer Mode
bps	Bits per second
CATV	Community Access Television
CDPD	Cellular Digital Packet Data
DCS	Distributed Control System
DDS	Digital Data Service
DSL	Digital Subscriber Line
FCC	Federal Communications Commission (USA)
GEO	Geosynchronous Orbit (Satellite)
Ghz	Gigahertz (frequency)
HMI	Human-Machine Interface
IO	Input/Output
ISDN	Integrated Service Digital Network
ISM	Industrial, Scientific, and Medical
Kbps	Kilo bits per second
LAN	Local Area Network
LEO	Low-Earth Orbit (Satellite)
mA	milliamperes
MHz	Megahertz (frequency)
MTU	Master Terminal Unit
PLC	Programmable Logic Controller
RS485	IT Standard. Communications language used by computers to talk to multiple devices
RTU	Remote Terminal Unit
SCADA	Supervisory Control and Data Acquisition

UHF	Ultra High Frequency (300 to 3000 MHz).
VHF	Very High Frequency (30 to 300 MHz)

System Architecture

System architecture is highly dependent upon the following factors:

- Economic (how much first cost and maintenance budget, etc.);
- Geographic (distance between stations, line of site, etc.);
- Quantity (how many stations);
- Data requirements (what data and which station receives and which station sends);
- Poll time (how frequently does the data need to be updated)
- Reliability (redundancy, backup features, parallel routes, etc).
- Intended function (what you want to do with the data and who needs to access it)

Systems similar to SCADA systems are routinely seen in large pumping facilities, treatment plants, etc. and are often referred to as Distributed Control Systems (DCS). They have similar functions to SCADA systems, but the field data gathering or control units are usually located within a more confined area. Communications may be via a local area network (LAN), and are normally reliable and high speed. SCADA systems on the other hand generally cover larger geographic areas, (more suited to pumping station applications) and rely on a variety of communications systems that are normally less reliable than a DCS.

SCADA Components

A SCADA system consists of four main components. These components are:

1. The central SCADA Master Terminal Unit (MTU) and software.
2. The Remote Terminal Unit (RTU),
3. The field instrumentation and control equipment, and
4. The communications network.

Remote Terminal Unit (RTU)

The SCADA RTU is usually small environmentally-hardened computer or programmable logic controller (PLC) that provides intelligence in the field, and allows the central SCADA master to communicate with the field instruments and to control equipment. The RTU is a stand-alone data acquisition and control unit. Its function is to control process equipment at the remote site, acquire data from the equipment, and transfer the data back to the central SCADA system. The RTU scans its inputs at a relatively fast rate. It then processes changes of state, timestamps them, and stores them in memory awaiting request from the SCADA master. The RTU provides data on a "polled" request by the SCADA master, which may be polling several remote stations within the SCADA system. Polling time is defined as the time interval frequency that data is interrogated at the remote site and is very important with regard to data update. There is not much vital information at a pumping station that would require a fast polling rate. SCADA systems become more economical, in terms of communication demands as the polling rate diminishes.

Field Instruments and Control Equipment

Telemetry devices' inputs/outputs connect to field sensing instruments and control equipment that provide digital (on-off contact closure) inputs, such as float or pressure switches, or analog (measurements) input, such as flowmeter rate sensors and thermostats. Some sensing instruments output a series of pulses (e.g. pulse-type flow rate sensors) where each timed pulse represents a multiple of a measured quantity. The digital inputs on a telemetry device can be configured to count these pulses.

When a remote telemetry device detects a change in any of its active analog or digital inputs, it transmits the data back to the central telemetry device for monitoring. Using the telemetry devices as a communications path, the monitoring instruments can send control commands back to the remote equipment to make simple changes (on/off) in the equipment status. Data are exchanged between multiple telemetry devices the same as if wires connected the devices.

Communications Network

The communication network consists of the communication channel(s), transmitting and receiving hardware, and the communication protocol (software).

The communication channel is the medium over which communications travels. Data transmission can be divided into two distinct categories — land line or wireless. Land line systems are physically connected to each other through wires or fiber optic cables. Land line systems are comprised of telephone,

coaxial cable, or optical fiber. Wireless systems are not physically connected, but instead, transmission takes place from the transmitter to the receiver over radio wave, satellite, laser, or microwave frequencies. Wireless systems are comprised of VHF and UHF radio, ISM-band radio, microwave, satellite, and cellular telephone.

Telemetry Communication Channels

- Telephone
  - Voice-grade telephone line, dial-up modems
  - Digital Data Service (DDS), with Digital Subscriber Line (DSL) and Integrated Service Digital Network (ISDN), and others. DDS communicates in multiples of 56 Kbps. This digital leased line, is commonly available, and is a good choice for most pumping station applications;
  - Frame relay (too expensive)
  - Asynchronous Transfer Mode (ATM), is a high speed cell-switching technology that is expensive and "over-kill" for pumping station SCADA applications;
  - Analog, 20 mA, dial-up modem. Auto-dialer. Analog telephone lines (increasingly difficult to obtain from the local telephone company because of limited frequency allocation) are limited in bandwidth and have slow rates of 1200, 4800, or 9600 bits per second (bps).
  - Data-grade, leased telephone line, T1 etc. T-1 is a high-speed digital connection ideal for most LAN/Ethernet connections, "overkill" for most pumping station SCADA applications.
  - Twisted pair metallic cable. Telephone or RS485. Private line, either underground (run with pipeline) or overhead (lease utility pole space. Telephone leased lines (run on telephone company overhead poles) are susceptible to storms.
- Coaxial Cable
  - Community Access Television (CATV). Cable leased through CATV /Internet provider. The speed is comparable with DSL.
- Optical Fiber
  - Local government or Utility Local Area Network (LAN)
  - Private line, usually run along with pipeline
  - Provide large data transfer (bandwidth).
  - Cost for fiber systems has declined by half over the last decade due to popularity. Cost effective only for very large stations, but may be economical if fibers can be shared with others.
- Very High and Ultra High Frequency (VHF and UHF) Radio. Privately-owed radio system commonly used where terrain and lack of land line phone service is a problem. Good for up to 30 miles. FCC license required and may be difficult to obtain in congested areas. If license already obtained, may be beneficial to expand system using existing frequency allocations.
- Microwave. Microwave radio transmits at high frequencies through parabolic dishes mounted on towers. A FCC license is required and microwave radio requires larger power supplies than other wireless options. Microwave is the most secure and reliable of the wireless systems but atmospheric interference can be a problem.
- Industrial, Scientific, and Medical (ISM) band Radio. ISM band radio is a FCC license-free frequency band. FCC rules require built-in interference avoidance. See description under Spread-Spectrum under ***Examples of SCADA systems***.
- Satellite. There are two types of satellite systems that offer SCADA-type applications: Low-Earth Orbit (LEO) and Geosynchronous Orbit (GEO). LEOs come in two sizes. The larger size is used mainly for high-speed, high-bandwidth data communications, and video conferencing. Small size LEOs are used for non-voice services such as vehicle tracking, environmental monitoring and two-way data communication. For situations where other systems do not provide communication, small LEOs may be the best choice. Third-party service provider is required to route the signal. A GEO satellite's primary purpose is weather imagery to optimize forecasting.
- Cellular. Originally designed for voice communication, Cellular Digital Packet Data (CDPD) uses the voice channel, but sends and receives data over the existing cellular infrastructure. Analog and digital mobiles use a network of base stations and antennas to cover a large area. The area a base station covers is called a cell; the location where the base station and antennas are placed is called a cell site. Cell sizes range from sixth tenths of a mile to thirty miles in radius. CDPD is a packet-switched network sharing the radio channel and there is no connection time charge. CDPD usage is measured in terms of the amount of data transferred, rather than how long the user is logged onto the network. An FCC license is not required but a monthly service fee is charged for the service. Coverage is restricted to cell phone coverage area but cell phone service provider may change cell towers or communication protocol, thereby effecting communications to the remote location.

Most radio transmission (with the exception of cellular, see description) requires line-of sight between the telemetry devices. Repeater stations may have to be used. Antenna positioning, gain, tuning, ambient frequency, noise, atmospheric conditions and terrain are important variables affecting range but transmissions can be repeated if interference is present.

Central SCADA Master Terminal Unit and Software

Master Terminal Unit (MTU). The MTU can be a dedicated computer, a network server or a PLC that communicates with the remote site RTUs and provides a Human-Machine Interface (HMI) to the SCADA system. Master SCADA systems usually offer high resolution computer graphics to display a graphical user interface or mimic the screen of the individual sites or the entire network. Operator workstations are connected to the host server by LAN. These workstations displays can serve as the graphic interface for the entire SCADA system using single or multiple screens to display each station. Other workstations may be dedicated to alarm displays, while others may be dedicated to trend displays. On small systems a single workstation may combine all these functions. To preserve data for archival purposes, the MTU needs to interface with large memory and retrieval hardware, called an historian. Data are time-stamped at the RTU and stored in the historian for future retrieval, reporting, or trending. Some of the main functions of the master SCADA station are

- Real-time data interface
- Graphic interface
- Data logging
- Alarm processing
- Trending and reporting capabilities, both active and historical
- Security system

Software. An important aspect of every SCADA system is the computer software used within the system. The most obvious software component is the operator interface or HMI package; however, software of some form pervades all levels of a SCADA system.

Many SCADA systems employ commercial proprietary software upon which the SCADA system is developed. At one time proprietary software often was configured for a specific hardware platform and may not have co-existed with the software or hardware produced by competing vendors, but this situation is no longer true.

Software products typically used within a SCADA system are as follows:

- Central host computer operating system: Software used to control the central host computer hardware. The software can be based on Windows, UNIX or Linux.
- Communications network management or configuration software: Software required to control the communications network and to allow the communications networks to be monitored for performance and failures.
- Graphic generation software for HMI displays.
- Database management and report generation.

Central Station Auxiliary Equipment. Along with telephone dialers, report printers, alarm printers, historians, and backup power supplies, conventional "hard data" equipment may be needed for central-plant operators as well as to supply real-time reports to local, state, and federal agencies.

- Annunciators. An annunciator monitors one or more discrete contacts or via serial interface and activates a warning light or alarm on malfunction. The addition of memory causes the annunciator to retain transitory alarms to inform a visiting operator of something gone wrong in the worker's absence. Recording annunciators print a record of each alarm as well as the time, data, acknowledgment, and time of return to normal condition of each occurrence.
- Recorders. Recorders produce a permanent record of analog or discrete variables on charts, which are frequently retained for many years. Flows and/or water levels are recorded in virtually all but the least-critical pumping stations. The major disadvantages of recorders are as follows: (1) using information from recorder charts is tedious, (2) it takes considerable effort to keep pens working and charts replaced, and (3) the maintenance of recorders is particularly poor in many pumping stations. Pen recorders are all but obsolete and have been superseded by electronic paper-less recorders.

Examples of SCADA systems

The final configuration of a SCADA system is dependent upon the factors aforementioned. SCADA systems most suited for remote pumping applications are:

- Spread-Spectrum Radio. Spread spectrum is a radio technology whereby numerous carrier frequencies are used. The sending radio jumps from frequency to frequency in a preordained pattern. This frequency hopping allows other users of spread spectrum to transmit at the same time using a different pattern of frequency hopping. Spread Spectrum radio uses ISM frequency bands, which are unlicensed. ISM rules specify that Spread Spectrum technology must be used to avoid interference with other transmissions. Additionally, these rules specify maximum power to be transmitted in-band and out-of-band (to avoid interference with adjacent bands), and how channels are defined. FCC allocates both the 902-928 MHz, 2.4, and 5.3 GHz band with low power (1 W maximum) assignments. Spread-Spectrum radios have a range of about 5 miles, line-of-sight, but some models have the ability to re-strengthen signals for the next radio in line and some models profess "extended ranges". These "repeater" radios are used to span distances and generally have built-in error correction, encryption and other features, making them a reliable, low-cost, secure and long-lasting solution for network communication.
- Voice-grade telephone line with Internet service. Using DSL service with compatible DSL modems at each end.
- Digital leased-line using DSL/ISDN service.
- Voice-grade telephone with dial-up service and modems.
- Leased-line T-1 with Ethernet. High speed connection directly between MTU and RTU.

Security

SCADA systems carry control signals and data that affect control action, either automatic or manually instigated. For these reasons system security is an important consideration. The primary security on a SCADA system can be applied at three levels: system network; software; and communication channel media.

System Network

- Password system. When the computer password system is properly used, the system is much more difficult to exploit. The system is similar to that used for a regular desktop computer, and it allows the system administrator to set up private passwords for each system user to allow certain predetermined activities or functions. The password system may also allow the administrator to require a regular renewal or change of passwords.
- Authentication. This Authentication is the process of accepting credentials from a user and validating those credentials against a designated authority. After the identity is known, the application can authorize the user to access resources on the system.
- Authorization. This Authorization is the process of determining whether the proven identity is allowed to access a specific resource. Establishment of an appropriate Access Control Lists (ACLs), to prevent intruders from accessing secured resources. The appropriate ACLs must be set on the appropriate resources to allow access by only the relevant users i.e. operators, administrators, programmers.
- Access. Locating SCADA server and equipment in a secure room.
- Trace log-ins. The availability of trace logs allows allow a system administrator to determine who has logged on and used the system, what they did, and when they did it.
- Secure data transmission technology. Secure options are available to help ensure that data transmissions are neither blocked nor intercepted. These include options for radio transmissions, as well as options for the local telephone network e.g. firewalls.
- Automated alarm tracking/monitoring options. Some SCADA systems offer additional alarm pager options that can send an alphanumeric page to the operator on duty when systems or operations alarms are triggered.
- Backup power supply if the primary source is compromised.

Software

The operating program should include built-in security features to disable computer viruses before they embed themselves into the program. Avoid operating systems such as Windows 2000 and earlier versions that have vulnerability. Windows XP should have service pack 2 plus the latest Microsoft updates. The addition of a software firewall is advised. The SCADA software should be similarly protected. Log on and user authentication, authorization, and access control should be featured.

Communication Channel Media

Microwave. Microwave signals are broadcast through the air, making them more accessible to

outsiders. Typically, a transmitter's signal is focused on its corresponding receiver. The signal path is fairly wide to be sure of hitting the receiver. Hackers intercept a microwave transmission by interfering with the line of sight between sender and receiver, or they can pick up an entire transmission from an antenna located close to but slightly off the direct focus point.

A microwave signal is usually neither shielded nor isolated to prevent interception. Microwave is, therefore, an insecure medium but because of the large volume of traffic carried by microwave links, it is unlikely that hackers will be able to separate an individual transmission from all the others interleaved with it. Microwave is therefore the most secure of all the wireless technologies.

Optical Fiber. Optical fiber offers two significant security advantages over other transmission media. First, the entire optical network must be tuned carefully each time a new connection is made. Therefore, no one can tap an optical system without detection. Second, optical fiber carries light energy, not electricity. Light does not emit a magnetic field; therefore, an inductive tap is impossible on an optical fiber cable. Fiber optic ancillary equipment, such as repeaters, splices, and taps along a cable are places at which data may be available more easily than in the fiber cable itself. Fiber as a system is much more secure than telephone cable or wireless transmissions.

Wireless. Wireless networking is vulnerable because of its ability to be received in a location several miles from its intended reception. A strong signal can be picked up easily using an inexpensive, tuned antenna. Transmission encryption is now standard on most systems to make signal deciphering more difficult.

Supervisory Control and Data Acquisition (SCADA) Systems

Supervisory control, a logical progression from simple telemetry, infers that data can be transmitted not only from the remote station to the central station but also from the central station to the remote station. Originally, SCADA systems were implemented with miniature mechanical relays for transmitting data over direct-current telephone lines to control and monitor remote gas and electric utility systems. Modern SCADA systems utilize microcomputer technology and voice-grade communication channels. The development of the "personal computer," which is used as the master station, has reduced the cost of very sophisticated SCADA systems to a relatively low figure. Remote terminals may be implemented either with programmable controllers or with microprocessor controllers. In either, it is practical to provide both (1) local automatic control at the remote station with the SCADA equipment, and (2) a method of overriding the local controls from the master station.

20-15. Design Considerations

The performance of an instrumentation and control system can be affected more by the way the equipment is installed than by any other factor. Manufacturer installation and maintenance manuals provide extensive information about how an instrument should be installed. The API RP550 [10] is an excellent guide for the installation of many instruments used in pumping stations.

Location and Access

Instruments and control devices require special maintenance. The location of process sensors is generally dictated by process considerations, but considerable flexibility is available in the location of other instrument and control devices. Wherever possible, process sensors should be installed so that they can be removed without draining pipelines. Bubble pipes should be installed so that they can be easily rodded. Magnetic flow tubes should be installed so that the electrodes can be wiped off without removing the flow tube from the pipeline. Sufficient flexibility must be designed into pipelines so that in-line devices can be removed without having to disassemble whole piping systems. If possible, valving and connections should be provided to permit in-place instrument calibration.

Protective Enclosures

Some instruments are designed for installation under field conditions and adverse environments. Many instruments and control devices, however, are designed for installation only in a clean, dry, air-conditioned environment. If such an environment does not exist in the pumping station, install a suitable protective enclosure for these instruments and, if necessary, use forced ventilation or even air conditioning in instrument enclosures. Because many electronic control instruments designed for process industries require rear

access for electrical connections, give careful consideration not only to panel design but also to the ready access needed.

Equipment Handling

Instruments and control devices are fragile, and some (e.g., magnetic flow tubes) are also heavy. Provide a method of removing and installing instrumentation and control equipment that permits the weight to be handled easily. These methods frequently entail special access hatches or doors and permanently installed hoisting beams or brackets.

Safety

Most modern instrumentation and control systems are electrically operated and must operate continuously, so they cannot be shut down for maintenance. Use low voltages (24 V dc) in instrumentation and control systems as much as possible. Make it easy to disconnect higher voltages quickly and safely from individual instruments for servicing. Use plug-in connections wherever possible so that the instrument can be easily replaced by a spare and taken to a shop for servicing.

20-16. References

1. Lipták, B. G., *Instrument Engineers' Handbook, Process Measurement,* Chilton Book Company, Radnor, PA (1982).
2. NEC, *National Electric Code,* National Fire Protection Association, Boston (updated periodically).
3. ASME MFC-3M, "Measurement of fluid flow in pipes using orifice, nozzle, and Venturi," American Society of Mechanical Engineers, New York (1985).
4. Miller, R. W., *Flow Measurement Engineering Handbook,* 2nd ed., McGraw-Hill, New York (1989).
5. Moorman, P., "Flow meters for water and wastewater treatment plants," *WaterWorld Review,* pp. 25–27 (Jan/Feb 1994).
6. Starrett, P. S., H. B. Nottage, and P. F. Halfpenny, "Survey of information concerning the effects of non-standard approach conditions upon orifice and venturi meters," 65-WA/FM-5, presented at the winter annual meeting of the American Society of Mechanical Engineers, Chicago, IL (November 7–11, 1965).
7. Shelley, P. E., and G. A. Kirkpatrick, "Sewer flow measurements—A state-of-the-art assessment," EPA-600/ 2-75-027 (November 1975).
8. MGD Technologies, San Diego, CA (see www.mgdinc.com).
9. White, G. C., *Handbook of Chlorination,* 2nd ed., Van Nostrand Reinhold, New York (1986).
10. API RP550, *Manual on Installation of Refinery Instruments and Control Systems,* Part 1, Sections 1–13, and Part 11, Sections 7–10, American Petroleum Institute, Washington, DC (revised periodically).

20-17. Supplementary Reading

1. Bean, H. S., *Fluid Meters,* American Society of Mechanical Engineers, New York (1971).
2. Considine, D. M., *Process Instruments and Controls Handbook,* McGraw-Hill, New York (1974).
3. Hayward, A. T. J., *Flowmeters,* Macmillan, New York (1979).
4. Shinskey, F. G., *Process Control Systems,* McGraw-Hill, New York (1979).
5. U.S. Bureau of Reclamation, *Water Measurement Manual,* Government Printing Office, Washington, DC (1971).

Chapter 21

Instrumentation and Control Applications

ROBERT S. BENFELL

CONTRIBUTORS
Mead Bradner
Horton Wasserman

Pumping stations of different types and sizes have very different instrumentation requirements. Large utilities with many facilities to operate and maintain generally need more extensive instrumentation than smaller utilities, partly because smaller utilities usually lack the staff specialists necessary to maintain instruments. Several typical pumping station configurations and the instrumentation and control devices that might be required for that type of station are considered in this chapter.

21-1. Process and Instrumentation Diagrams

Control systems are frequently depicted on process and instrumentation diagrams (P&IDs). The level of detail shown on P&IDs varies considerably with their purpose. The figures included in this chapter are very simplified and show only the pumping station main process flow and the location of major monitoring instruments. No attempt is made to show control equipment. The following balloon symbols (circles) appear on the figures, as exemplified in Figures 21-1 and 21-2:

A, B	Force main A, Force main B
AE	Analytical element
ARC	Analytical recorder/controller
FC	Flow controller
FE	Flow element (such as a magnetic flow tube or Venturi meter)
FIQ	Flowmeter with instantaneous and totalized indication
FQ	Flow integrator
FQR	Flow recorder with totalizer
FR	Flow recorder
FT	Flow transmitter
HS	Hand switch
KS	Timer or time clock
LAH	Level alarm high
LAHH	Level alarm high–high (back-up alarm)
LAL	Level alarm low
LC	Level controller
LCV	Level control valve (altitude valve)
LG	Level gauge
LR	Level recorder
LS	Level switch
LSH	Level switch high
LSHH	Level switch high–high (back-up alarm)
LSL	Level switch low
LSLL	Level switch low–low (interlock to shut down pumps)
LT	Level transmitter
M	Motor
PC	Pressure controller
PI	Pressure indicator or gauge
PSH	High-pressure switch

PSL	Low-pressure switch
PT	Pressure transmitter
SS	Speed switch
ZS	Position or limit switch
○	Field mounted
⊖	Mounted on front of panel
⊖	Mounted inside or behind panel

Note that a transmitter (such as FT and PT) may be designated as equipped with a local indicator by inserting the letter "I" in the symbol: FIT and PIT. The order in which output functions (such as Q or I) are listed is not important.

Other acronyms (not used in balloons) include

AS	Air supply
ASD	Adjustable-speed drive
CNTL	Control
MAG	Magnetic
P	Purge
P&ID	Process and instrumentation diagram
PLC	Programmable logic controller
S	Solenoid-operated
VSD	Variable-speed drive

The balloon symbol as used on P&IDs represents the function of an instrument, not the actual hardware used to perform the function. Thus, a physical pressure switch might be used to measure a level and (because the primary purpose takes precedence) would have LS as the functional symbol. The distinction between a switch and a controller is also important. A switch is a device the normal function of which is to detect an abnormal condition. A controller may be the same physical device, but its normal function is actual control. Thus, a pressure switch used to actuate a low-pressure alarm has the function symbol PSL; a pressure switch used to start and stop a pump in order to maintain pressure in the system has the functional symbol PC.

21-2. Well Pump with Hydropneumatic Tank

A well pump with a hydropneumatic tank (shown in Figure 21-1) is common in small water systems—frequently with a single well.

Pumping Unit Control

While there are differing theories about how to operate a hydropneumatic tank, the one proposed here ensures trouble-free operation. The maximum water level in the tank is typically set at one-third to one-half full to provide an adequate air-pad volume for reasonably constant-pressure operation, and air is added or vented as required only when the tank is at or close to the maximum water level. The operating sequence is as follows:

- As water is drawn from the tank, pressure (as measured by PSL) will fall, which causes the well pump to start and refill the tank.
- When the water level reaches the maximum operating level (LC), the pump is stopped.

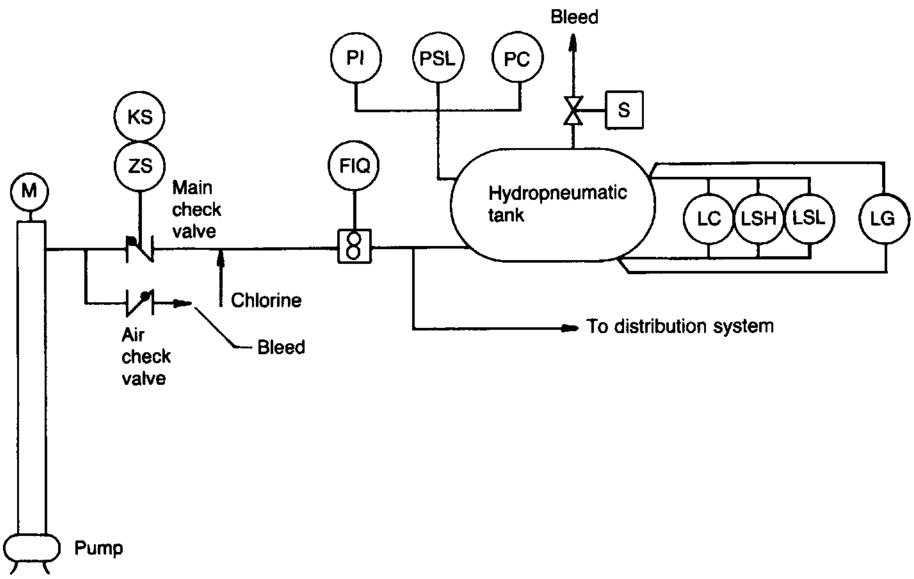


Figure 21-1. Well site with a hydropneumatic tank.

- Air is added or vented as necessary to adjust tank pressure (PC) to the maximum desired value.

In those installations where adequate air volume can be trapped in the well pump discharge piping, provision is required only for air venting. Because excess air can be vented easily and very quickly, the controls are quite simple. In installations where air must be added, an extra level switch is required to prevent the addition of air (which would air bind the tank) as the tank water level falls. Level switches on hydropneumatic tanks should always be installed in an external "cage" with isolating valves between the cage and the hydropneumatic tank so that the switches can be serviced without venting the tank.

In deep well applications, provide positive interlocking to eliminate any possibility of restarting a pump during backspin. A limit switch (ZS) on the check valve and time-delay relay (KS) or a zero speed switch can provide this protection. Also, flow detectors are sometimes installed on the intermediate shaft lubrication system to prevent the pump from being started before lubrication is established.

Chlorination Control

In most well pumping stations, chlorination equipment is operated at a constant rate. The chlorine injector pump is simply turned on at the same time that the well pump is turned on. At well sites that have multiple well pumps, step-rate control can provide different chlorine feed rates for different pump combinations.

Alarms

The following malfunction-monitoring devices and alarms may be furnished:

- Tank pressure low (PSL)
- Tank water level high (LSH)
- Tank water level low (LSL)
- Shaft lubrication failure
- Well water level low
- Pump discharge flow failure (ZS)
- Intrusion
- Chlorine leak (if applicable)
- Power failure.

Other Monitors

A totalizing water flowmeter (FIQ) is desirable on all wells and is frequently mandated by regulatory agencies. A pressure gauge (PI) and a level gauge (LG) should be furnished on the hydropneumatic tank for monitoring control system operation. Note that the level gauge is connected to the hydropneumatic tank separately from the switch cage so that it can remain in service if the switch cage is isolated.

21-3. Booster Stations

Booster stations are commonly used to lift water from one distribution zone to a higher distribution zone. The configuration shown in Figure 21-2 is controlled by the level in the zone 2 reservoir, but

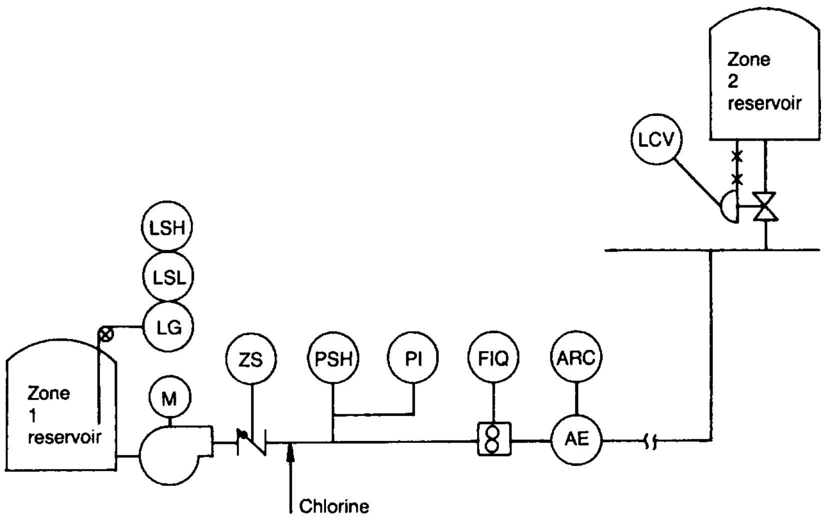


Figure 21-2. Booster pumping station.

requires no telemetry between the pumping station at the zone 1 reservoir and the zone 2 reservoir.

Pumping Unit Control

The booster pump(s) is started by a time clock—in the evening, if possible, to take advantage of off-peak power rates. Water enters the zone 2 reservoir until the reservoir is full and the altitude valve (LCV) closes. This mode of operation causes the discharge pressure on the pump as measured by a high-pressure switch (PSH) to increase and shut down the pump.

For such a system to operate properly, the booster pump(s) must be of significantly greater capacity than system demand and the zone 2 storage reservoir must have ample capacity to supply the distribution system until the time clock initiates the next pumping cycle. With this type of system, it is frequently necessary to start the booster pump(s) manually or reset the time clock during high demand periods. It may also be necessary to start the pumps manually in case of unusual demand, such as could be caused by a fire.

A level gauge with low level switch (LG/LSL) is furnished on the zone 1 reservoir to shut down the pump(s) in case there is insufficient water in zone 1.

Chlorination Control

Where the zone 1 reservoir is large, water may remain in the reservoir for a long enough period to lose some or all of its chlorine residual. In this situation, chlorination equipment must be installed, and residual control is recommended because the residual chlorine in the water drawn from the reservoir is unpredictable.

A water sample from the discharge of the booster pump is passed through the residual analyzer/controller (AE/ARC), which compares the actual residual with the desired residual and adjusts the chlorine feeder until the difference between the actual and desired residuals is essentially zero.

Alarms

The following malfunction-monitoring devices and alarms may be furnished:

- Zone 1 reservoir level high (LSH)
- Zone 1 reservoir level low (LSL)
- Pump discharge flow failure (ZS)
- Chlorine leak (if applicable)
- Intrusion
- Power failure.

Other Monitors

A totalizing flowmeter (FIQ) is frequently furnished to record the amount of water transferred between zones, and a pressure gauge (PI) used in conjunction with the flowmeter can indicate any changes in pump performance. A chlorine residual recorder is shown as part of the chlorine residual controller.

21-4. High-Service Pumping Station

Ideally, storage reservoirs would be built for all service zones. In actual practice, however, many service zones do not have storage reservoirs, and the pumps supplying such zones must operate as necessary to furnish instantaneous demand. Multiple pumps are frequently used for this service, and they may either be selected to have a relatively flat head versus flow curve or be operated with a variable-speed drive to maintain constant pressure. A common arrangement is to have one small, constant-speed pumping unit to maintain pressure during low flow periods and two or more larger, variable-speed units (as shown in Figure 21-3) for normal or high demand.

Pumping Unit Control

The following control strategy might be used. The small pumping unit runs constantly until the pressure falls below a minimum set point as measured by an electronic trip (PC1) connected to a pressure transmitter (PT). A large pumping unit is then started and the small unit is shut down. A second electronic trip (PC2), which is set at a slightly lower pressure, is used to start the second large pumping unit if the first unit fails to maintain the required pressure. The speed of the large pumping units is controlled by the PIC, which compares the desired pressure with the actual pressure and adjusts the variable-speed drives as necessary. Because the large units simply slow down when water demand decreases, pressure cannot be used to shut the large units down when less pumping capacity is required. Therefore, electronic trips (FC1 and FC2) connected to the flow transmitter (FT) must be used to shut down the large units when demand can be safely handled by fewer pumps.

Chlorination Control

Chlorine demand varies over a wide range in this application, and compound loop control is recom-

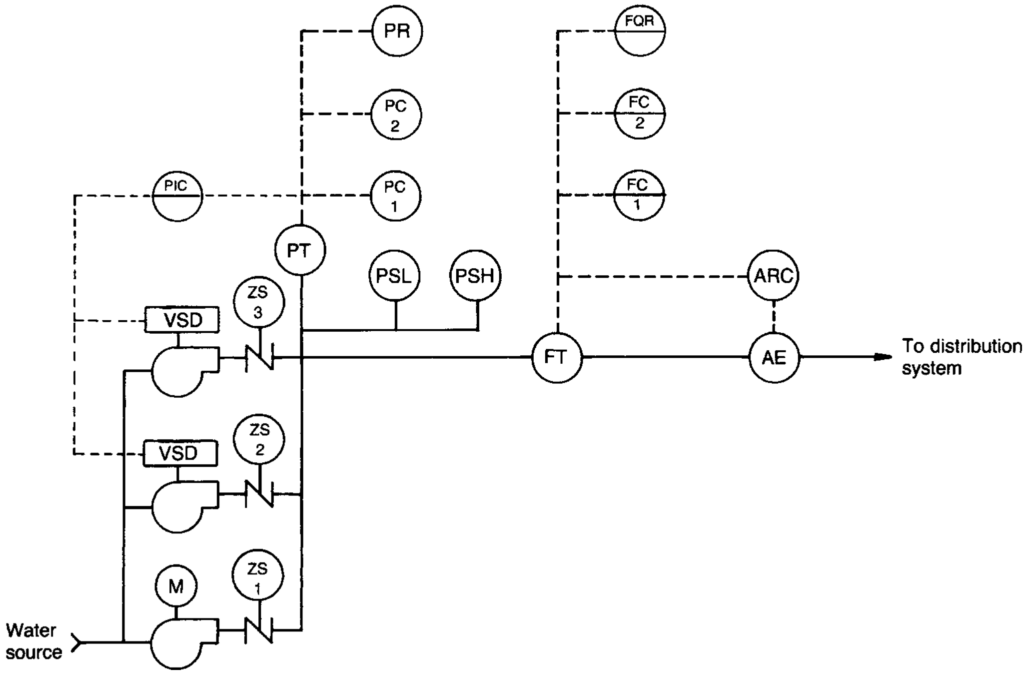


Figure 21-3. A distribution service pumping station.

mended. The output of the chlorine residual analyzer/controller (ARC) is multiplied by the flow to determine the chlorine feed rate. The action of the chlorine residual controller is the same as described previously for booster stations. By multiplying the controller output by flow, the feed-rate corrections are automatically made small at low flow rates and larger at higher feed rates.

Alarms

The following malfunction monitoring devices and alarms may be furnished. Note that the electronically transmitted pressure signals used to control the pumps are not used for alarms. Separate pressure switches are used to protect against the failure of the pressure transmitter or its associated electrical circuitry.

- Discharge pressure low (PSL)
- Discharge pressure high (PSH)
- Pump discharge flow failure (ZS1, ZS2, or ZS3)
- Loss of water source (ZS1, ZS2, and ZS3)
- Intrusion
- Chlorine leak
- Power failure
- Variable-speed drive malfunction (produced by the drive controller).

Other Monitors

A pressure recorder (PR) and flow recorder with an integral totalizer (FQR) are shown in Figure 21-3. A chlorine residual recorder is shown as part of the residual controller.

21-5. Small Wastewater Lift Station

Small lift stations and drainage sumps within larger facilities typically consist of a pair of submersible pumps—one duty and one standby. Such a station can be monitored and controlled by four float switches, as shown in Figure 21-4.

With an increasing water level, the duty pump is set to start at about 80% of the available fill/draw range (LC-B). The follow pump is started at 90% of the available fill/draw range (LC-C). A high-level alarm (LAH) is generated by the controller any time the follow pump is called. The high-level alarm should be electrically locked such that it can be reset only at the pumping station (HS-A). Both pumps are stopped at 0% of the available fill/draw range (LC-A).

Provide a high–high level alarm at 100% of the available fill/draw range (LSHH). The float should be set at such an elevation that it will not be fouled by floating debris in normal operation. A duty pump

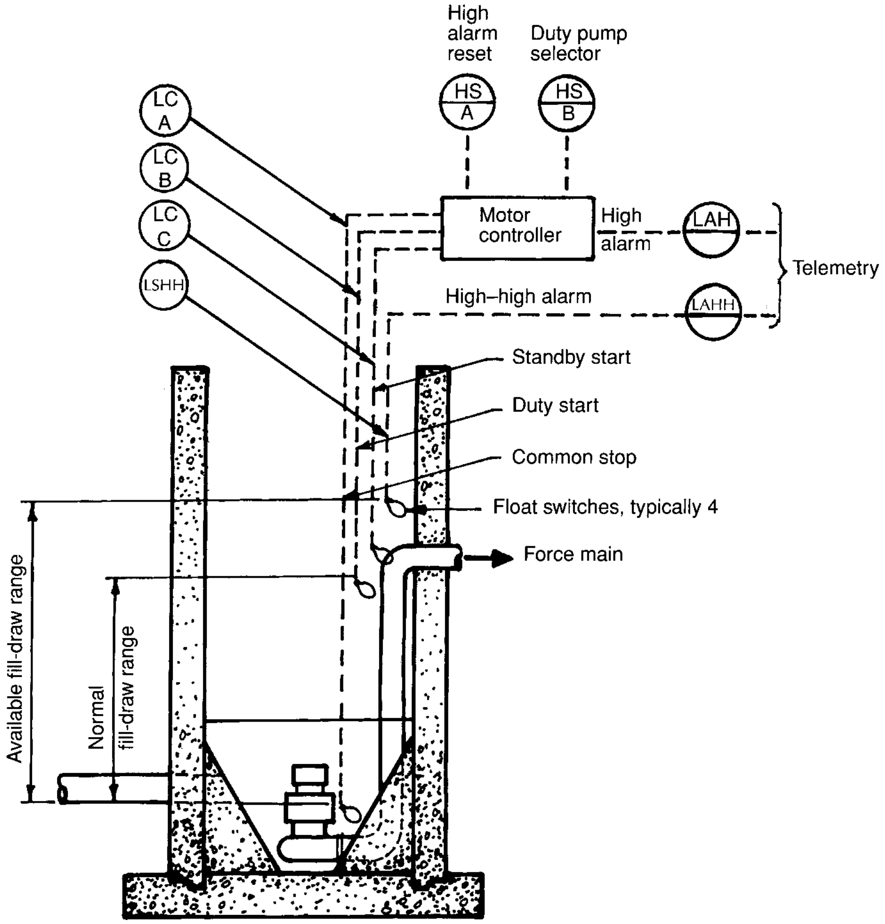


Figure 21-4. Controls for a small wastewater lift station with two C/S pumps.

selector switch is required (HS-B). Manual alternation is recommended to avoid the added complication of automatic alternators. If automatic alternation is used, however, a three-position selector switch (A-B-AUTO) should be provided.

One advantage of manual alternation, in addition to operator control of equipment wear, is the ability to provide an alarm-before-standby-start feature. With this arrangement, the high-water alarm device has a lockup feature that does not allow reset until the water level is lowered below the level at which the follow, or standby, pump shuts down. The alarm is set to initiate prior to starting the standby pump. The advantages of this feature are:

- It gives warning that something unusual has occurred and operation of the standby pump is required. The difficulty could be excessive inflow into the wet well, failure of the duty pump, or partial clogging of the duty pump.
- It gives operation and maintenance staff an opportunity to determine the cause before a genuine problem (flooding, backed-up sewer, etc.) causes damage.
- The alarm condition will remain as long as the duty pump cannot successfully control liquid level, and thereby it produces a signal that does not appear to the casual observer as a spurious alarm.

The above arrangement requires only four level devices, as shown in Figure 21-4. Furthermore, almost the entire fill/draw range can be utilized for each pump cycle. If the lead pump fails, the operator must visit the site to cancel and reset the high-level alarm—an important first step to ensure reliability. The second step is the immediate repair or remedy for the problem. Note that a careless operator could circumvent the reliability of the station and prevent alarm recurrence by simply exchanging duty and standby pumps. Utilities should be aware of, nurture, and continually

assess the sense of duty and responsibility in its workers and, for critical tasks, assign only those who demonstrate these virtues.

Float switches, although widely used in small lift stations, have a number of serious disadvantages: (1) when untethered, they tangle badly even in stilling wells; (2) if tethered to prevent tangling, the power cords are severely flexed by the turbulence in the wet well, cable life is shortened, and adjustment is inconvenient; and (3) they become quickly coated with grease that alters their buoyancy. The life of float switches is only about 2 years. A float is, however, unsurpassed as an independent switch for the high-water alarm. To facilitate testing a high-water alarm float, attach a light nylon cord to the free end so the float can be tilted by lifting the cord.

Two superior systems for small stations include (1) packaged bubbler systems and (2) pressure cells. A pressure cell can be suspended in a PVC pipe for protection from grease, sludge, and currents. The bottom of the pipe is set at about the mid-height of the submersible pump volute—above possible sludge banks and (to protect the cell from floating grease) below the lowest water level. Both the pipe and the pressure cell can be cleaned if necessary, although maintenance personnel usually dislike doing so (see Section 24-11).

21-6. Intermediate-Sized Lift Station

A lift station of intermediate size, as shown in Figure 21-5, may have three constant-speed pumping units that are typically each less than 60 hp in size. This type of pumping station differs from the small lift station described in Section 21-5 in that it has a separate dry well for the pumping units and may have a small surface structure to house the control equipment.

Pumping Unit Controls

Pressure switches LC-A, LC-B, and LC-C are connected to a bubbler system to control the pumps through very simple relay logic. Ordinarily, the pumps are sized so that two pumps can handle the peak flow and one pump is a standby unit. The amount of control logic required in this type of station can typically be handled by less than half a dozen relays. If remote monitoring of the station is required, however, a solid-state programmable controller or packaged microprocessor controller can provide both the control and telemetry functions at a lower cost than mechanical relays and a separate telemetry unit.

A major consideration for the control strategy of this type of station is limiting the number of pump

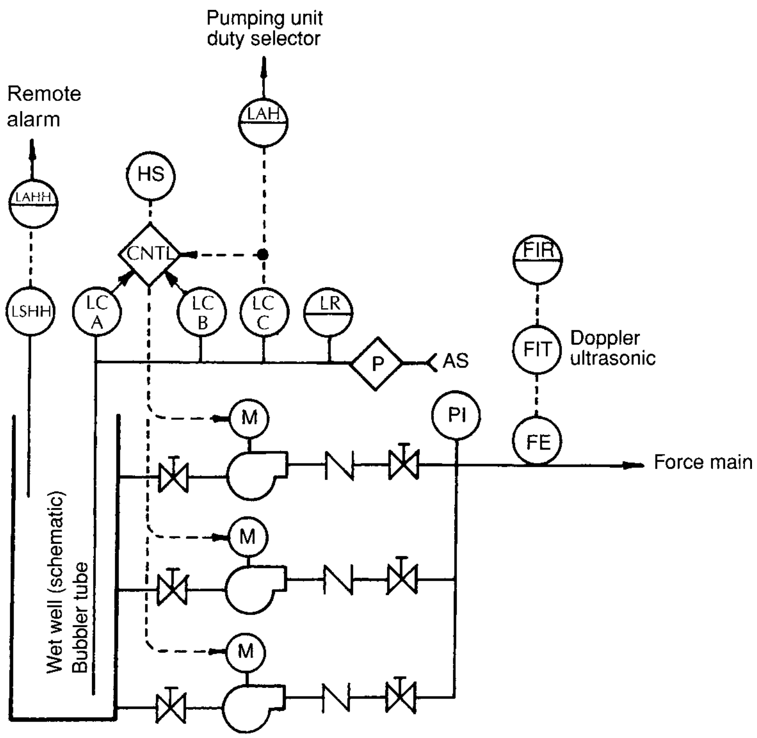


Figure 21-5. Intermediate-sized wastewater lift station.

start/stop cycles. The lead pump might be set to start at 80% of the available fill/draw range and stop at 5% of this range (LC-A). The follow pump would be set to start at 85% of the available fill/draw range and stop at 10% of this range (LC-B). If the standby pump is configured for automatic start, it would be set to start at 90% and stop at 15% (LC-C). Regardless of whether the third pump is configured to start automatically, LC-C should generate a high-level alarm that requires manual resetting at the station (LAH).

A manual duty selector switch is shown (HS). Motor duty-cycle considerations frequently require automatic alternation of the lead and first follow pumps driven at constant speed. For this situation, the duty selector switch requires three positions:

Pumps 1 and 2 are duty pumps: switch marked 1/2
Pumps 2 and 3 are duty pumps: switch marked 2/3
Pumps 1 and 3 are duty pumps: switch marked 1/3.

Auxiliary Systems

The auxiliary systems in this type of station include an air compressor for the bubbler and a dry well sump pump. Package equipment with built-in controls is suitable for both units.

Alarms

The following malfunctions should be monitored and alarmed to a central, continuously staffed facility:

- Wet well high level
- Third pumping unit started (if automatic)
- Dry well high level
- Instrument air pressure low
- Power failure.

21-7. Large Wastewater Pumping Station

The instrumentation and controls shown in Figure 21-6 are installed in an actual large pumping station operated by a sophisticated wastewater utility with a large number of pumping stations, many of which are larger than the one shown. This station was selected because it represents a complex instrumentation and control system that might well be used in other, simi-

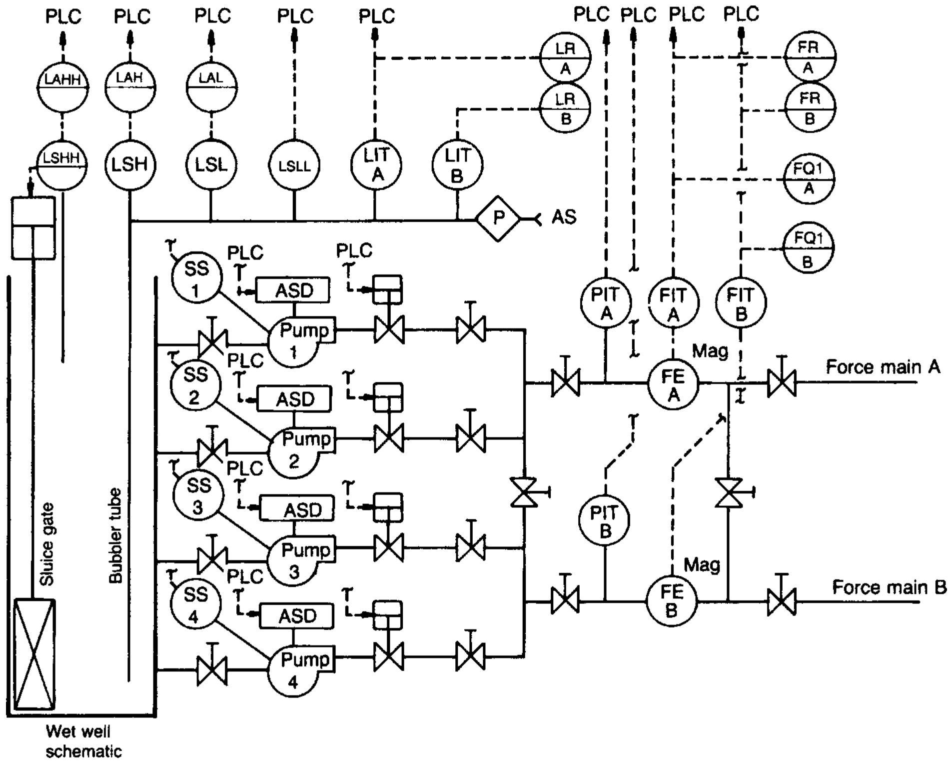


Figure 21-6. Simplified P&ID of the Sunset Pumping Station (see Figures 17-18 through 17-20). After Brown and Caldwell.

lar stations. The station has four 335-kW (450-hp), adjustable-speed pumping units and a maximum flow rate of $1.4\,m^3/s$ (32 Mgal/d) at a head of 52.4 m (172 ft). The instruments installed are listed in Table 21-1.

A solid-state programmable controller is used for the execution of all control logic, including pumping unit sequencing, speed control, and control of all station auxiliaries. The programmable controller also serves as the remote terminal unit for remote monitoring and control from the treatment plant. Accordingly, all signals generated by the instrumentation system are interfaced to the programmable controller whether needed for control or only for monitoring. About 130 relays and considerable analog instrumentation (which are normally required for a station of this type) are replaced by the programmable controller.

Pumping Unit Controls

The wet well is operated with a varying level to take advantage of the influent sewer depth/flow characteristic and to maintain scouring of the upstream sewer at low flows (see Figure 12-22). All pump sequencing is handled as a function of the wet well level, and all pumps running in a given level zone operate at the same speed and are linearly controlled by the water level within the zone.

The electronic level transmitter (LIT-A) produces a 4- to 20-mA signal proportional to wet well level. It is used to drive a recorder (LR-A) and the programmable logic controller (PLC) to provide pump speed control and sequence functions. A second electronic level transmitter (LIT-B) permits monitoring a much larger level range than is used for control. LIT-B is connected to a second recorder pen (LR-B). Pressure switches, independent from the electronic level transmitters, are connected to the bubble tube to provide high-level alarm (LSH/LAH), low-level alarm (LSL/LAL), and low–low level alarm (LSLL/LALL). A high–high level float switch acts as a backup to the bubbler system.

As illustrated in Figure 12-22, the level controller must be adjusted to drive the lead pump from minimum pumping speed to full speed as the wet well level increases from minimum to approximately 33% of influent pipe level so as to match the Mode I control curve shown. The follow pump is then required.

The PLC now activates the follow pump and must also change the speed-control adjustments to cause the pumping units to follow the Mode II curve. Both the control gain and set point must be changed to match the new curve. The results of changing the control adjustments are to slow the lead pump to a speed that discharges about 50% of its capacity and to accelerate the follow pump to the same speed as the lead pump. The result of these adjustments is that two pumps now pump the same flow discharged previously by one pump. The switchover is quite smooth. As flow continues to increase, full-speed operation will again be attained at approximately 66% of influent pipe level.

The switchover to three-pump operation is similar. The PLC must activate the second follow pump and readjust the speed control gain and set point to follow the Mode III curve.

Operation on decreasing level is reversed. The level switching points from Mode III to Mode II are delayed to approximately 60% of influent sewer level and from Mode II to Mode I, the delay is to approximately 30% of the influent sewer level so as to eliminate hunting at the switch points.

The need to readjust both the speed-control set point and gain may not be immediately obvious. The controller set point represents the vertical position,

Table 21-1. Instruments in a Large Wastewater Pumping Station

Instrument	Parameter	Instrument Type	Purpose
LSHH	Wet well level	Float switch	Alarm; close influent gate
LSL	Wet well level	Pressure switch on bubbler	Alarm
LSH	Wet well level	Pressure switch on bubbler	Alarm
LSLL	Wet well level	Pressure switch on bubbler	Stop pumps
LIT-A	Wet well level	Pressure transmitter on bubbler	Pump sequence and speed control; monitoring
LIT-B	Wet well level	Pressure transmitter on bubbler	Wide-range monitoring
SS	Pump shaft speed	Reluctance pickup	Pumping unit and discharge valve control
PIT	Force main pressure	Pressure transmitter with diaphragm seal	Monitoring
FIT	Force main flow	Magnetic	Monitoring

and the gain represents the slope of the pump control line segments in Figure 12-35. Control loop gain includes not only the control system gain but also the gain of the pump and associated drive system. If one pump is running, the gain of the pump and drive system may be considered to be G. If two pumps are running, the gain of the pump and drive system becomes 2G. Therefore, the gain of the speed controller must be halved to maintain the same total gain or slope for the pump control line. Similarly, with three pumps running, the gain of the pump and drive system becomes 3G, and the gain of the speed controller must be one-third of the gain with one pump running to maintain the same slope. Executing such a sophisticated control strategy is difficult in conventional control systems, but it is easily executed in PLCs.

Due to the high discharge head of this pumping station, pumps must be started and stopped against a closed discharge valve in much the same way as required for high service water pumping stations. After a pumping unit is started and a speed switch indicates that the pump shaft is rotating at a preset minimum speed, the discharge valve is opened. When a pumping unit is to be stopped, the discharge valve is first closed. When the valve actuates the closed-position limit switch, the pump motor is stopped. The pump shaft speed switches are also used to ensure that the motor is never energized while the shaft is rotating in reverse, as might happen if the discharge valve failed to close completely.

Auxiliary System Controls

As might be expected, a pumping station of this size has a large number of auxiliary systems that are essential to station operation. All auxiliary system control logic is executed by the programmable controller system.

Dry Well Drainage Sump Pumps

Two pumps are furnished and periodically alternated by the operators. Because there is always a possibility of raw wastewater getting into drainage sumps in wastewater pumping stations and an instrument air system is available, a bubbler is used in the drainage sump with pressure switches to control the pumps and generate a low-level alarm. An independent radio-frequency probe is used for the high-level alarm.

Instrument Air Compressors

Two compressors are furnished and periodically alternated by the operators. Receiver pressure switches provide control. A system alarm is generated whenever the alternate compressor is needed.

Air-Gap Tank

An air-gap tank ensures positive isolation between domestic water systems and station service water. Simple float switches are used to control make-up water and to generate high- and low-level alarms.

Seal Water Pumps

As with the instrument air system, two pumps are furnished and controlled by simple pressure switches. An alarm is generated whenever the second pump is required. A bladder-type accumulator (a sealed pressure tank with a rubber bladder to separate a precharge gas from the water and, thus, provide a variable-volume storage for the water) is used on this system to eliminate the need for hydropneumatic tank controls, as described in Section 21-2. The limited volumetric capacity of the expansion tank requires automatic alternation of the seal water pumps to limit the motor starting frequency. An interlock between the air-gap tank low-level alarm and the seal water pumps is needed to protect against loss of water supply.

Fluid Power Pumps

The station is furnished with two complete fluid power systems: one for the influent sluice gate and one for the pumping unit discharge valves. Each fluid power system has two pumps that are used for the normal operation of valves or gates and for charging accumulators for emergency operation of the valves or gates. Pressure switches are used to control the pumps. Reservoir level float switches and temperature switches are also furnished to protect the systems from malfunction.

Utility Power Supply Monitoring

The pumping station is furnished with two electric utility power services and an emergency generator. Phase-sensitive relays are installed in the electrical switchgear to detect undervoltage, phase unbalance, and phase reversal on each incoming power supply. If

the power supply is not within specifications, these relays (1) prevent actuation of the large, solid-state, adjustable-frequency drives; (2) signal the emergency generator to start; and (3) generate appropriate alarms.

Alarms

The following malfunction-monitoring devices and alarms are provided:

- Dry well sump level high
- Seal water pressure low
- Pump discharge valve fluid power pressure low
- Wet well level high
- Wet well level low
- Wet well level high–high
- Influent sluice gate closed
- Air-gap tank level high
- Air-gap tank level low
- Odor control system failure
- Ventilation system failure
- Instrument air system trouble
- Sluice gate fluid power pressure low
- Control power system trouble
- Programmable controller trouble
- Raw wastewater pumping unit trouble (four alarms)
- Utility power failure (two alarms)
- Emergency generator failure.

All of the alarms are locally annunciated and interfaced to the programmable controller system for relaying to the supervisory control system.

Chapter 22

Vibration and Noise

JERRY G. LILLY
WILLIAM D. MARSCHER

CONTRIBUTORS
Roger J. Cronin
Ray A. Gall
Richard O. Garbus
Willard O. Keightley
J. Davis Miller
Aloysius M. Mocek, Jr.
William E. (Ed) Nelson
Randall R. Parks
Jerry P. Pollack
James W. Schettler
Theodore B. Whiton

Vibration is one of the most vexing problems with pumping machinery, and it is the cause of considerable altercation and litigation. Noise can become a significant, annoying problem. Excessive vibration from primary equipment can be transmitted directly to the building structure, which causes uncomfortable (and sometimes dangerous) structural vibration levels. Excessive vibration of equipment and piping can destroy portions of the equipment (such as drive shafts, bearings, and seals), loosen or break pipe anchors, and even cause pipes to burst under certain conditions.

To make this chapter easier to use, the presentation is divided into four parts:

- Avoiding vibration problems. Sections 22-1 and 22-2.
- Troubleshooting excessive vibration. Section 22-3.
- Vibration analysis. Sections 22-4 through 22-12.
- Noise analysis. Sections 22-13 and 22-14.

The first two parts require no background knowledge of the subject and are easy to read. The third part is written for those who wish to delve more deeply into the subject of vibration. The fourth part is, of necessity, a simplified mathematical presentation of noise but with enough worked examples to make it easy to follow.

References to a standard or code are given in abbreviated form such as ANSI/HI/9.8-1998. Specific sections may be referenced as, for example, ANSI/HI 9.8, 2.2.3-1998.

22-1. Problems of Vibration and Noise

For the reader's convenience, the problems of vibration and noise in pumps and drivers are treated separately.

Pumps

Sources or causes of vibration include:

- The reaction of the impeller vane as it passes the casing cutwater.

- Pumps operating off the best efficiency point (BEP) and thereby creating eddies within the pump.
- Eddies created by water flowing through bends, valves, and other obstructions.
- In every rotating (e.g., motor) or reciprocating (e.g., engine) machine, some imbalance and shaft misalignment always exists.
- Resonance. If the natural frequency of the machine or system is nearly equal to the frequency of excitation, however small, the resulting vibration can become destructive.

Much of this chapter is devoted to the problems of lateral or translational vibrations, where the vibrating element actually moves back and forth. Rotating shafts may vibrate in both translational and/or torsional modes. In the torsional mode, the centerline of the shaft may not physically move in space, but the shaft does rotate at slightly different instantaneous rotational speeds with different phases at various points along the length of the shaft, which causes an oscillating torque. This torsional (twisting) vibration of the shaft can develop stresses high enough to cause the shaft to fail without any audible or visible warning signs prior to the failure. The subtle character of torsional vibration makes it all the worse.

The impact of a new pumping station on environmental noise levels in the community should be evaluated prior to final site selection, because noise adversely affects residential property values, causes sleep interference and general annoyance, and may lead to legal action against the owner of the plant. Building codes often include noise level limits at property lines. Noise problems are most likely to occur when facilities are located within 300 m (1000 ft) of a residential neighborhood. To protect the owner from legal challenges should a subsequent complaint develop into a court case, documentation of environmental noise levels by a qualified acoustical consultant prior to the start of construction in these instances is vital.

All too often, nothing whatever is done to reduce the noise level within the pumping station. Excessively high noise levels here can impair speech intelligibility and cause temporary or even permanent hearing loss to operating personnel exposed over extended periods of time. It is usually impractical to enclose a noisy machine in a sound-absorbing housing, partly because of the need to service the machine and partly because such measures are relatively ineffective. Common practice, therefore, is to enclose all the noisy machinery within thick concrete walls and ceilings. Requiring workers to wear ear protection and designing for reduced maintenance time and worker occupancy is helpful. The noise level in the reverberation field can be reduced by acoustical treatment (see "Indoor Sound Propagation" in Section 22-13). Although the noise level cannot be reduced by much more than 6 to 8 decibels, this reduction is often enough to permit conversation and substantially increase the level of comfort. At 2005 prices, the cost of acoustical treatment varies from \$40–\$130/m^2 (\$4–\$12/ft^2).

Drivers

Sometimes electric motors are the cause of excessive noise or vibration. Vibration tends to have unusually high spikes at one and/or two times the line frequency (60 and/or 120 Hz). If there is a broken rotor bar or bad stator winding, a high-pitched noise (and accompanying vibration acceleration) is often evident at the "slot pass" frequency (about 100 times running speed). In motors driven by AF controllers (particularly the older models), strong vibration pulses are evident even for healthy motors at 6×, 12×, 18×, and sometimes 24× the frequency with which the controller feeds the motor at any given speed. These frequencies seldom cause dynamic or structural resonances of any significance. However, these harmonics can sometimes excite linked-rotor system torsional critical speeds that are difficult to track down with standard vibration equipment that cannot directly sense torsional oscillations, even if shaft stresses are becoming excessive. If shaft failures are occurring at the shaft shoulder near the pump impeller or coupling hub, this situation should be investigated.

Engines

If the driver is an internal combustion engine, the primary driving frequencies (besides one times and two times running speed and the pump vane passing frequency) are the cylinder stroke frequency (i.e., the number of cylinders times the engine speed) and its first 10 harmonics. For most engines, the second harmonic (i.e., two times engine speed times number of cylinders) is the strongest, and no natural frequency of the engine-support floor structure or the engine-gear-pump train should be allowed to exist anywhere in the typical operating speed range. Resonances of the other first 10 harmonics cannot, however, be ruled out, and they should be investigated if there is excessive noise or vibration in such installations. Offset or right-angle gear boxes can also be a torsional excitation source at the gear mesh

frequency (number of gear teeth times running speed). Gear mesh excitations can also cause gear tooth damage and/or excessive noise due to axial resonances of the drive shafts, particularly if the drive shafts are hollow. Filling a hollow drive shaft with grease may eliminate the problem by energy-absorbing damping.

For every engine-driven pump—whatever its size—designers should specify that the pump manufacturer shall make (or have a qualified analyst make) at least a torsional vibration analysis of the pump-driver system. A translational vibration analysis should also be made for the engine-pump-support structure. Many low-cost pump bidders are not qualified for such work, so prequalify bidders to protect both the client and your reputation as a designer. Alternatively, prequalify third-party specialists to perform the analyses and require the pump manufacturer or the contractor to retain only a firm from the list. This approach avoids the potential of a conflict of interest on the part of the individuals performing the analyses. See Appendix C Paragraph 11050–1.05B.3 for details.

Active Noise Control

The use of electronically generated sound waves to cancel or reduce unwanted sound (noise) is called "active noise control." This technique has been known for decades, but it has not been widely implemented because of practical limitations in its application. Recent advances in signal processing hardware and software have improved the situation considerably, but active noise control is still generally restricted to low-frequency applications (typically below 200 Hz) and to noise contained inside a duct or a pipe. Once the noise has escaped into the listening environment, this technique is no longer a viable means of achieving a significant noise reduction over an extended region of space.

22-2. Avoiding Vibration Problems

By following a few simple rules, a nonspecialist can—fortunately—avoid about 90% of all vibration problems. The presentation herein is focused on minimizing vibration in pumping station machinery and its related systems. Necessary concepts are explained in simple terms, and their practical use is described. The information is relevant to small- or medium-sized centrifugal pumps of all types, reciprocating pumps, ventilation fans, and electric motor and diesel drivers. For stations with pumps larger than 75 kW (100 hp), or with engine or variable-frequency drives, review the more detailed procedures presented in later sections of this chapter or have a formal vibration analysis made.

Pump Selection

Select a pump that will operate close to its best efficient point (BEP). Plot the maximum and minimum headloss envelope, and choose a pump suitable for the entire range. Then, as conditions change (flow rate increases, pipe becomes rougher), it is necessary only to change the impeller to meet the new conditions. Beware of applying large factors of safety to the flow rate, because this blunder could result in an oversized and therefore unreliable pump.

Balancing

Adequately balance all rotating components of diameter equal to at least one-third of the pump impeller diameter. Balance all components (in two planes) that have a length of at least two-thirds their diameter. The entire rotor system should be check-balanced in the assembled condition. A balance standard shown to be conservative for pumping station equipment is $e = kW/N$, where k is a constant equal to 0.423×10^{-3} for SI units, e is the imbalance in $k \cdot m$, W is the mass of the balanced component in kg, and N is the peak operational speed in Hz. In U.S. customary units, the equation is $e = 16\ W/N$, where e is in oz · in., W is weight in lb, and N is speed in rev/min. Reports to demonstrate compliance with the balancing standards should be submitted as product data. Do not accept any loose rotating component fits where the mass of the component multiplied by the radial clearance approaches the above criterion. Check the shafts to ensure that, if shaft bow is detectable, the amount of bow times the supported component weight also does not exceed the above criterion.

Accurate alignment is necessary to make the centerline of the driving and driven equipment coincide as closely as possible both in concentricity and parallelism. To specify adequate coupled shaft alignment for a rigid coupling or one with one axial location of flexibility ("single engagement"), a good rule is to maintain an error of no more than 37 to 50 μg (1.5 to 2 mils) in the concentricity between coupling hub centerlines. This concentricity criterion may be loosened by an additional 1 to 2 mm per m (1 to 2 mils

per in.) of length between coupling hubs for double engagement or "spacer" flexible couplings, depending on the coupling type. The maximum permissible angle between shafts at the point of coupling engagement is 5 to 10 min, which is automatically maintained if the above offset specifications are maintained. Coupling manufacturers often quote larger amounts of offset and parallel misalignment than the above values, but these quotes account only for coupling survivability, not equipment bearing survivability, and they also do not account for the increased equipment vibration that misalignment might cause.

Pumps with stuffing boxes (packing glands) are less sensitive to misalignment than those with mechanical seals, and the maximum criteria above are satisfactory where stuffing boxes are used. Use the minimum criteria for pumps with mechanical seals.

Alignment should be performed only by qualified technicians or millwrights. Alignment is best determined by the reverse dial indicator method or its modern optical equivalent (although the latter is best performed only by an expert) as described in detail on pages 2.413ff in the *Pump Handbook* [1]. Keep in mind that the above alignment criteria can be applied with the equipment cold, but steps must be taken to ensure that they also apply to the "hot" alignment (i.e., with the pump or fan running, after warm-up). Also, be sure to account for significant machining inaccuracies in whichever coupling faces or rims are used for measurement (reverse dial indicator largely accounts for that). Furthermore, compensate for gauge support arm sag, because the sag can be larger than the alignment specification level. Once alignment is achieved, the pump or fan feet should be doweled to the baseplate to avoid loss of alignment from slip at bolted joints. Grout baseplates to the foundation pad with nonshrinking epoxy grout (per manufacturer's recommendations), and take care to leave no air pockets of significance between the grout and baseplate by the liberal use of vent holes—some located by tapping the baseplate top and listening for hollow sounds to find voids. See Section 12.10, Items 27–31 and Section 25–8.

Pump Support

The suction and discharge piping end flanges and the opposing pump nozzles must be properly supported. Avoid the use of unrestrained pressure-bearing "expansion" or "flexible" joints at pump nozzles for pipes equal to or larger than 150 mm (6 in.) in diameter unless the contained pressure times the nozzle cross-sectional area is within the manufacturer's nozzle load limits, and the pump and piping natural frequencies are well removed from the range of vane pass frequencies. Although such joints relieve any piping thermal expansion or Bourdon-tube effects from "loading" the equipment nozzles, they do not allow the piping to absorb the cross-sectional nozzle hydraulic load (i.e., the pressure times the open area). This nozzle hydraulic load can produce a large thrust perpendicular to the nozzle opening and severely load the pump casing. Failure to account for such loads has caused serious operating alignment problems, casing rubs, and system damage in many installations.

Natural Frequencies of Pump and Piping Supports

Design piping and pump supporting structures (including floors) to have natural frequencies that are at least 25% less or 25% more than the operating range of the key excitation frequencies in the pump. Typically, these excitation frequencies equal (1) the running speed, and (2) the number of vanes times running speed. Loaded-floor natural frequencies can be significantly different from those of the bare floor. In addition, the floor stiffness can influence the natural frequency of the pump because the manufacturer calculates the natural frequency on the basis of an infinitely stiff floor. For horizontal and vertical nonclog pumps, include the pump mass (plus the water contained) and pedestal stiffness when calculating loaded-floor natural frequencies. For vertical turbine pumps, include the motor and pump bowl assembly masses, and the discharge head and column piping stiffness when calculating loaded-floor natural frequencies. The component masses and stiffnesses are available from the manufacturers upon request.

These calculations can be made by the finite element method, but it is possible that the same models used for the structural design can be used to calculate the building floor and pipe structural natural frequencies after bulk elements are included for the pump and motor, as described above. Natural frequencies for the pump itself with added mass and stiffness of the floor and piping added are best found by the finite element method.

Pipe Support Spacing

Use a liberal number of unequally spaced pipe supports. This rule includes discharge piping for submersible pumps. To avoid vibration problems, piping must

be well supported in three perpendicular directions (axes x, y, and z)—not just in the vertical direction. To avoid or suppress vibration, both the weight of the piping and adequate stiffness must be considered in designing proper supports (see Section 22-10).

Drive Shaft Offsets

Do not offset drive shafts at an angle. In the past, drive shaft manufacturers have often recommended at least a 3-degree offset to help lubricate needle bearings, but it has been found that an offset is not needed for bearing lubrication. Furthermore, an offset excites vibration at a frequency of two and four times the running speed.

Baseplates and Foundation Pads

Foundation pads should extend beyond the machinery baseplate by at least 75 mm (3 in.) or one-half of the floor-plus-pad thickness, whichever is larger. The effective thickness from subfloor beams and gussets need not be taken into account in this calculation. The mass of machinery foundations should be 5 to 10 times that of the machine itself. The foundation should have an adequate aspect ratio. That is, the "footing" lines from the centerline of the pump inclined 30 degrees to the vertical should pass through the bottom of the foundation mass—not through its sides. The centerline of the pump is considered to be at the center of the impeller hub except for vertical turbines.

Pump Suction Piping

If possible, use straight suction piping up to the pump suction nozzle, and keep any required bends in a single plane. As a minimum, avoid piping reducers or bends within 5 to 10 suction pipe diameters of the pump inlet, unless the pump has been specially designed to accommodate such fittings closer to the pump (as in some vertical nonclog pumps). If in doubt, seek the pump manufacturer's advice.

Single and Double Volutes

The vane passing force is generally two to three times higher for single volute pumps than for "twin" or "double" volute pumps. Where practical, choose double volute pumps (preferably with an odd number of impeller vanes), because they minimize the vane passing force and maximize both BEP and, particularly, off-design efficiency. Single volutes are often necessary because of cost or solids-passing capability. The hydraulic imbalance present in single volutes is minimized, at some decrease of efficiency, by increasing the clearance between impeller and volute tongue.

Pump Suction Conditions

Ensure adequate pump suction conditions with sufficient NPSHA including all significant effects (see Figure 10-13) and a sump design in which surface and subsurface vortices are suppressed.

Recommended Detailed Design Practices

The potential for vibration problems is often given insufficient attention at the beginning of the pumping station design process. By following appropriate design rules, and with a bit of luck, it is likely that the vibrational behavior of the installed equipment will be acceptable. However, this is a risk not worth taking for pump drivers of more than 56 kW (75 hp), and the cost to all parties in those situations where vibration problems do occur quickly overshadows the "savings" of not performing analyses and test procedures such as those recommended below.

Controlling Excitation Forces

The first step in controlling excitation forces is to limit mechanical operational forces that can act to drive machinery vibration. The primary forces that must be controlled are caused either by imbalance of large diameter rotating components such as coupling hubs or pump impellers, or by misalignment of coupled shafts. These issues are discussed above.

Although mechanical forces may be the most common cause of excessive vibration, another common problem is the use of a pipe roughness coefficient that is too conservative and a resulting specified total discharge head (TDH) that is higher than the one encountered. A pump selected on such a basis operates well to the right of its BEP, discharges at a higher flow rate than it should, and increases the hydraulic excitation forces. In many instances, the pump will be operating in a cavitation region. The motor draws more current and may be overtaxed. Eliminate this problem by ensuring the proper range of C values is used for selecting both pump and impeller.

The primary frequency of hydraulic excitations is the vane-passing frequency. The forces and the resulting vibration depend upon several factors, controllable at the design and specification stage. The most important method of minimizing vane-passing vibration is to design proper suction piping, to use double volute or centered volute pump designs wherever possible, and—most important—to operate the pump at or near the BEP.

For all types of volutes, it is possible to lower vane passing forces at some expense of efficiency by opening up the clearance between the impeller vanes and the volute tongue or diffuser vanes. This opening is called the "B-Gap." Designers should, in general, be wary of excessive vane-passing vibrations from any pump with a diametral B-Gap of less than 4% of the impeller diameter, and a B-Gap of 6 to 10% or more is preferable, although excessive B-Gap can encourage discharge recirculation, as discussed below, when operating below the BEP. A sound approach is to follow manufacturer guidance on the setting of the B-Gap, while enforcing appropriate vibration specifications (discussed below) and performance guarantees throughout the operating range.

Reduced Flow

For all types of centrifugal pumps, the vibrations, shaft deflections, and bearing loads usually increase as the pump is run at lower flows. They do not decrease as is commonly assumed. In fact, if centrifugal pumps are operated well below BEP (say, at 50% of the design flow at any given speed), suction and/or discharge recirculation will probably occur, accompanied by possibly large excitation forces at vane pass and at frequencies below running speed. The problem in both suction and discharge recirculation is that stalling is induced on the vanes or tongue similar to the stalling that occurs around an airplane wing at a bad angle of attack. This condition causes strong eddies, which resemble and act like miniature tornadoes, to be spawned and "kicked upstream" of the stalled passage. These "stall cells" often subsequently rotate near the impeller at a speed somewhat less than running speed (usually about 10 to 40% in the diffuser or volute or about 60 to 90% at the inlet of the impeller). The high local velocity and low local pressure associated with these cells (keep in mind the analogy of a tornado) interact with the vanes of the impeller and cause new and usually unexpected excitation force frequencies. In summary, *running a centrifugal machine at flows (loads) below its rated capacity (i.e., well below its BEP) typically causes more, not less, stress and vibration, and decreases machine life and reliability*. (Of course, vibration due to unbalance or misalignment decreases at lower speeds.) These consequences should be considered in the lifetime cost and factored into design and purchasing decisions. Pumps can be selected for both present and future (larger) capacities by choosing a model that can meet both requirements merely by changing the impeller. The motor must be capable of operation at both flow rates as well. In many high-specific-speed pumps, power requirements at reduced (less than BEP) flow rates can can be higher than at BEP.

Selection of Machines

For reliable operation and long life, select centrifugal machines that will operate as close to their BEP as possible throughout their operating range, and keep the flow rate within the manufacturer's maximum and (especially) minimum limits. Designers should pay attention to system changes with time because (1) flow resistance can increase with time due to increasing pipe roughness (particularly in long force mains), (2) the pump capacity can decrease significantly as clearance around the wear rings increases, and (3) erosion further decreases discharge capacity. The effect of system changes on suction line losses, and therefore on NPSHA, must also be analyzed and compared to the manufacturer's requirements. In reciprocating pumps, the effect of "acceleration head" due to the unsteady pulsing flow in long suction lines must be considered (see ANSI/HI 6.1–6.5-2000).

Acoustic "organ pipe" resonances must be avoided (as discussed later in this chapter), or they may become exciting forces. Concern with acoustic resonance is particularly apt in reciprocating machines. Their strong flow and pressure pulsations at piston or plunger frequency are able to excite certain lengths of manifold or piping, and they are often run across a broad speed range. Accumulators are often needed near the inlet or discharge manifolds to detune and/or dampen potential acoustic resonances.

Net Positive Suction Head

To minimize hydraulic forces for all types of pumps, it is important to operate the pump with sufficient NPSHA (see Section 10-4). In addition to causing cavitation, inadequate NPSHA can excite unexpectedly high vibrations at various natural frequencies. From Bernoulli's equation, any increase in local velocity decreases static pressure and thus increases the

cavitation potential, so it is important to keep the pump inlet flow velocities low (preferably below about 2 m/s or 6 ft/s) and evenly distributed. Hence, a large suction pipe (at least equal to the pump suction flange ID) should be used. Pipe reducers should preferably be at least five pipe diameters upstream from the pump flange. Maintaining this distance from more aggressive suction disturbances such as valves and elbows is even more important. Besides avoiding cavitation, this practice also ensures well-distributed flow velocity and pressure at the inlet and throughout the pump, and thus minimizes the hydraulic excitations in general. High velocity in the discharge is not usually important because the pump increases the static pressure well above the cavitation point.

Wet Wells

Minimize the possibility of vortex formation and discourage whirling flows around column piping by using care in designing the wet well. Follow the advice in Chapter 12 (preferably) or in ANSI/HI 9.8-1998.

Avoidance of Resonance

Resonance may cause excessive vibration even when mechanical and hydraulic forces are properly controlled. To avoid this problem, distribute the installed system (pump, piping, pedestal, and floor) mass and stiffness in such a manner that no natural frequencies are close (within at least 25%) to excitation frequencies, such as one or two times the running speed or one times the vane-passing frequency. This analysis is done in the design stage by successive trials, using appropriate computer models, such as finite element analysis. In such a model, pump and motor component drawings of sufficient detail must be obtained from the manufacturers—at least overall external dimensions, weight, and center of gravity (relative to the mounting flange)—and pedestal or base (including gussets), floor (including subfloor beam and column dimensions and locations), piping (if rigidly connected, including joint directional rigidity), and water mass must be included. Simple manual calculations are not likely to be sufficiently accurate to avoid resonances because of the complexity of these interlinked components. So much effort may be overkill for small stations, but it should be included in the design of large stations.

A common misconception is that if only the pump is designed stiffly enough, then natural frequency resonances will be avoided. Because the lower natural frequencies of assembled pumping and compression systems usually depend strongly on the base and floor design as well, as discussed by Reichert et al. [2], any reliable analysis must include the pump, base, and floor together. Similarly, any test for the lower natural frequencies of installed machines and possible resonances is unlikely to be valid in the pump manufacturer's shop (although hydraulic/aerodynamic performance tests and rotordynamic testing should be valid). Only when all equipment is installed on its final foundation, filled with water, and with all bolts tight, will representative natural frequencies be present in the lower frequency ranges and detectable by a shaker or impact test.

Avoiding all matching between significant forcing frequencies and all natural frequencies may not be feasible with some pumps, particularly those operating at variable speed with a turn-down ratio of 30% or more. For example, if the first natural frequency occurs at 70% of the running speed in a vertical pump with a variable frequency 2:1 turn-down range, it is unlikely that mechanical design changes can be made that would move such a natural frequency up enough or down enough to be out of the speed range added to the resonance avoidance margin of 25%. Although vibration isolation pads can be used to decrease the reed frequency as much as necessary, the resulting flexibility of the revised system is likely to cause alignment problems, and also is likely to decrease the second natural frequency into the running speed range, thereby trading the original resonance for a new one. If resonance occurs in such instances, there are several alternative solutions:

- Tighten the balance specifications, and tighten the maintenance schedule and procedures, for example, by field trim balancing in which mass is added to the coupling hubs without disassembly.
- Reduce the natural frequency as much as possible by, for example, installing vibration isolation pads under foundation bolts until alignment approaches a potential problem due to static or low-frequency hydraulic forces. Unfortunately, reducing the natural frequency is accomplished by successive trials until the second natural frequency is within 25% of the running speed range.
- Use "lockout speed range" settings on the speed controller to avoid operation within $\pm 25\%$ of the problem natural frequency. The use of lockout settings may cause a larger "hole" in the operating speed range than is tolerable from an operational standpoint. If the amplification factor at the edges of the compromised range results in acceptable vibrations at maximum imbalance and misalignment and at minimum flow, the lockout range can be

relaxed to ±15% and even perhaps to ±10%. The natural frequency sometimes shifts with ambient temperature or with water level in the sump.

Rotordynamic Analysis

Rotordynamic analysis of the pump, compressor, or fan rotating system should be performed by the manufacturer or a third-party consultant. The authors prefer that the work be the product of third-party specialists who have a demonstrated record in the field and are registered professional engineers. This analysis requires specialists, and details are not presented here. Some of the items that should be requested, however, are discussed below.

Usually the rotordynamic analysis performed by the manufacturer is for lateral vibrations only, not for torsional or axial vibrations. All three analyses should be performed, particularly for variable-speed systems. It should be made clear in the request for quotes (or in the equipment specifications) that a complete analysis is required so that all manufacturers are bidding on an equal basis and will include the cost of the analysis in their bid. A paper by Corbo and Malanoski [3] is an excellent reference for writing a purchase order or project specification. It describes step-by-step procedures and the results required to demonstrate compliance with requirements as outlined in Appendix C. The cost of the rotordynamic analyses and the torsional analyses, sometimes called the dynamic analyses of the system, should also include the pump and motor frames, the supports for the motor or engine, and other appurtenances such as intermediate bearings and gear reducers. A complete analysis, including finite element stress analyses for fatigue strength, usually costs in the neighborhood of $20,000 (in 2005 dollars) per set of like pumps. One consulting firm has found that 80% of the pumps submitted failed to meet these requirements and needed some kind of remediation. In all instances, the changes resulted in improved performance and exceptionally smooth-operating equipment. See Section 1.05.B.3 in Appendix C for more details.

Regardless of whether a structural vibration and rotordynamic analysis is performed on a pumping station machine and drive train, acceptance should be based upon the concurrence with the specifications of the results of a vibration test performed on-site in the final, fully assembled and running condition. If the vibration is found to be excessive, the manufacturer should be required either to fix the problem or prove that the problem is not related to the equipment. If the latter is true, a knowledgeable consultant should be retained to ensure that the manufacturer is correct and to help find a fix to the problem.

Lateral Rotordynamic Analysis

The minimum rotordynamic analysis should consist of the determination of rotor critical speeds (i.e., non-critically damped rotor natural frequencies) and mode shapes from a frequency of zero up to the maximum excitation frequency. If the bearing stiffness has a large tolerance or might shift significantly with time, such as from the Lomakin effect (see subsection "Description of Basic Vibration Terms, Concepts, and Equipment" later in this chapter) in centrifugal pumps (as the clearances increase due to wear in normal service), a critical speed map should be constructed in which the value of these frequencies versus the range of possible bearing stiffness values is plotted. The predicted critical speeds need to be compared with the excitation frequencies to determine whether a resonance is possible. If so, the offending natural frequency must be moved or sufficiently damped by, for example, using a different style of bearing or coupling.

If any of the rotor natural frequencies might (with reasonable allowance for system degradation) match the frequency of an excitation force within the intended speed range, a "forced response" analysis should be performed. Sometimes this analysis is done only for one time running speed excitation. It is then called an "unbalance response" analysis. In such an analysis, it is assumed that the rotor is driven by a worst-case excitation and the amount of vibration that can occur at the various critical locations along the rotor (e.g., at bearings, wear rings, and mechanical seals) is calculated. These results can be compared with available clearances to determine whether rubbing is likely and whether the bearings might become overloaded.

Vertical turbine pumps exhibit nonlinear shaft dynamics because of the large shaft excursions that occur in the lightly loaded long length/diameter ratio bearings. Given the flexibility of the lineshaft and the weak support provided by the pump casing column piping, and given the relatively large assembly tolerances and misalignments in the multiple lineshaft bearings of these machines, the contribution of each bearing to the net rotordynamic stiffness is nearly random and is constantly changing, as explained by Marscher [4]. As a result of the changing stiffness, there is no single value for each of the various theoretically predicted natural frequencies. Instead, the natural frequencies of the lineshafting and shaft in the bowl assembly must be considered on a time-averaged and location-averaged

basis and they constantly vary between two limiting states. A useful simplified method of predicting lineshaft reliability with a worst-case model is known as the "jump rope" model, and is documented by Marscher [4]. Another important vibration problem in certain vertical pumps is the vibration of the shaft enclosure tubes that provide a jacket of contaminant-free lubricating fluid to the lineshaft bearings. The natural frequencies of these tubes (supported by "spider" assemblies) can be analyzed for various end supports using finite element analysis or by using the multiply-supported beam models of Blevins [5]. Vertical pump vibration analysis of the stationary structure, the lineshafting, the shaft enclosure tubes, and the pump and motor rotors should be done simultaneously by means of finite element analysis.

The possibility of rotordynamic instability due to lateral motion needs to be analyzed in centrifugal and axial flow machinery, particularly in large fans and certain motors wherein cross-coupling forces (those induced by rotor motion perpendicular to the motion, as discussed in the subsection "Basic Vibration Terms, Concepts, and Equipment" later in this chapter) are often large enough to overwhelm the damping. Rotordynamic instability refers to phenomena whereby the rotor and its system of reactive support forces are able to get "out-of- synch" with each other and become self-excited. Potentially catastrophic vibration levels ensue even if the original excitation forces are quite low. The characteristic of rotordynamic instability, sometimes called "shaft whip," is that it is whirling at about one-half running speed and begins when running speed exceeds twice the first bending natural frequency of the shaft. Very few pumps have natural frequencies in the running speed range with damping low enough that this process can occur, but motors and fans sometimes do.

If an unstable machine is encountered, a typical design modification to reduce the tendency for rotordynamic instability involves bearing changes. The type of bearing most likely to participate in instability problems is the plain journal bearing, because it has very high cross-coupling (although it also has high beneficial damping). Bearings that discourage whirling lubricant flow tend to decrease cross-coupling—dramatically so in terms of the axially grooved and tilt-pad-style bearings described in the *Pump Handbook* [1].

Torsional Vibration Analysis

Unlike lateral rotordynamic analysis, which usually can be performed independently by various manufacturers, torsional analysis is valid only when performed for the entire linked-up drive train. Torsional analysis of only the driver or the pump or compressor alone is without value. Also, the flexibility, backlash, and gear ratio of couplings and gear assemblies, if any, must be estimated and included in any accurate model. Methods of manually calculating the first several torsional natural frequencies for simple rotor systems are given in Blevins [5], but a finite element analysis is generally needed for complicated pumping station systems. A detailed description of how such analyses should be performed (and an excellent basis for a specification on the conduct of such analyses) is given by Corbo and Malanoski [6].

Forced response torsional analysis is necessary if any of the torsional natural frequencies coincide with a strong excitation frequency. To determine the frequencies at which large values of vibratory excitation torque are expected and the value of the torque occurring at each of these frequencies, the pump torque at any given speed and capacity can be multiplied by a "per unit factor." The per unit factor at important frequencies can be obtained from pump, compressor, motor, and control manufacturers for a specific system, and it is typically in the range of 0.01 to 0.1 zero-to-peak for important excitations. The most important torsional excitation frequency from an electric motor is the motor rotating speed times the number of motor poles. Unsteady hydraulic torque from the pump is also present at a frequency equal to the running speed times the number of impeller vanes, and the maximum intensity equals the delivery torque divided by the number of impeller vanes, although typically the per unit factor is 0.03 to 0.1. The gear mesh frequency (number of teeth times the rotational speed for a given gear) is usually a strong torsional excitation frequency with a typical per unit factor of about 0.02.

The worst-case torsional vibrations in pumping station rotors often occur due to temporary excitations during start-up, trip, and motor control transients. Therefore, it is wise to make a time-integration analysis to determine the transient peak stresses caused by these transients.

Particular care should be taken with systems involving adjustable-frequency drives (AFDs). Besides sweeping the excitation frequencies through a large excursion and therefore increasing the chances of a resonant encounter (Marscher [7]), AFD controllers provide new control pulse excitations at multiples of the motor running speed, commonly at 6×, 12×, and 18×, and often at whole-fraction submultiples as well. The controls manufacturer can predict these frequencies and their associated per unit factors.

Judgment of the acceptability of the assembly's torsional vibration characteristics should be based on whether the forced response shaft stresses are below the fatigue limit by a sufficient factor of safety at all operating conditions. The recommended factor of safety is 3.0 if all stress concentrations (such as keyways) are taken into account. If specific stress concentration factors are not known, assume that they have a value of 3.0 and use a revised factor of safety of 9.0 at any shaft location that has a step, attached part, thread, or keyway.

Vibration Specifications

Specifications concerning acceptable vibration levels, the method and frequency range of measurement, and the method of interpretation of the results should be clear and reasonable. Vibration-minimization responsibilities, such as who will do the pre-design analysis and who will do the post-installation testing, must be delineated and the responsible parties must be identified. There should be a list of items required, such as results from certain types of validated computer programs or testing procedures. Responsibility for each item should be assigned in the bid request so that bids are on an equal basis. The inclination or ability of low-bid suppliers to provide an appropriate level of technology should be specifically required—not assumed—in the bid and contract.

Injury to machines due to excessive rotor vibration consists of wear or fatigue damage to the pump internal components, such as bearings, annular seals, mechanical seals, and shafts. Most of this damage depends on the total displacement associated with the vibration. However, as machines are made faster, they become smaller, and hence the amount of vibration displacement they can tolerate decreases proportionally to the machine speed. Therefore, the allowable running speed vibration velocity (displacement times running speed) is roughly constant regardless of the running speed of the machine. Historically, machine survival versus failure data support the use of constant vibration velocity versus speed as an acceptance criterion in assessing vibration severity (see Rathbone [8], Blake [9], Baxter [10], and Hancock [11]). However, the raw information in these references was based on measurement equipment that could not distinguish between various frequency components. Therefore, vibration severity could be plotted only as unfiltered (total vibration including all frequencies) displacement readings versus machine running speed and not filtered (individual values at specific frequencies) velocity values versus frequency. Unfortunately, in many specifications (with the notable exception of ANSI/HI 3.1–3.5-2000 and ANSI/HI 6.1–6.5-2000), it is assumed that the original data can be interpreted as velocity versus frequency, and the specifications are written on that basis. Be very cautious in using such specifications because instability and hydraulic problems that cause rubbing at low frequencies tend to be overlooked, and the specifications may require unnecessarily small vibration displacements at high frequencies. Such specifications cause the rejection of good equipment at the expense of all involved, as discussed by Marscher [12].

Typical specifications for pumping stations to establish vibration test types, measurement locations, and acceptance criteria are contained in ANSI/HI 9.6.4-2000, API 610, Mil-STD-167-1 (Ships), and various ISO machinery vibration specifications, such as ISO 2372. These specifications require the monitoring of bearing housing vibration by using an accelerometer or velocity probe in three perpendicular directions at the location of each of the bearings in the drive train. The acceleration readings are typically integrated to obtain vibration displacement and/or velocity values—the typical terms for the criteria. Occasionally, the specifications contain procedures for the installation and evaluation of shaft proximity probes mounted in some bearing housings to measure shaft versus housing displacement.

Acceptability criteria vary among the various specifications. A conservative approach, consistent with all of the quoted specifications, is to require that vibration does not exceed any of these three criteria at any frequency: 0.05 mm (2.0 mils) displacement peak-to-peak, 6 mm/s (0.25 in./s) zero-to-peak (or just peak) velocity, and 1.0 *g* peak acceleration.

In essence, vibration velocity can be thought of as displacement times the frequency at which it occurs times a scaling constant. Likewise, acceleration can be thought of as displacement times frequency squared, times a scaling constant. The value of scaling constants is not a key to the discussion here, although knowledge that the above relationships exist brings out the important point that vibration displacement, velocity, and acceleration are all measurements of the same quantity, but with different emphasis given to vibration that occurs in different frequency ranges. At low frequencies, for example, displacement is the most stringent criterion, while at high frequencies acceleration is the most stringent. Depending on the type of equipment, displacement may be more likely to reject unreliable machines and pass reliable machines than acceleration would be. On the other hand, for machines prone to problems

that show up at high frequencies (usually not true of pumping station equipment), the opposite would be true. Vibration velocity is used in many specifications as a "middle-of-the-road" single acceptance criterion, but this practice is not recommended because it tends to be blind to potential rubs at low frequencies and is too stringent as a general criterion for most machinery at high frequencies. In practice, the 1.0 *g* peak acceleration specification given above is less stringent than the displacement and velocity specifications except for high frequencies. Generally speaking, high acceleration measurement is useful only as a flag that there is something unusual (possibly harmless) in the system, and by itself does not necessarily indicate that there is a problem with the machine.

For vertical pumps, the maximum allowable displacement may be significantly higher than 0.05 mm (2 mils) due to the geometrical lever principle acting to magnify the motion associated with a tall motor or pump bearing tower (ANSI/HI 9.6.4-2000).

Piping vibration is in a special class. Pipes can be allowed to vibrate up to roughly three times that allowed for the machine to which it is attached (see Wachel and Bates [13]).

22-3. Troubleshooting Excessive Vibration

Vibration problems may sometimes occur in existing stations, in new stations because of corners cut during the design process, or in new stations in spite of following recommended practice. These problems can often be solved by a combination of inspection and common sense. If the problems persist, then vibration experts with appropriate vibration measurement equipment can be called in. Make sure that these experts focus on solving the problem at hand instead of just gathering data. Prior to calling in consultants, the following procedures can serve as a guide, but be prepared to modify them to fit the particulars of your situation.

Basic Troubleshooting Procedures

Obtain a history of the problem: Is it recent or has it always occurred? Has the system or its operation recently changed? If the problem is recent in an existing installation, it is likely to be caused by looseness, clogging, erosion, or wear. Pump disassembly may be required for answers, but even so, delay for the moment. Before dismantling the pump for inspection, follow the procedures given below as far as conditions permit, and answer the following questions: (1) Does the problem relate to flow, the sump level, or some other circumstance? (2) Is the vibration constant or intermittent? (3) Have components been failing? (4) Which locations shake beyond the specifications, and which do not? (5) Is there a pattern to the locations that shake and/or make noise? A list of checking procedures follows, more or less in the order to be performed.

- A common fault is a pump operating off the BEP. Install a calibrated pressure gauge on the discharge and another on the suction pipe. Read static pressures and total dynamic pressures when the pump(s) is (are) running. It is helpful to obtain the flow rate (see Section 3-9). Compare with the manufacturer's curve for the impeller. Check for a worn impeller.
- Check for loose or broken connections, and faulty or improperly set valves. Check all bearing housing, foundation, and piping bolts, gasketed joints, and isolation pads for proper installation and tightness.
- Check for pump baseplate "soft foot" (bottoms of all feet not coplanar) by following the procedures on page 2.288 in the *Pump Handbook* [1].
- Inspect all baseplates and foundation blocks for significant cracking that might decrease machine support stiffness.
- Have a millwright check driver/pump shaft alignment.
- Check the shafts for straightness (e.g., by rotating the shaft by hand while a dial indicator is mechanically held against it).
- Listen near the pump inlet for the crackling sound associated with cavitation. Crackling noises heard near the pump suction suggest inlet cavitation due to insufficient NPSHA, severe impeller vane stalling, internal recirculation due to swirling or skewed flow at the inlet, operation too far from BEP, or submerged vortices in the sump. If any noise is detected, place a calibrated pressure gauge of the proper range close to the pump suction. Ensure that there is sufficient NPSHA by noting static pressure at a tap at the pump inlet flange, and compare it with the manufacturer's NPSHR versus capacity curve. Is NPSHA adequate? If not, perhaps the suction line is clogged or there is a sump problem. If, in spite of inlet crackling noise, NPSHA is adequate, recirculation or sump problems are suggested. Other clues are: (1) inlet cavitation noise is relatively constant, whereas (2) recirculation noise tends to be throbbing, and (3) sump noise tends to come and go without following any pattern.

- Place a calibrated pressure gauge of the proper range at the discharge, preferably about three to five diameters downstream. Use a certified pump system head/capacity curve or a flowmeter in the discharge line to determine flow rate. For variable-speed systems, use a tachometer or strobe to determine pump speed. At a given speed and flow, compare the discharge head minus the suction head with the value obtained from the last test performed, and to the manufacturer's head/capacity curve. If the TDH is larger than expected and/or flow is lower than expected, some form of discharge blockage is possible. If the TDH is lower than expected, wear of the wear rings or erosion of the impeller vanes or volute tongues is possible. If the TDH indicates that the operating point lies outside the range of 75 to 110% of BEP, especially after years of service, the impeller or the volute tongue may be excessively worn. If the station is new or if the vibration has always occurred, the designer may have used an unrealistic roughness coefficient for the piping.
- Ensure that expansion joints and flexible joints such as Dresser® couplings are constrained properly, especially if the pressure is high.
- What are the bearing shell or lubricant temperatures, at least approximately? Bearing lubricant temperatures are typically 60 to 82°C (140 to 180°F), and the shell and housing temperatures are usually 11°C (20°F) lower. The loss of oil viscosity at elevated temperatures can reduce bearing support film thickness and allow scuffing and excessive bearing wear to occur, and thermal growth mismatch between rotating and stationary parts may allow radial or axial rubbing. Excessive housing temperatures indicate severe pump/driver misalignment, need for lubricant addition or change, or (contrary to intuition) excessive grease in a packed bearing cavity.
- Are any wear particles or pumpage contamination visible in samples taken from the lubricant? The most common cause of bearing failure in pumps is lubricant contamination—usually by water passing a leaky seal or packing. Condensation from humid air onto the cool walls of an oil sump or inside a grease-packed bearing with badly worn lip seals that allow the free ingress of air is often overlooked. Sometimes the contamination is varnish, carbon deposits, or metal flakes from bearing surface fatigue. If the lubricant change interval has passed or if the bearing was overloaded or overheated for even a few minutes, the effect snowballs to more degradation.
- Following disassembly, if rubbing was involved, was it on just one side of the rotating or stationary components, or full circumference on each? Does corrosion appear to be a factor? If fracture occurred, are fatigue striations evident? If so, what does a high-power microscope indicate concerning how many cycles it took from crack initiation to failure? Such information is useful in determining whether the problem was some sort of brief transient event or is inherent in the pump or system. If fatigue occurred in relatively few cycles, there was a severe overload condition such as recirculation caused by running the pump below its recommended minimum continuous flow rate. Millions of cycles indicate gradual degradation, such as gradually worsening rotor imbalance or pump/driver misalignment. Microscopy performed by a qualified metallurgist can often determine the root cause of failure so that it can be avoided in the future.
- Check the balance and fit of the disassembled pump impeller and coupling hub. Poor fit can be remedied by reaming the bore and installing a sleeve of proper fit or, sometimes, by chrome plating or knurling the shaft followed by grinding to the proper dimensions. The impeller/shaft assembly should then be rebalanced.

More Detailed Test Procedures

If the above approach does not lead to a solution of the problem, detailed vibration testing may be required. In the most common type, "signature analysis," an accelerometer output is sent to a fast Fourier transform (FFT) analyzer to document the amount of vibration at each frequency within a tested range. Typically, this range is from several Hz to beyond the pump vane pass frequency. The frequencies at which most of the vibration is occurring and the locations where the vibration is the greatest are used as clues to determine the cause of the vibration.

Vibration testing often ends here. However, it is recommended that vibration testing includes experimental modal analysis (EMA). EMA involves artificially exciting a machine or structure (e.g., with an impact hammer), preferably while the machinery is running so that all bearing stiffnesses are representative (see Marscher [14]). The purpose of EMA is to determine the natural frequencies of a pump (or other machines), its rotor system, and the attached system components. These frequencies can be compared with excitation frequencies to ensure that resonance will not occur. They can also be used to confirm that all equipment and supporting structures have adequate separation margins between the excitation frequencies and natural frequencies.

Recognizing Vibration Problems

The great majority of pump vibration problems can be solved by: (1) re-balancing the rotor assembly, (2) alignment of the couplings when the system is at its rated conditions—especially if it is hot (see Dodd [15]), and/or (3) running the pump within the bounds of its specified head versus capacity curve. Remaining vibration problems are usually caused by a resonance of a pump internal natural frequency or a systemwide natural frequency. During resonance, the rotor vibrations can exceed internal clearances, and excessive bearing loads can occur, even if loads such as imbalance are within normally acceptable limits.

In performing vibration troubleshooting, generalized charts (such as those in the *Pump Handbook* [1]) for matching symptoms to possible causes can be useful for many typical or simple problems. However, do not rely too heavily on such lists, especially if their initial application does not lead to an immediate resolution of the problem. Persistent pump vibration problems are usually due to an unexpected combination of factors, some of which are specific to the particular pumping system, such as mechanical or acoustical piping resonances, or hot-running misalignment of the pump/driver due to thermal distortions of the piping or baseplate. With this warning, a list of some of the more common pump vibration symptoms and their causes is given in Table 22-1. The following predictive maintenance and troubleshooting list can be used for interpreting the reason for many common vibrations and therefore could be the kernel of any future predictive maintenance software. The list is *not* meant to be all-inclusive and is in the order of the observed frequency—not in the order of likelihood or importance for reliability.

Typical Fixes for Vibration Problems

Although many pump vibration problems are caused by operation off the BEP, most machinery vibration problems are traceable to excessive imbalance or misalignment. Determination of the reason for the imbalance or misalignment (and its subsequent elimination) is the cure in these cases. Most vibration problems that persist after proper balance and alignment, however, are related to some form of resonance.

When resonance occurs, simple trial-and-error field fixes, such as addition of gussets at appropriate locations for stiffening, can often be effective. Such "fixes" may, however, only shift the problem to a slightly different frequency. If "cut-and-try" methods cannot be made effective within a reasonable time span, it is usually cost effective to bring in a vibration expert from either the manufacturer or from a specialty consulting firm. The main priority in selection should be competence and experience. Fee differential is unlikely to compensate for your own personnel time, the cost of hardware changes, and lost operating time wasted by an inexperienced consultant.

In solving resonance problems, vibration experts may employ one of the following:

- Change the stiffness of the resonant component and/or its support (possibly including the floor). An increase in stiffness raises the natural frequency—possibly enough to drive it out of the resonance range. However, it is possible that the increase leaves the natural frequency still within the operating speed range but now excited by larger forces at the top of the speed range. It is also possible that a lower natural frequency, not previously a problem, can be moved into the operating speed range. For these reasons, it might be better to decrease rather than increase stiffness to drop the natural frequency below the operating speed range. Drawbacks to this latter approach are: (1) a higher natural frequency that was not previously resonant might drop into the operating speed range, and (2) decreasing stiffness may make static forces (such as weight) and alignment more difficult to manage.
- Add weight to locations of the largest vibratory motion, thus decreasing the problem natural frequency. Measurements need to be taken to confirm whether the new frequency is out of the operating speed range. Potential drawbacks are greater sensitivity of the revised structure to seismic loads and the possibility of reducing a previously higher natural frequency into the operating speed range.
- Install vibration isolators between the vibration excitation source and the problem vibration location by adding flexibility between the two. Be careful to ensure that the vibration is sufficiently diminished in all potential problem locations and that the isolation does not hide an excessive excitation force that could cause other problems if untreated. The use of isolators is not often practical.
- Design and attach a dynamic absorber (made of a dead weight and a suspending rod) to a point of maximum motion for the problem natural frequency's mode shape. The mass and adjusted rod stiffness is chosen so that the mass/rod reed frequency equals the problem natural frequency. If the system is tuned just right, the motions of both natural frequency/mode shapes cancel each other at

Table 22-1. Troubleshooting Tips for Identifying the Source of Vibration Problems

Frequency of vibration (in terms of multiples of running speed)	Other symptoms	Probable causes
0.05× to 0.3× (may be broadband)	Vibration response is unsteady and fluctuates somewhat in frequency.	Diffuser or volute wall stationary passage stall.
Exactly ½×, ⅓×, or ¼×	Vibration may increase dramatically shortly after the above (0.05× to 0.3×) appears, except in vertical pumps with four or more lineshaft bearings, where it is common, and generally not harmful.	Light rub or unexpected looseness combined with shaft or bearing support natural frequency.
0.42× to 0.48×	Near shaft natural frequency, and orbit "pulses," forming an inside loop. Vibration onset is sudden at a speed roughly twice the excited natural frequency, and locks onto the natural frequency in spite of speed increase.	Rotodynamic instability due to fluid whirl in close clearance, e.g., "shaft whip."
0.6× to 0.93×	Smaller peaks at f× and at ±(1 − f)× "sidebands" of the first several multiples of running speed where f is the frequency of the amplitude modulation caused by pressure pulsations. Amplitude modulation is manifested as a beat frequency f. Often accompanied by rumbling noise and beating. Occurs at capacities below BEP, but improves at very low capacity. Independently, depends on speed and flow.	Internal flow recirculation in the impeller, probably at its suction, but possibly at the discharge.
Less than 1×	Increased broadband vibration and noise level below running speed as NPSHA decreases, especially at high flows. Often accompanied by decreased vibration and noise above 1×.	Cavitation without recirculation.
1×	(1) Stronger on shaft than on housing. Hydraulic performance and/or suction pressure normal. Axial vibrations within normal limits and vibrations above runout increase roughly with speed squared.	Imbalance in rotating assembly.
	(a) Vibration highest on drive and pump IB (inboard) housing.	Coupling imbalance.
	(b) Vibration highest on driver IB housing.	Driver rotor imbalance.
	(c) Vibration high on machine inboard (IB) or outboard (OB) housing, low on driver.	Machine imbalance.
	(d) Natural frequency, 1×.	Resonance.
	(2) Axial vibration is over ½ of radial vibration, or/and vibration increases much slower than the square of the speed. Also, bearing temperature is high.	Machine/driver misalignment at the coupling.
	(3) Discharge pressure pulsations are strong at 1× but not at impeller vane pass in a single volute machine.	Clogged or damaged impeller passage.
	(4) Same as (3), but vane pass also strong, especially at flows far above or below the design point.	Volute tongue designed too close to impeller OD, or clogged or damaged volute, or excess impeller eccentricity.
2×	(1) Axial vibrations are low.	
	(a) Both shaft and housing vibrations are strong, and discharge pressure pulses are strong, but impeller vane pass vibrations are low in a twin volute pump.	Clogged or damaged impeller passage.

Table 22-1. (Continued)

Frequency of vibration (in terms of multiples of running speed)	Other symptoms	Probable causes
2× (*Continued*)	(b) Same as (a), but with unusually high vane pass vibration and discharge pulsations.	Volute vanes designed too close to impeller OD, clogged or damaged volute, or excessive impeller/volute eccentricity.
	(c) Shaft vibrations stronger than housing vibrations, maybe exceeding clearance. Decrease in shaft first natural frequency. Multiples of running speed may be strong.	Looseness in bearing retainer, or cracked shaft.
	(d) Vibration of one bearing housing high, shaft may be quiet.	Loose or cracked bearing housing.
	(e) Shaft vibrations stronger than housing.	Torsional excitation.
	(f) Motor driver frequency equals electrical line frequency and highest vibration on motor.	Electrical problem with motor.
	(2) Axial vibrations are over ½ of horizontal, 1× vibrations are also high, and bearing oil temperature is high.	Pump/driver misalignment at the coupling.
Number of impeller vanes × running speed	(1) Discharge pressure pulsations reasonably high and both shaft and housing vibrations high.	Volute too close to impeller OD due to design or excessive rotor eccentricity.
	(2) Same as (1), but housing vibrations much higher than shaft vibrations.	Piping mechanical resonance at vane pass or loose bearing housing.
	(3) Discharge pressure pulsations high at vane pass frequency but suction pressure pulsations reasonably low.	Acoustic resonance in discharge pipe.
	(4) Suction pressure pulsations high.	Acoustice resonance in suction pipe.
	(5) Rotor or casing natural frequency close to vane pass.	Resonance.
Several multiples of running speed, including 1×, 2×, 3×, 4×, and possibly higher	(1) Orbit shows sharp angles or shows evidence of "ringing," and/or spectrum shows exactly ½ or ⅓ ×. Grinding noises may occur.	Internal rub or poorly lubricated gear coupling.
	(2) Orbit is fuzzy but does not pulse or ring. Seal coolant flow is unexpectedly high or low, and may exhibit high temperature. Spectrum may also exhibit ½ or ⅓ ×.	Jammed, clogged, or damaged seal.
	(3) Orbit pulses, usually in one direction much more than the other and shaft vibrates more than the housing. May also observe at exactly 0.5×, 1.5×, 2.5×, etc.	Shaft support looseness, especially in bearing insert or cap retention.
	(4) Shaft orbit fairly steady, and housing vibrates more than shaft. Often combined vibration over a broad range of low frequencies.	Looseness in casing, pedestal, or foundation or bearing housing.
Non-integer multiples of running speed	(1) Vibrations stronger at problem frequency at certain points in the piping or on the foundation than on the pump.	System structural resonance.
	(2) Vibrations high in piping but not foundation. Suction or discharge pressure pulses as strong or stronger relative to background of spectrum than the vibration is in the piping or the machine.	Hydraulic piping dynamics.
	(3) Vibrations low in piping and foundation at a fluid machine natural frequency.	Machine resonance, excited by turbulence.

the running speed and dramatically reduce the vibration level at the absorber attachment point. The disadvantages of this approach are: (1) it can be unsightly, (2) if the absorber mass is not large enough it can be difficult to maintain proper adjustment, and (3) the concept works only for constant-speed applications. The use of dynamic absorbers is tricky and best left to vibration experts.

Vibration problems that are not related to imbalance, misalignment, or resonance are usually caused by excessive hydraulic forces. If the cause is cavitation from insufficient NPSHA, the NPSHA should be increased to the manufacturer's requirement or even more. If vibration is caused by excessive vane pass interaction, then the impeller/stator vane gap may be too small, and the gap should be opened to about 10% of the impeller diameter (if acceptable to the manufacturer and if the required flow rate and head can be met). If a centrifugal pump is being run at relatively low flow, suction or discharge recirculation may be at fault—possibly with accompanying rotating stall. If the flow cannot be increased, then a bypass line can be placed between the pump discharge and a location in the suction plenum or suction line at least five pipe diameters upstream from the pump suction flange. A valve can be placed on the bypass line, and bypass flow can be increased gradually until the vibration problem ceases. Alternatively, a new impeller less sensitive to low flow can be installed. Although efficiency may decrease several percent, the operating cost is usually less than it is with a bypass line.

Description of Basic Vibration Terms, Concepts, and Equipment

Vibration analysis of machinery and its piping and foundations may be divided into two parts:

- *Structural dynamics*—the vibration of stationary components such as housings, piping, and supporting structures. An important factor in considering structural vibrations is that the amount of vibration at the point of interest may be strongly related to other system components far removed.
- *Rotordynamics*—the vibration of the rotor assembly. The rotor vibration is influenced by all factors relevant to structural vibration but, in addition, may be strongly affected by unexpected reaction forces and gyroscopic effects.

Concepts of key importance for both structures and rotordynamics in all types of rotating machinery are defined and explained below:

- *Excitation forces* supply the energy in a vibrating system. They include imbalance, at a frequency of one times running speed and misalignment, usually at both one and two times running speed. In machinery with vanes or lobes, vane-passing pulsations occur at the number of impeller vanes times running speed. In centrifugal pumps, the level of vane-passing pulsations (and the vibrations they cause) are dependent on the ratio of the capacity at the BEP to the actual pump flow rate. As the departure from BEP increases, the flow path directions inside the pump become less well matched with the impeller vane and volute tongue angles. Running a centrifugal machine at flows below its rated capacity at a given speed causes increased stress and vibration, and decreased machine life and reliability as dramatically stated by Agostinelli et al. [16]. In contrast, the pulsations of screw and reciprocating machines usually increase with load. Strong pulsation frequencies in screw pumps occur at one and two times the number of screw lobes, whereas in reciprocating machines, strong pulsations occur up to four or more times the plunger or cylinder stroke frequency (a value equal to the number of cylinders times running speed).
- *Natural frequencies* are those frequencies at which mechanical systems vibrate freely after energy is imparted, such as after striking a tuning fork. Machinery systems have not only one, but an infinite number of natural frequencies. Only natural frequencies close in frequency to strong excitation forces are of practical significance and merit the designer's or user's attention.
- *Mode shapes* are the vibrating movements associated with a given natural frequency. Each natural frequency has a unique mode shape. The mode shape becomes more complicated as the natural frequency becomes higher. At lower frequencies, simple mode shapes are most likely to possess motion consistent with the action of exciting forces, and these are the most troublesome to the plant designers and maintenance engineers. The lowest natural frequency of vertical pumps and motors resembles a reed blowing in the wind and is, therefore, called the "reed frequency" (ANSI/HI 9.6.4-2000). The second natural frequency has an initially similar motion that starts at the base in one direction, but then it turns and finishes with reed motion at the top in the opposite direction. The lowest natural frequency in horizontal pumps is usually a swaying back-and-forth motion of the pump casing allowed through flexure of its support base, and a second natural frequency is a see-saw motion of the pump casing (as viewed from above).

Both sets of these vertical and horizontal pump mode shapes and natural frequencies are often strongly influenced by (1) the stiffness and/or mass of the foundation and building floor, and (2) the added mass of rigidly attached piping. Hence, the floor and piping should be considered together with the pump as a total, connected system. Remember these statements during the design process and during troubleshooting.

- *Damping* is the absorption of vibration energy. The strongest damping typically occurs in components where there are tight fluid-filled clearances such as in journal bearings and impeller wear rings. Handbook charts and readily available computer programs exist to estimate and optimize the damping of bearings and seals.
- *Resonance* occurs when the frequency of an excitation force equals the natural frequency. Resonance stores vibration energy from cycle to cycle. Very large vibrations build up during resonance even if the excitation forces are reasonably low—especially if the force is applied near a point of maximum vibration in the natural frequency's mode shape and if the damping is small. The affected machine is then observed to be "balance-sensitive" or "alignment-sensitive," because the resulting high vibration can be temporarily eliminated by very fine balancing or alignment. However, as these vibrations degrade slightly after brief operation, the vibration problem usually returns.
- *Amplification factor* refers to how much more (or less) a pump or system component moves during vibration than it would move if the force causing the vibration were static instead of oscillating. If the excitation frequency is well below the system's first natural frequency (<75%), the amplification factor is about 1—that is, the structure's maximum motion in response to the force is essentially the same as if the slowly oscillating force were static. Systems responding as though to a static force are known as "stiffness-dominated" or "rigid structures," and they are unlikely to have balance-related resonances (by far the most common type of resonance problem). At the other extreme are systems with important excitation frequencies well above (by at least 25%) those natural frequencies that have simple, easily excited mode shapes. These are known as "mass-dominated" or "flexible structures," and the amplification factor for them is less than 1. Hence, there is even less vibration motion than would be expected if the excitation force were applied statically. Although amplification factors less than unity sounds like an ideal situation, the high degree of flexibility implied in such systems makes them very prone to distortion from static forces and may lead to serious misalignment between the driver and the pump. The distortion may be accompanied by internal rubs and possible large dynamic misalignment in spite of the low amplification factor. Between the rigid and flexible structures are "resonant" structures, in which the excitation force frequency and natural frequency are close to each other—a situation to be avoided because of the high amplification factor and consequent high vibration levels.
- *Parametric resonances* refer to large nonlinear vibrations that can occur in response to looseness or rubbing. Such resonances are typically caused by bearing support looseness or a rub at a bearing, seal (or other close running clearance), or slip in a gear or spline coupling. The symptoms are a pulsating orbit, with a large amount of vibration at exact whole fractions of running speed, such as $^1\!/_2$, $^1\!/_3$, $^1\!/_4$, etc. In addition to the above, rotor systems may be strongly affected by more subtle phenomena. The most important of these phenomena are listed below.
- *Gyroscopics* describes the reluctance, due to conservation of angular momentum, of a rotating object of large radius (such as an impeller) from changing the direction of its axis.
- *Cross-coupling* is a force that develops perpendicularly to rotor motion when the rotor moves to an eccentric position within its bore clearances. It is caused by the higher pressure of the flow dammed up upstream from the minimum clearance pinch point, and it occurs in combination with the restoring bearing force that acts to support the rotor opposite in direction to the motion. The non-intuitive cross-coupling force acts in the opposite direction of the damping vector and can become an unexpected factor to cause decreased energy dissipation due to the vibration—in some circumstances to the point that the vibration becomes self-excited and unstable. After a certain threshold speed is passed, the self-excitement produces "rotor dynamic instability," in which the rotor orbit takes up the entire clearance, whirls at about half rotor speed, and rapidly wears out the machine. Rotor-dynamic instability is extremely destructive, but, fortunately, it is not common in the type of equipment used in pumping stations.
- *Added mass* refers to the effective inertia of the fluid surrounding the rotor. It is developed from three sources: (1) the fluid trapped in the impeller passages adds mass directly; (2) the fluid displaced by the presence of the impellers and shaft adds its

mass because, as the rotor vibrates within the fluid, it must displace this mass; and (3) the fluid in close clearances must accelerate off to the sides much faster than the rotor vibration acceleration to make room for the rotor motion. Item (3) can, potentially, add mass many times its displaced mass.

- *Lomakin effect* is an unexpected support force that occurs in pumps at annular seals such as wear rings due to the action of Bernoulli's effect during the normal leakage process. This effect, discussed in detail by Black [17] and Marscher [18], can change the rotor support stiffness dramatically and hence the rotor natural frequencies. The effect thereby either avoids or induces possible resonance between strong forcing frequencies at one and two times the running speed and at one of the lower natural frequencies. Although the effect is usually beneficial, it strongly depends on the annular seal diametral clearance, and loss of this clearance due to erosion and wear in service can lead to loss of Lomakin effect and the appearance of a resonance problem where there was none before.

Vibration Measuring Equipment

Modern vibration test equipment can be divided into three levels of detail and sophistication:

- *Hand-held sensor/meter packages*. The advantages of this equipment are that it is easily portable, easy to use, inexpensive, and the sensor is relatively lightweight and rugged. The disadvantages are (1) the frequency information it provides is not very distinct; (2) it is difficult to distinguish hydraulic forces and mechanical resonances from excitation harmonics of multiples of running speed caused by imbalance and misalignment; (3) only one probe at a time can be used, so the powerful modal testing discussed by Marscher [14] cannot be performed; and (4) the velocity probes used in these units rapidly lose accuracy outside of the range 10 to 17 Hz.
- *Single-channel FFT or real-time analyzers*. These devices are rather complicated and best used by experts. Velocity probes can be used with them, but accelerometers are better because they (1) have a broader accurate range of frequency (2 to 17,000 Hz), and (2) are robust. FFTs may also take input from eddy-current shaft proximity probes that sense the displacement of a rotating shaft relative to some stationary point on the pump housing. The advantages of the single-channel FFT are (1) it is moderate in cost; (2) it gives accurate plots of running vibrations versus frequency (thereby allowing the broader frequency range taken up by resonances and hydraulic forces to be detected relative to the narrow frequency range vibration response of mechanical forcing frequencies); and (3) it provides enough frequency resolution to distinguish subtle but important frequency differences between resonances, hydraulic forces, and mechanical forces. An important disadvantage of single-channel FFTs is that experimental model analysis (EMA) testing and recording of shaft position versus time "orbits" cannot be obtained. Another disadvantage is that the timing relationship or frequency correlation between pressure pulsations or vibrations in a possible "problem source" area and vibrations in the problem "symptom area" cannot be obtained. These procedures often play an important role in vibration problem solving.
- *Multi-channel FFT or real-time analyzers*. As with single-channel FFTs, these are also based on the fast Fourier transform. They are heavier (7 to 36 kg or 15 to 80 lb) and cost more than single-channel units, but they allow measurements in two or more channels at once, so that the EMA, orbit, and correlation measurements mentioned above can be performed. As with single-channel FFTs, any type of voltage-output transducer can be used, including velocity probes. The more accurate accelerometers and displacement proximity probes are, however, preferred.

Data Types and Formats

Vibration test data are usually plotted in three different forms:

- A Cartesian plot of vibration amplitude versus frequency ("signature plot" or "spectrum"). Sometimes this plot may be combined with a plot of the phase angle (the "lag angle" from application of the exciting force to the responding vibratory motion) versus frequency. It is then called a "Bode plot."
- A Cartesian plot of vibration amplitude versus time, similar to a typical oscilloscope trace ("time" plot).
- A polar plot of vibration versus time in a plane perpendicular to the shaft axis (an "orbit").

The amplitude scales may be linear or dB (base 10 logarithm). A dB scale is often used to improve the resolution of natural frequency peaks. The amplitude

scale generally represents RMS (root mean square) values unless specifically scaled otherwise. To convert RMS values at a specific frequency to peak-to-peak, multiply by 2.83, and to convert from peak-to-peak to zero-to-peak, divide by 2.

22-4. Introduction to Vibration and Noise Calculations

The goals of the following 11 sections are to provide for a deeper understanding and appreciation of vibration, to present means for evaluating a particular situation, and to offer suggestions for reducing vibration or noise to an acceptable level.

The calculation procedures presented here are generally simplified expressions that are meant only to give an idea of the severity of a given situation. If accurate predictions of noise and vibration levels are required, a qualified specialist should be consulted. It may not be easy to find qualified specialists, but a good place to start may be either the National Council of Acoustical Consultants, the Institute of Noise Control Engineers, or the Vibration Institute.

Nomenclature

Many of the following symbols are unique to this chapter and, hence, are omitted from Chapter 2.

Symbols

a	Acceleration amplitude of a vibrating body [m/s^2 (in./s^2)].
B	Bulk modulus of fluid [Pa (lb/in.^{2})].
c	Speed of sound in air or speed of wave propagation in fluids or solids [m/s (ft/s)].
cm	Centimeters
d	Static deflection of vibration isolator [mm (in.)].
dB	Decibel.
dBA	A-weighted decibel.
D	Diameter of pipe or resonator [mm (in.)] (see ID and OD).
D_n	Noise dose (dimensionless).
e	Wall thickness of pipe [mm (in.)].
e_o	Radius of motion of eccentric mass [mm (in.)].
E	Modulus of elasticity [Pa (lb/in.^{2})].
E_t	Total vibrational energy [N · m (ft · lb)].
f	Frequency (Hz).
f_d	Pump driving frequency (Hz).
f_n	Resonance frequency of a multiple degree-of-freedom system (Hz).
f_o	Natural frequency of a single-degree-of-freedom system (Hz).
F	Dynamic force [N (lb)].
g	Acceleration of gravity [9.81 m/s^2 (386 in./s^2 or 32.17 ft/s^2)].
G	Modulus of elasticity in shear [Pa (lb/in.^{2})].
Hz	Hertz (frequency in cycles per second).
I	Moment of inertia [m^4 (in.^{4})].

ID	Inside diameter (of a pipe).
J	Mass moment of rotational inertia [$kg \cdot m^2$ ($lb_m \cdot in.^2$)].
k	Spring stiffness constant [N/m (lb/in.)].
K	Torsional rigidity [$N \cdot m/rad$ ($lb \cdot in./rad$)].
L	Length of pipe or shaft between support points [m (in.)].
lb	Pounds force.
lb_m	Pounds mass $= lb/g = lb \cdot s^2/32.2\,ft$.
L_p	Sound pressure level (dB or dBA).
$L_{p(rev)}$	Reverberant field sound pressure level (dB or dBA).
L_{dn}	Day–night sound level (dB).
m	A small mass in kg (lb_m).
m	Mass per unit length [kg/m ($lb_m/in.$)].
M	Mass [kg (lb_m)].
NR	Noise reduction (dB).
NRC	Noise reduction coefficient (dimensionless).
NRC_a	NRC of added acoustical material (dimensionless).
OD	Outside diameter (of a pipe).
p	Acoustic pressure (μPa).
P	Static pressure of fluid in pipe [Pa ($lb/in.^2$)].
R	Radius of gyration [m (in.)].
rad	Radian (1 rad = 57.3 degrees).
RMS	Root mean square.
S	Surface area [m^2 (ft^2)].
S_o	Cross-sectional area [m^2 ($in.^2$)].
S_a	Surface area of added acoustical material [m^2 (ft^2)].
STC	Sound transmission class (dB).
STC_c	Composite (or effective) STC (dB).
T	Repetition time (period) of a vibrating object.
T	Temperature.
T	Transmissibility—the ratio of transmitted vibrational force with isolators to that without isolators.
v	Velocity amplitude of a vibrating body [m/s (in./s)].
w	Weight of pipe or shaft per unit length [N/m (lb/in.)].
W	Weight of a vibrating body [N (lb)].
x	Displacement amplitude of vibrating body [m (in.)].
γ_n	Coefficient (dimensionless) for bending resonances in piping (the subscript n is the resonance order).
δ	Damping coefficient [kg/s (lb_m/s)].

δ_c Critical damping coefficient—the minimum damping required to allow a displaced system to return to the rest position without undergoing oscillations [kg/s (lb_m/s)].

δ_r Damping ratio—the ratio of the damping coefficient to the critical damping coefficient (dimensionless).

ε Efficiency of vibration isolator (percentage).

λ Wavelength, m (ft).

μPa Basic unit of acoustic pressure (1 μPa $= 1.45 \times 10^{-10}$ lb/in.^{2}).

ρ Fluid density [kg/m^3 (lb_m/ft^3)].

ω Angular frequency [rad/s ($\omega = 2\pi f$)].

Definitions

A-weighted: A commonly used frequency weighting that closely approximates the frequency response of the human ear.

Absorption: Absorption refers to the conversion of acoustical energy into other forms of energy (usually heat). Sound-absorbing materials are usually rated by the noise reduction coefficient (NRC), a numerical value (usually between 0.1 and 1.0) that approximates the ratio of acoustical energy absorbed to the energy incident upon the material.

Acoustic pressure: The small pressure fluctuations that occur about the static atmospheric pressure (101.3 kPa or 14.7 lb/in.^{2} at sea level) that is sound.

Amplitude: A quantitative descriptor of the level or strength of a noise or vibration signal or waveform.

Attenuation: The reduction in amplitude of a noise or vibration signal.

Blade passage frequency: The frequency associated with the motion of individual blades of a fan or pump. It is the same as the pump frequency for a centrifugal pump.

Critical damping: The minimum damping required to prevent oscillation in a displaced mechanical system.

Critical speed: Any rotational speed of a shaft that excites a resonance.

Damping: The mechanism by which energy is removed from a vibrating system.

Day–night sound level: The energy average of the A-weighted sound pressure level measured over a continuous 24-h period with the sound levels biased upward 10 dB between the hours of 10:00 P.M. and 7:00 A.M.

Decibel: A dimensionless unit used to quantify a relative amplitude of an acoustic or vibration signal. Decibel is also used as the basic unit for the sound pressure level that is relative to a standard reference pressure of 20 μPa.

Direct field: The region near the source of a sound that is not influenced by the acoustical characteristics of other surroundings (e.g., room boundaries). In this region, the sound pressure level decreases approximately 6 dB with every doubling of distance (valid only for sources that are small in size compared with the distance).

Directivity: Directional radiation properties of a source that may make more noise in one direction than in another.

Displacement: The physical change in the position or angle of a body or particle as measured from its normal rest position.

Driving frequency: The frequency of forced vibration.

Dyne: The unit of force that causes 1 g to be accelerated at 1 cm/s^2(= gram/9.81 m/s^2).

Frequency: The inverse of the time required for a body or particle in a vibrating system to go through a full cycle of motion.

Fundamental: The mode of vibration with the lowest natural frequency.

Harmonic: A signal that is usually generated along with the fundamental. The frequency of a harmonic is always an integer multiple of the fundamental frequency.

Inertia base: A heavy mass (usually concrete) added to the frame of a piece of vibrationally isolated equipment.

Mode: A natural, repetitive pattern of harmonic motion in which every particle moves at the same frequency.

Natural frequency: The frequency of the natural or normal modes of vibration of a vibrating system. A system at rest will vibrate only at its natural frequencies after being displaced by an impulsive force.

Node: Any point in a vibrating system where the particle motion is zero for a given mode of vibration.

Period: The time required for vibrating particles to complete a full cycle of motion for a given mode.

Phase: The relative time or displacement between two points in a vibrating system. Any two points are said to be "in phase" (i.e., the phase shift is 0 degree) if their relative motion is always in the same direction at the same time.

Pump frequency: The primary frequency generated by the pump in hertz (cycles per second). It equals n z rev/s where n is the number of pressure pulses generated in the fluid with each full rotation of the shaft.

Resonance: A condition in which a vibrating system responds with maximum amplitude to a driving force. This condition occurs only at the natural frequencies of the vibrating system.

Reverberant field: The region in a room (usually far from the source) where the sound pressure level is reasonably constant with position and distance from the source.

Rotor frequency: The frequency (in hertz) of the shaft of a pump or other rotating machine.

Short circuit: Any mechanism that allows vibrational energy to bypass a vibration-isolating device, thus rendering it ineffective (e.g., a spring compressed until the coils touch).

Sound pressure: A synonym for acoustic pressure. The sound pressure level is defined as 10 times the common logarithm of the square of the ratio of the weighted or unweighted acoustic pressure to the standard reference pressure (20 μPa).

Sound transmission class: A single-number rating for a material that describes the ability of the material to resist sound transmission.

Spectrum: The frequency distribution of a sound or vibration signal.

Standing wave: A periodic wave that does not appear to propagate but remains stationary. The wave is actually formed by two or more progressive (moving) waves of equal frequency traveling in opposite directions.

Stiffness: The ratio of the change in force to the change in displacement or deflection of an elastic element.

Static deflection: The deflection of an elastic element caused by the static (dead) load of the mass it supports.

Transmission loss: A value (usually expressed in decibels and a function of frequency for a given material) proportional to the ratio of incident to transmitted energy in a sound- or vibration-isolation system. A wall or barrier provides a transmission loss of sound. A spring or rubber mount is usually involved in the transmission loss of vibration.

Wavelength: The distance between "in phase" positions on an acoustic or vibration wave at any given instant. The wavelength is related to frequency, f, via the wave propagation speed, c, as $\lambda = c/f$.

Wave resonances: Resonances in vibration isolators that can permit the transmission of noise through the vibration isolator with little or no transmission loss. Wave resonances usually occur only at frequencies much greater than the fundamental resonance frequency of the vibration-isolation system.

22-5. Vibration and Noise Characteristics

To develop a good foundation for discussing vibration and noise issues in a pumping station, it is important to understand the characteristics and sources of vibration and noise as well as the techniques for measuring these quantities.

In reciprocating machinery such as engines, vibration is primarily due to the dynamic forces exerted on the machine by the reciprocating and rotating internal parts (pistons, crankshaft, etc.). The vibration levels of this equipment can be relatively high. The spectrum is typically dominated by the fundamental frequency (e.g., the cylinder firing rate in a diesel engine) and usually contains several (3 to 10 or more) prominent harmonics.

In centrifugal equipment, vibration is caused by unbalanced rotating parts, misaligned shafts, and fluctuations in the load. The vibration of centrifugal equipment is typically lower in level than comparable reciprocating machinery, and the spectrum usually contains only the fundamental shaft frequency, the blade passage frequency, and one or two harmonics.

Piping vibration is usually caused by the transmission of vibrations from the primary equipment to which it is connected. Piping can be excited by the pipe wall contact with the pump inlet or discharge (even through a flexible connection) or by the pressure fluctuations within the fluid in the pipe. In addition to the frequencies associated with the pump, pipes also vibrate at their own natural or resonance frequencies.

Building vibration results from dynamic forces within the primary vibration elements (i.e., equipment and piping). In addition to vibrating at the frequencies associated with the equipment and piping, the structural elements of the building also have their own natural frequencies. Excessive vibration of the building's structural elements (floors, walls, etc.) is usually a potential problem only when a structural resonance frequency is very nearly equal to one of the

driving frequencies associated with the primary equipment. It is also possible to transmit vibration into the earth and, eventually, into adjacent structures. Aspects of this problem are beyond the scope of this chapter.

Translational Vibration

Translational vibration is usually measured with a contact sensor (an accelerometer or velocimeter), which is a small device rigidly attached to the vibrating body. The accelerometer produces an electrical output (voltage) proportional to the instantaneous acceleration of the body, whereas a velocimeter produces a voltage proportional to the velocity. Translational vibration problems in pumping stations almost always occur at frequencies below 100 Hz, and usually the problem is associated with a resonance frequency.

The waveform describing the translational motion of a vibrating body at resonance is usually sinusoidal, as shown in Figure 22-1. The three curves describe exactly the same motion (displacement, velocity, and acceleration) of a single point on the object that is vibrating—all as a function of time. All curves have the same shape and repetition period of T seconds. The only difference between these curves is the amplitude (vertical) scale and the relative phase (shift in time).

The acceleration, velocity, and displacement amplitudes of a sinusoidal signal are related to each other by

$$x = \frac{v}{2\pi f} = \frac{a}{4\pi^2 f^2} \qquad (22\text{-}1)$$

where x is the displacement amplitude in meters (inches), v is the velocity amplitude in meters per second (inches per second), f is the frequency in hertz, and a is the acceleration amplitude in meters per second squared (inches per second squared). Vibration levels are sometimes reported as peak levels (the maximum deviation from the zero or "at rest" value), as peak-to-peak levels (the maximum total swing from positive to negative values), or the root mean square (RMS) value. The RMS value is 0.707 times the amplitude (or peak value) for sinusoidal waveforms (as shown in Figure 22-1).

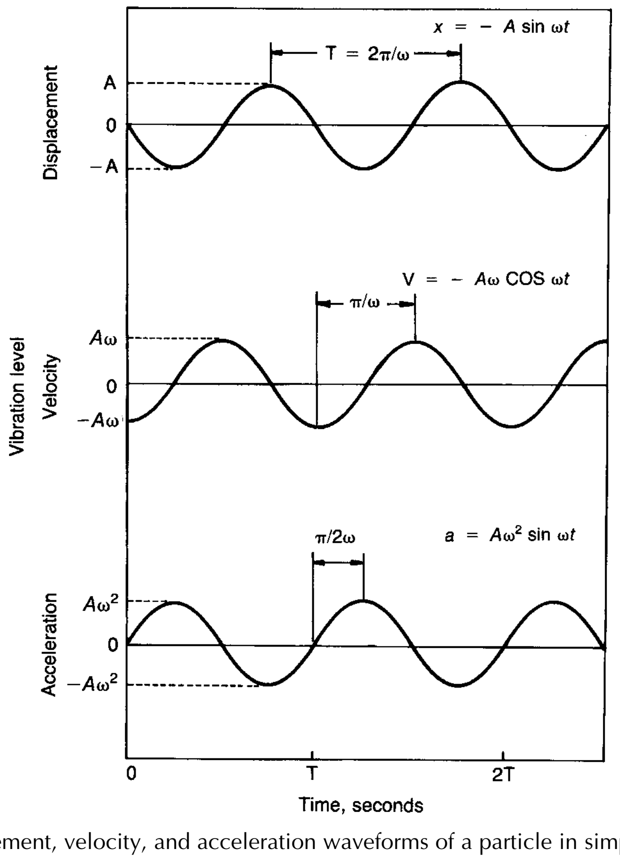


Figure 22-1. Displacement, velocity, and acceleration waveforms of a particle in simple sinusoidal motion.

For example, an object vibrating sinusoidally with a peak displacement of 1 mm (0.04 in.) at a frequency of 10 Hz (600 cycles/min) could be described accurately by any of the following (from Equation 22-1):

- 62.8 mm/s (2.47 in./s) peak velocity at 10 Hz
- $3.95\,\mathrm{m/s^2}$ (12.8 ft/s) peak acceleration at 10 Hz
- 0.0444 m/s (0.571 ft/s) root mean square velocity at 10 Hz
- $7.90\,\mathrm{m/s^2}$ (0.805 g) peak-to-peak acceleration at 10 Hz.

An important concept to keep in mind when analyzing vibrating systems is the buildup and dissipation of energy in the system. As a power source begins operation from rest, the system vibration rapidly builds up to a steady-state level. After the system reaches the steady state, energy is dissipated at a rate equal to the rate at which energy is put into the vibrating system. This rate is usually only a very small fraction of the total power consumption of the source. The energy dissipation is in the form of conversion to heat via various damping mechanisms as well as energy transmission to remote systems (such as building structure and the earth).

The total vibrational energy present in the system can be determined by computing the instantaneous potential energy at peak displacement (when the instantaneous velocity is zero) or by computing the instantaneous kinetic energy at peak velocity (when the instantaneous displacement is zero). Either way, for a point mass the result yields

$$E_t = 2\pi^2 M x^2 f^2 \tag{22-2}$$

where E_t is the total vibrational energy in the system in Newton-meters (inch-pound force), M is mass of the system in kilograms (pounds mass), x is the displacement amplitude in meters (inches), and f is frequency in hertz (see Section 22-4).

Solving Equation 22-2 for the dynamic displacement yields Equation 22-3, which demonstrates that, for a fixed amount of vibrational energy, the displacement is inversely proportional to the frequency and the square root of the mass. Thus, if the mass and energy remain constant, doubling the frequency results in a 50% reduction in the dynamic displacement.

$$x = \frac{1}{2\pi f}\sqrt{\frac{2E_t}{M}} \tag{22-3}$$

By combining Equations 22-1 and 22-2, it can be seen that the vibration velocity, v, is independent of frequency at constant energy and the acceleration, a, is directly proportional to frequency at constant energy.

Torsional Vibration

Torsional vibration is difficult to measure because the shaft must be rotating during the measurement. The technique usually employed is to mount a lightweight, multitoothed gear near each end of the shaft at an accessible location and to install eddy-current sensors on a rigid foundation in close proximity to the gears. As the shaft rotates, the eddy-current sensor produces a pulsed output signal whose frequency is equal to the instantaneous shaft rotational speed in cycles per second (Hz) times the number of evenly spaced teeth on the gear. Torsional vibration of the shaft is detected as a periodic modulation of the output frequency, which indicates that the shaft speed is fluctuating at the sensor location. A frequency-to-voltage converter then transforms this pulsed signal into a low-frequency (usually below 100 Hz) waveform proportional to the instantaneous angular displacement in degrees or radians (or angular velocity in degrees per second or radians per second). This waveform can then be processed by conventional spectrum analysis techniques to obtain the torsional vibration levels at the frequencies of interest. Of course, for sinusoidal torsional waveforms, Equation 22-1 can also be used to convert angular displacement to angular velocity and angular acceleration if a consistent system of units is used.

Noise

Contrary to the characteristics of vibration, noise is usually only of concern at frequencies greater than 100 Hz. Noise problems are also almost never single-frequency problems and, as a result, some form of spectrum or frequency analysis is usually required to solve acoustical problems.

Noise is usually measured with a sound level meter, which is a hand-held instrument incorporating a microphone, an amplifying circuit, a frequency-weighting network, a display, and, perhaps, a set of filters (octave or one-third octave) for spectrum analysis. The microphone generates an electrical output proportional to the instantaneous acoustical pressure oscillations, and the sound level meter displays a frequency-weighted (or unweighted) average sound pressure level.

Noise levels are reported as a sound pressure level, L_p (in decibels), which is computed from the acoustic pressure, p, by

$$L_p = 20 \log_{10} (p/p_{ref}) \qquad (22\text{-}4)$$

where p_{ref} is the standard reference pressure of 20 μPa (0.0002 dynes/cm^2). The standard reference pressure is the approximate threshold of hearing for young adults in the ear's most sensitive frequency region (500–2000 Hz).

Subjectively, a noise level increase or decrease of 1 dBA is difficult to detect with the ear even though it does represent a 12% change in the acoustic pressure amplitude. A 10-dBA increase is generally perceived as a doubling of the loudness; conversely, a 10-dBA decrease is perceived as a 50% reduction in loudness. This nonlinear dependence is primarily due to the logarithmic response characteristic of the human ear.

Some typical A-weighted noise levels are given in Table 22-2. Noise levels can vary substantially with time, with location, and, in some instances, with changes in weather conditions. In reporting noise levels from an actual noise source or from an actual environment, it is extremely important to document the relevant conditions of the measurement, including the exact location of the microphone, the time and duration of the measurement, and the atmospheric conditions (primarily temperature, atmospheric pressure, wind speed, and direction). Without this back-up information, the validity of a noise measurement can be challenged.

Noise in and around pumping stations is usually broadband (meaning that it contains sounds at all frequencies simultaneously). A measure of an existing noise level is not complete unless the frequency weighting is also given. The most common frequency weighting is A-weighting, which closely approximates the frequency sensitivity of the human ear. Noise levels measured with this frequency weighting should be dimensioned as dBA for clarity.

To perform design calculations to predict noise levels with a reasonable degree of accuracy ($\pm$5 dBA), it is necessary to measure or have information concerning the frequency content (spectrum) of the noise sources under consideration. An explanation of this level of analysis is beyond the scope of this chapter; if this level of accuracy is required, consult a professional acoustical engineer.

Table 22-2. Typical Noise Levels

Noise source or environment	Noise level (dBA)
Quiet suburb	40–55
Busy city	60–75
Gasoline-powered lawnmower[a]	80–90
Diesel truck (at 15 meters)	75–95
Chain saw[a]	95–120

[a]At operator position.

22-6. Applicable Codes

Vibration

There are no established legal limits for vibration levels of equipment and piping associated with pumping stations, but there are guidelines for normally acceptable vibration levels of major pieces of equipment. Below 600 rev/min, the vibration criteria are based on peak-to-peak displacement; above 600 rev/min, the criteria are based on peak velocity (see Figure 22-1). The relationship between displacement and velocity is given by Equation 22-1. The Hydraulic Institute (ANSI/HI 9.6.4–2000) has published acceptable field vibration limits for clear liquid and nonclog horizontal and vertical centrifugal pumps. Vibration limits for reciprocating and rotary pumps are not available. Table 22-3 contains recommended vibration criteria (compiled from various sources) for major equipment used in pumping stations.

The vibration criteria listed in Table 22-3 refer to vibration levels at the shaft bearing housings. Vibration levels at other, less rigid portions of the machine may be higher. If the vibration level of a particular machine exceeds these values by more than 25% (particularly if the vibration level increases steadily with time), the unit and the system should be analyzed carefully to solve the problem. If the excessive vibration is present while the equipment on a rigid foundation is operating independently from the rest of the system, it should be returned to the manufacturer for balancing and/or alignment to correct the problem. If the vibration levels are only exceeded under load, the problem may be due to an interaction with another portion of the system.

Vibration criteria for piping systems are discussed in detail by Wachel and Bates [13] and are shown in Figure 22-2. The main danger of excessive piping vibration is failure due to repeated stress from distortion of the pipe. Stress in a piping system is primarily a function of the maximum displacement of the pipe as well as its vibratory mode shape. The allowable pipe displacement decreases with increasing frequency, as illustrated in the curves of Figure 22-2, because increasing frequency implies more vibrational energy input into the system and a greater number of stress cycles over the lifetime of the piping system. An alternative peak-to-peak pipe velocity criterion of 25.4 mm/s (1.0 in./s) has also been suggested

Table 22-3. Equipment Vibration Levels[a]

	Peak-to-peak displacement below 600 rev/min		Peak velocity above 600 rev/min[b]	
Source	mm	in.	mm/s	in./s
Centrifugal pumps[c]				
Clear liquid	0.125	0.005	7.8	0.31
Nonclog	0.20	0.008	12.5	0.50
Electric motors[d]	0.08	0.003	5.0	0.20
Fans				
Centrifugal	0.15	0.006	9.5	0.38
Axial	0.10	0.004	6.3	0.25
Generator sets				
Diesel	—	—	20	0.80
Gasoline	—	—	15	0.60
Air compressors				
Reciprocating	0.4	0.016	25	1.00

[a]Levels refer to filtered vibration amplitude at the vibration frequency and at the running speed (forcing frequency).

[b]See Equation 22–1 for the relationship to displacement.

[c]See ANSI/HI 9.6.4.4 for recommendations on vertical pumps.

[d]Limits apply to base-mounted motors. Use pumps for pump-mounted motors.

by Wachel and Bates [13]. This velocity criterion, which includes allowances with a nominal safety factor for typical conditions found in real piping systems, is also shown in Figure 22-2 along with the approximate threshold of human perception.

Vibration criteria for structures are discussed briefly by Harris and Crede [19]. They state that a peak-to-peak level of 25.4 mm/s (1.0 in./s) is generally regarded as an upper limit for safe, continuous vibration levels of structural floors. This is not to say

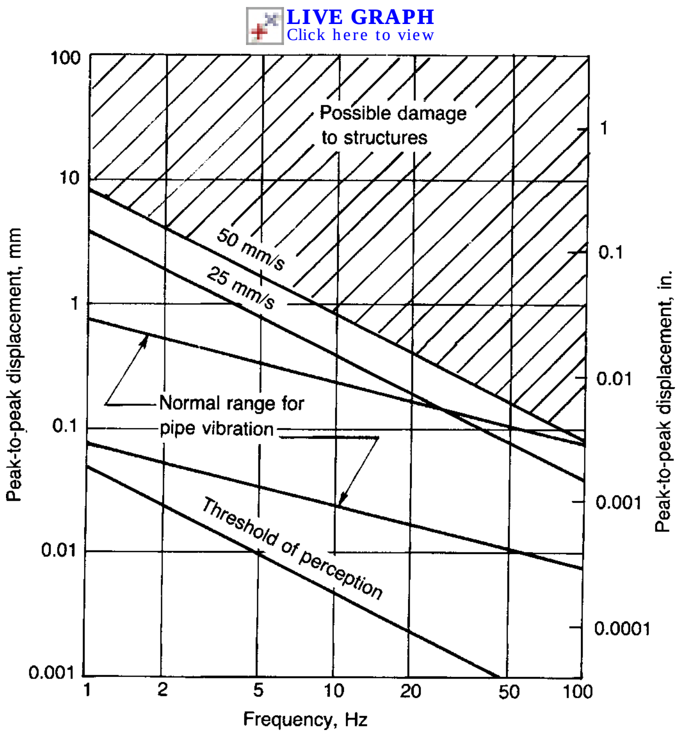


Figure 22-2. Vibration displacement criteria.

that structural failure will result at higher levels. During earthquakes, peak-to-peak vibration levels sometimes exceed 50 mm/s (2 in./s) without damage. But the probability of eventual damage increases as the vibration level increases. Continuous vibration levels exceeding 50 mm/s (2 in./s) present a serious threat to the safety of most structures.

Noise

Legal limits on noise generally fall into two categories: (1) occupational exposure, and (2) community exposure. Occupational noise exposure limits have been established by the federal government and are administered by OSHA [20]. The federal noise exposure regulations apply to employers who engage in interstate commerce, but most states have passed similar legislation extending these limits (some states are even more restrictive) to virtually all occupational environments. Compliance requires substantial design and investigative effort. The intent of the law is to prevent hearing loss that can be induced by exposure to high noise levels over extended periods of time. Reducing noise levels in the work environment is also advantageous for improved speech intelligibility, which in turn improves worker safety. The relationship between background noise level and the vocal effort required for adequate voice communication at various distances is shown in Figure 22-3.

In essence, the law restricts the length of time an employee can spend in a given noise environment during any 24-h period. Maximum allowable exposure times for various noise levels are listed in Table 22-4.

In most work environments, employees are exposed to noise that varies continuously throughout the day. The noise exposure under these conditions can be estimated by computing the noise dose, D_n,

$$D_n = \sum_{i=1}^{n} \left(\frac{C_i}{T_i} \right) \quad (22\text{-}5)$$

where C_i and T_i are the cumulative exposure time and maximum allowable exposure times for the ith noise interval. If D_n exceeds 1.0, the employee is overexposed. Sometimes, it may be necessary to install portable dosimeters on the employees to monitor the actual noise level and compute the noise dose on a continuous basis.

If working conditions in any facility exceed the local or federal regulations, each employee should wear approved hearing protectors (earplugs or earmuffs) whenever the noise exceeds 90 dBA to avoid

LIVE GRAPH
Click here to view

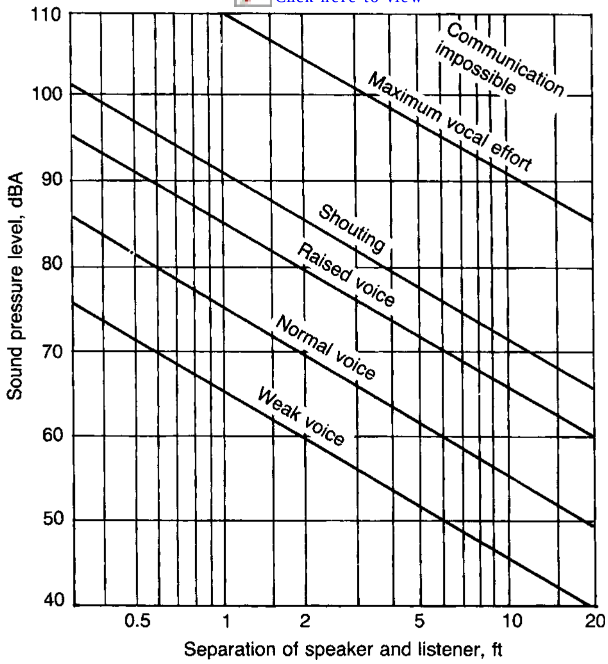


Figure 22-3. Relationship between voice and sound levels.

Table 22-4. Maximum Allowable Employee Noise Exposure

Exposure time (h/d)[a]	Noise level (dBA)
8.0	90
7.0	91
6.2	92
5.3	93
4.6	94
4.0	95
3.5	96
3.0	97
2.6	98
2.3	99
2.0	100
1.7	101
1.5	102
1.4	103
1.3	104
1.0	105
0.87	106
0.76	107
0.66	108
0.57	109
0.50	110

[a]Exposure times for all noise levels in the 80- to 130-dB range can be computed from the equation $T = 8/2^N$, where the exponent $N = (1/5)(L_p - 90)$.

permanent hearing loss. Good-quality hearing protectors provide an insertion loss (effective noise reduction to the inner ear) between 10 and 20 dB if they are installed properly. It is important to emphasize that extended exposure to high levels of noise can *permanently* damage hearing.

If the noise levels in a pumping station are expected to exceed federal or local regulations, steps should be taken during the design of the facility to mitigate the problem. These steps include the following:

- Design the facility in such a way that workers do not have to spend much time in noisy areas.
- Specify and/or select less noisy equipment.
- Install acoustical lining in noisy areas.
- Isolate noisy equipment with acoustical enclosures.
- Provide sound-absorbing barriers around noisy machinery.

Although there is no federal legislation affecting noise radiated into the environment, many local and state governments have such legislation to protect the welfare of the population living near permanent noise sources. Typically, these limits are much lower than the occupational noise exposure levels because the intent is not to protect against hearing loss but to preserve a quality community environment.

A typical example of a community noise ordinance is illustrated in Table 22-5. The maximum allowable noise level is often a function of how the source and receiver property is zoned (not necessarily how the property is being used). Most ordinances also have allowances for somewhat higher noise levels for short periods of time as well as a requirement for lower noise levels (as much as 10 dB lower) on residential property during sleeping hours. The ordinance should also address the situation where existing ambient noise levels already exceed the ordinance requirements.

In critical situations, it may be necessary to monitor existing noise levels in the surrounding community prior to completing the design of the pumping station. This information on background noise can be used to assess the acoustical impact of the facility and make the appropriate design decisions to minimize the impact. The preferred descriptor of environmental noise levels is the day–night sound level, L_{dn}. The L_{dn} is a time-weighted energy average (over a 24-h period) of the continuously varying A-weighted sound pressure level at a point. The L_{dn} is usually determined by direct measurement with a special instrument called a "community noise monitor," which is designed for computing noise statistics at permanent or semipermanent outdoor installations. In special circumstances where the noise level does not vary with time over the 24-h day, the L_{dn} is approximately equal to the continuous A-weighted sound pressure level plus 6 dBA.

The way in which noise is perceived by average residential communities is shown in Figure 22-4. For reference, typical L_{dn} values for various areas are given in Table 22-6.

In addition to meeting the legal requirements of any state or local noise ordinances, it may be necessary to provide additional noise control measures to prevent any adverse community noise impact. Such measures would particularly apply to communities with nonexistent or poorly written ordinances, or where the ordinance requirements were inadequate

Table 22-5. Commonly Used Maximum Permissible Community Noise Levels (dBA)

	Receiving property		
Source property	Residential	Commercial	Industrial
Residential	55	57	60
Commercial	57	60	65
Industrial	60	65	70

LIVE GRAPH
Click here to view

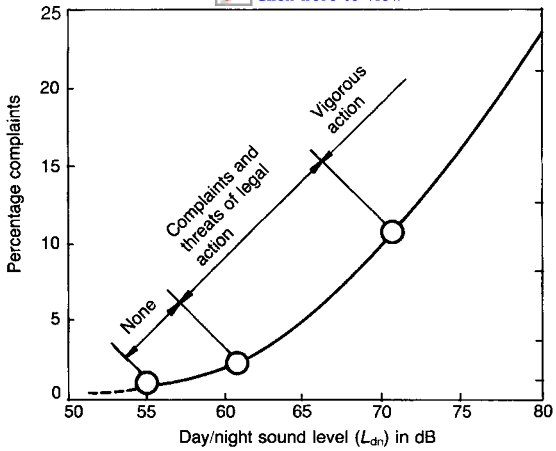


Figure 22-4. Average community reaction to noise.

to meet the needs of the community. To avoid an adverse community reaction, design pumping stations to meet the following four criteria:

- Do not increase the existing L_{dn} more than 3 dBA.
- Do not increase the existing L_{dn} more than 1 dBA if the existing L_{dn} is more than 60 dBA.
- Do not increase average hourly nighttime maximum noise levels more than 5 dBA in residential areas.
- Do not increase average hourly nighttime maximum noise levels more than 2 dBA in residential areas if the existing hourly nighttime maximum levels exceed 60 dBA.

The reason for restricting the hourly maximum noise level on residential property during the nighttime hours (usually defined as 10:00 P.M. to 7:00 A.M.) is to minimize sleep disturbance. Noise-induced sleep interference is more dependent on maximum noise levels than on average noise levels [21]. Maximum nighttime noise levels are quite volatile, that is, they usually vary by as much as 10 dB or more from hour to hour as well as on a daily basis. Consequently, it is best to base a design on an average noise level taken over several nights. These four basic criteria can usually ensure little or no acoustical impact on community noise levels.

Table 22-6. Typical Day–Night Sound Levels at Various Locations

Location	L_{dn} (dBA)
Near major international airport	75–85
Major city in downtown area	70–80
Noisy urban residential	60–70
Suburban residential	50–60
Quiet rural	45–55

22-7. Equipment Vibration

The vibration limits recommended in Table 22-2 for major items of equipment in pumping stations apply to equipment mounted directly to a rigid foundation. Vibration levels are usually higher if the equipment is mounted on a flexible structure because both the support and the machine contribute to the total vibration level. All structures are flexible to some degree, but a rigid structure is defined in the ANSI/HI standards as one that has a fundamental natural frequency higher than 125% of the highest rotating speed of the machine.

When vibrating equipment is rigidly mounted to a massive structure (such as a building floor slab) the majority of the vibrational energy generated by the machine is transmitted into the structure. Much of this energy is absorbed by the building (which causes vibration of the building's structural elements), and some is dissipated into the earth via the foundation. If the building is structurally capable of handling this vibrational energy, and if the resulting noise radiated by the vibrating structure is within acceptable limits, the rigid mounting of equipment is preferable for most pumps because, unless the pump or motor excites a building resonance, it puts less stress on the machine.

If the rotational speed of the equipment happens to match one of the resonance frequencies of the supporting structure, the structure may vibrate at excessive levels. Obviously, if the structure supporting the equipment vibrates at an excessive level, this vibration also contributes to the vibration observed at the machine. Often, the equipment is thought to be out of balance when this occurs; in fact, it may be within acceptable tolerances if removed and tested on a rigid foundation. There are three potential solutions to the problem of excessive structure vibration caused by rigidly mounted equipment:

- Avoid machine speeds that excite structural resonances.
- Modify the structure to increase the resonance frequencies so that they are above the highest machine frequency.
- Isolate the equipment from the structure with vibration isolators.

Avoiding the resonance condition altogether is the preferred technique, but when this is not feasible consider vibration isolation of the equipment.

Vibration isolators reduce the amount of vibrational energy transmitted to the structure supporting the equipment and, hence, can reduce the vibration levels of the structure. Vibration isolators do not reduce the amount of vibrational energy generated by the machine, and, in fact, the vibration level of a machine mounted on them is usually greater than that of a rigidly mounted machine, particularly at low frequencies (i.e., in the region of the machine's shaft speed). The amount of additional vibration is primarily a function of the total mass of the machine. Equipment that is very massive (such as a large diesel generator set) usually has vibration levels that are almost independent of mounting, while lighter equipment (such as an air compressor) is subject to a significant increase in vibration level when moved from a rigid to an isolated mounting.

Because vibration isolation of equipment is an added cost, it should be considered only when there is a need for reduced noise or vibration. In general, vibration isolation of equipment does not reduce the airborne noise radiated by the equipment. Although vibration isolation can significantly reduce structure-borne noise, its effects are not noticeable unless the airborne noise is also controlled. In most centrifugal pumps, the vibrational energy is primarily low frequency, and structure-borne noise is not usually a problem. But structure-borne noise may be a problem for reciprocating pumps and rotary pumps if they have fundamental pump frequencies greater than 100 Hz. Therefore, most pumps do not require vibration isolation in well-designed pumping stations.

Equipment that should be vibration-isolated in most pumping stations includes generator sets, fans, and air compressors. This equipment usually contains sufficient low- and high-frequency energy to cause excessive structure-borne noise in other areas of the building. Structural vibrations from this equipment may also be a problem for computer equipment and other sensitive electronic instruments.

22-8. Vibration Isolation Theory

Equipment vibration in its simplest form can be modeled as a small mass, m, rotating about the center of gravity of a rigid body with mass M at a distance, e_o, from the center of gravity of mass M at a fixed rotational speed. The rotating mass produces an oscillating force, F_o, whose magnitude is

$$F_o = m e_o \omega^2 \tag{22-6}$$

If the equipment is rigidly attached to the supporting structure, all of this force is transmitted into the structure, which causes it to vibrate at the frequency of rotation of the eccentric mass. The transmitted force (and resulting vibration) can be reduced by reducing (1) the eccentric mass, (2) the eccentricity, e_o, or (3) the rotational speed. Unfortunately, there are practical limits to equipment balancing (to reduce e_o), and equipment speed cannot always be reduced. The alternative method of reducing vibration transmission into the structure is by isolating the equipment with a resilient element such as a spring or a compliant pad. This isolating element, if properly designed and installed, can reduce the force transmitted into the structure and the resulting vibration dramatically.

The natural frequency of a simple, single-degree-of-freedom (one mass) isolation system mounted to a rigid structure (in the absence of damping) can be expressed as

$$f_o = \frac{1}{2\pi}\sqrt{k/M} = \frac{1}{2\pi}\sqrt{kg/W} = \frac{1}{2\pi}\sqrt{g/d} \tag{22-7}$$

where f_o is the natural frequency of vibration in hertz, k is the spring constant of the isolator in Newtons per meter (pounds force per foot), M is the total mass supported by the isolator in kilograms (pounds mass), W is the total weight supported by the isolator in Newtons (pounds force), g is the acceleration of gravity [$9.81\ \mathrm{m/s^2}$ ($32.2\ \mathrm{ft/s^2}$)], and d is the static deflection of the isolator due to the weight of the equipment in meters (feet). At frequencies much less than f_o the isolators provide no isolation and the system acts as though the isolators were not present. Within ± 50 of the resonance frequency, the isolation system reinforces and amplifies the vibration of the equipment as well as the vibration of the supporting structure. Only at frequencies much greater than f_o does the isolation system reduce the vibrational energy injected into the structure.

The efficiency of a simple, single-degree-of-freedom vibration isolation system is defined as

$$\varepsilon = 100\,[1 - (F_2/F_1)] = 100\,[1 - (x_2/x_1)] \tag{22-8}$$

where ε is the efficiency (a percentage), F is the vibrational force, x is the displacement, the subscript 2 means transmitted to the structure with isolators, and the subscript 1 means transmitted to the structure with rigid connections.

In general, the efficiency is a function of frequency, but usually the efficiency at the driving frequency or the resonance frequency of the supporting structure is

the main concern. The ratio F_2/F_1 is often called the transmissibility, T, of the system. At the lowest driving frequency, which is the rotational speed of the machine, the transmissibility is expressed as

$$T = \frac{F_2}{F_1} = \sqrt{\frac{1 + (2\delta_r f_d/f_o)^2}{(1 - f_d^2/f_o^2)^2 + (2\delta_r f_d/f_o)^2}} \qquad (22\text{-}9)$$

where f_d is the driving frequency, f_o is the natural frequency of the isolation system, and δ_r is the damping ratio, δ/δ_c. Steel spring vibration isolators typically have damping ratios between 0.01 and 0.03 (1 to 3%). The transmissibility can be evaluated at any frequency, f, by substituting f for f_d in Equation 22-9. The frequency and damping ratio dependence of the transmissibility of the single-degree-of-freedom system is illustrated in Figure 22-5.

In addition to reducing the vibration levels of structures that support equipment, it is also important to reduce the vibration levels of the equipment itself. The general expression for the displacement amplitude of a rigid mass mounted on vibration isolators with a spring stiffness constant, k, is (for $f = f_d$)

$$x = \frac{F_o/k}{\sqrt{[1 - (f_d/f_o)^2]^2 + [2\delta_r(f_d/f_o)]^2}} \qquad (22\text{-}10)$$

which is presented graphically in Figure 22-6 as a function of frequency and damping ratio. At low frequencies ($f \ll f_o$) the displacement amplitude asymptotically approaches the value F_o/k. At high frequencies ($f \gg f_o$) the displacement amplitude approaches the value $F_o/(M\omega^2)$. In the resonance region the displacement amplitude is controlled entirely by the damping factor.

Damping is the process by which vibrational energy is converted into heat. The damping ratio, δ_R, is the ratio of the damping factor, δ, and the critical damping factor, δ_C. The damping factor, δ, is a physical property of the isolator that is relatively constant (there is a slight temperature dependence) with frequency. Materials with high damping factors do not readily vibrate because the energy is rapidly converted into heat and it is not allowed to build up to high levels. Critical damping is the amount of damping required to prevent a displaced mechanical system from oscillating during its return to a rest position and is a function of the system parameters. For viscous damping, δ_C equals two times the mass times the natural frequency in radians per second. Consequently, there are no fixed values of δ_R for a given

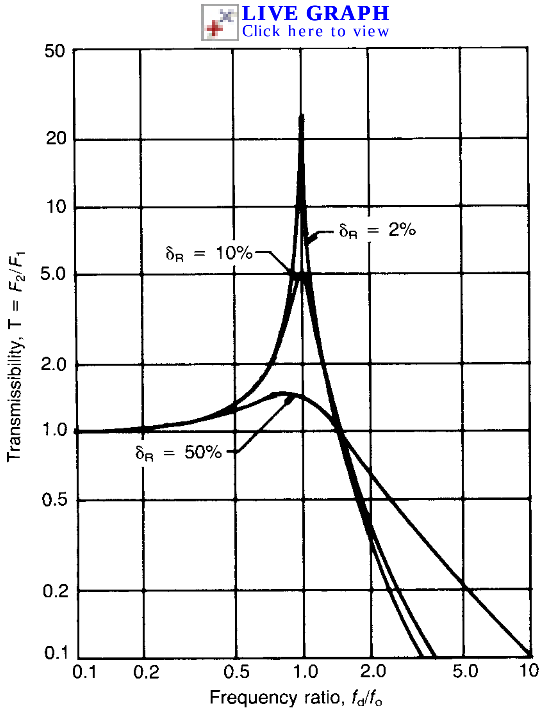


Figure 22-5. Transmissibility of a simple vibration isolation system.

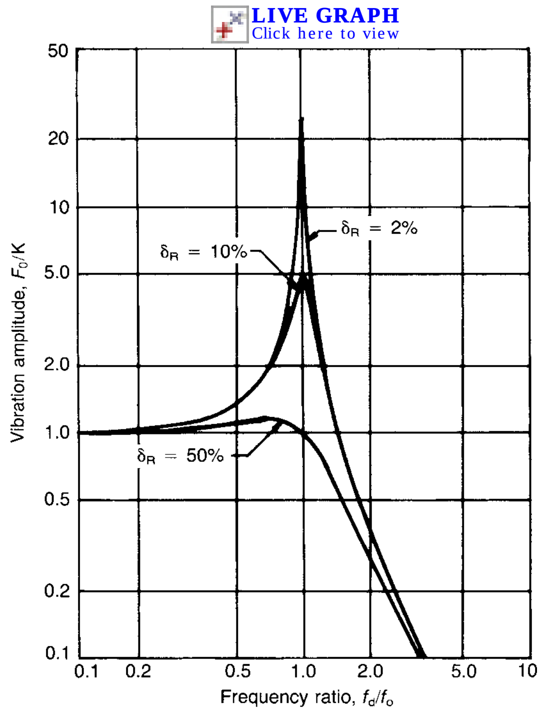


Figure 22-6. Amplitude response of a simple vibration isolation system.

material. Most spring-mounted vibration isolator systems, however, have damping ratios less than 5%. Systems with cork or rubber isolators may have δ_R values in the 10 to 20% range.

Although Figures 22-5 and 22-6 are similar in appearance, a completely separate but related phenomenon is described in each. Note that both curves show comparable response at low-frequency ratios. At high-frequency ratios (e.g., 2 or more), however, an increase in the damping ratio slightly reduces the vibration amplitude (Figure 22-6) of the equipment but increases the transmissibility (Figure 22-5) and, hence, increases the displacement amplitude of the building. Thus, depending on whether the equipment or the building is of greater concern, the addition of damping may be detrimental except at f_d/f_o ratios between 0.4 and 1.5.

For high-efficiency vibration isolation, the lowest driving frequency should be 5 to 10 times the natural frequency of the isolation system, and the damping factor, δ_r, should be low. The relationships found in Equations 22-7 through 22-9 are summarized in Figure 22-7. The required isolator static deflection needed to achieve a particular isolation efficiency at a given rotational speed is shown for a damping ratio of 0.02. Be cautious because the assumption in the preceding analysis is a linear single-degree-of-freedom system (i.e., a rigid body mounted to a resilient element obeying Hooke's law with viscous damping) supported from an infinitely rigid structure. In reality, none of these assumptions is strictly true, but they

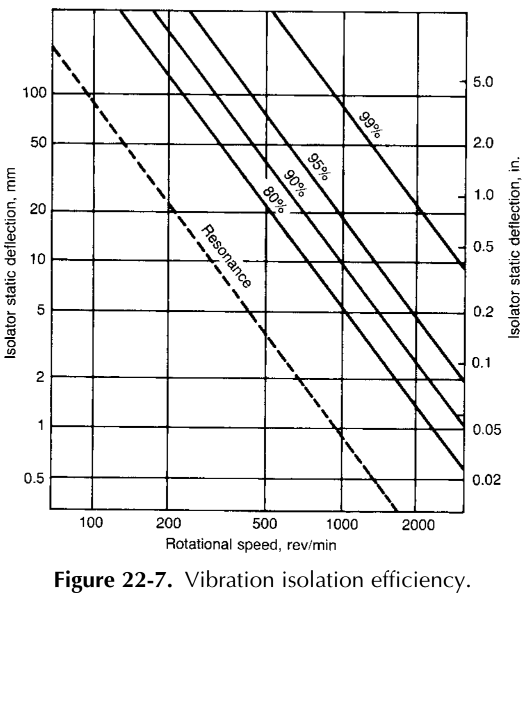


Figure 22-7. Vibration isolation efficiency.

are approximately valid for frequencies less than $10f_o$ when the natural frequency of the supporting structure is much greater than $10f_o$. At higher frequencies, other factors (such as wave effects in which energy is transmitted directly into the structure via the isolation element) take over to make the isolation efficiency much less than would be predicted with the above equations. In practice, the vibration isolation efficiency very rarely exceeds 95% at any frequency.

Warning: Do *not* be confused with the isolation efficiency figures, which are a function of the linear ratio of the applied to the transmitted dynamic force (see Equation 22-8). Human perception of noise and vibration levels follows a logarithmic response, and the perceived reduction is always less than that derived directly from the isolation system efficiency.

22-9. Vibration Isolators

There are three major types of vibration isolators. The least expensive is the pad type, which is usually made of rubber, cork, fiberglass, or a combination thereof. These isolators generally provide static deflections of less than 5 mm (0.2 in.) and, hence, are effective (efficiencies greater than 90%) only for equipment with driving frequencies greater than 25 Hz (1500 rev/min).

The most common vibration isolator is the coil steel spring. Springs are manufactured in a variety of designs that provide static deflections from 10 to 100 mm (0.4 to 4.0 in.). Springs are suitable for isolating all types of equipment with driving frequencies greater than 300 rev/min. A typical open spring isolator (so called because the spring is not enclosed in a metal housing) is illustrated in Figure 22-8. Metal housings are often used to provide lateral restraint to resist forces due to earthquakes. The threaded rod in the top plate has height adjustment nuts for leveling the equipment after mounting. The neoprene rubber pad under the base plate reduces the transmission of high-frequency energy (via wave resonances in the steel coil) into the structure. Anchor bolts in the base plate prevent equipment creep, but they must be isolated from the base plate with a neoprene washer and grommet to prevent high-frequency energy transmission into the structure via the anchor bolt. For stability reasons, the coil diameter of open springs should be at least 0.8 times the operating height of the spring.

For extremely sensitive installations (which are not anticipated in pumping stations) or for very low-frequency equipment (less than 300 rev/min), air springs may be necessary. Air springs have an enclosed volume of air to support the equipment at

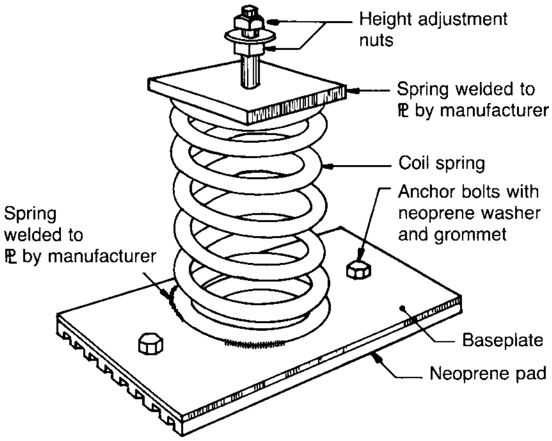


Figure 22-8. Open spring vibration isolator.

each support point. Air springs do not exhibit a linear load deflection curve (Hooke's law), and as a result, the simple theory presented above does not apply. Air springs do, in general, provide superior vibration isolation, partly because of the resulting low natural frequency and also because of the absence of high-frequency wave resonance effects. Air springs should be used only with the guidance of a qualified vibration consultant.

The recommended method of isolating a centrifugal pump from its supporting structure is illustrated in Figure 22-9. The machine is rigidly mounted to an inertia base, a concrete-filled frame designed to add mass to the equipment. The frame is then supported 50 mm (2 in.) above the housekeeping pad with steel springs. Seismic restraints should be provided at each support point with cushioned bumpers to limit motion to no more than 8 mm (0.3 in.) in any direction. Discharge piping and electrical power should be connected with flexible fittings.

The primary purpose of the inertia base in Figure 22-9 is to reduce the amplitude of the vibration of the pump, not to reduce the vibration level of the supporting structure. Adding mass to equipment that is already supported by vibration isolators may improve the theoretical efficiency slightly because it lowers f_o. However, if enough mass is added to make a significant improvement in the isolation efficiency (by increasing the ratio f_d/f_o), the springs would probably overload, because springs typically can provide a safety factor of only 20 to 50% of load capacity before "short circuiting." The inertia base does not substantially decrease the vibratory force transmitted into the structure because it does not change the transmissibility. Furthermore, the inertia base does not reduce the driving force, because from Newton's first law ($F = ma$), the decrease in acceleration, velocity, or displacement is only due to a corresponding increase in mass. Thus, a 20% increase in mass results in a 20% decrease in equipment displacement (or velocity or acceleration) amplitude with no change in transmissibility or transmitted force into the structure. The efficiency of the isolation is controlled entirely by the selection of the springs (the static deflection), as shown in Figure 22-7.

In pumping stations, reciprocating air compressors and large, high-pressure centrifugal fans and pumps are usually the only machines that require an inertia base. The amount of additional mass required varies from 1.5 to 5 times the equipment weight and depends on the level of the vibration forces in the machine and the total weight of the machine. Heavy equipment (such as diesel generators) has sufficient

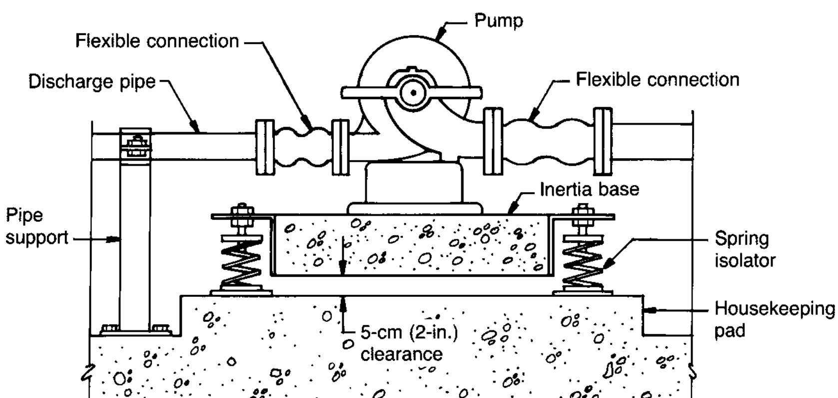


Figure 22-9. Vibration isolation of a centrifugal pump.

mass and rigidity to make inertia bases unnecessary. The only penalty for overdesigning the mass of the inertia base is the increased cost of the base and isolators.

If it is necessary to isolate a pump from the structure, it is absolutely mandatory that the pump, drive shaft, and motor (or engine) all be mounted on a common, rigid foundation (preferably a concrete inertia base) to avoid vibration-induced misalignment between these components. This isolation requirement automatically eliminates the possibility of vibration isolation for some pumps (such as vertical turbines) with motors that must be supported by a separate structural element. So as not to degrade the overall isolation provided by the isolation mounts, piping and electrical connections must also be flexible. It is difficult to specify exactly how much relative motion must be allowed, but about 5 to 10 times the displacement amplitude of the pump is recommended. This displacement tolerance can usually be achieved by using flexible rubber hose (with nylon cord reinforcement) connectors.

Example 22-1
Vibration Isolation of a Centrifugal Pump

Problem: A 150-kW (200-hp) close-coupled centrifugal water pump is driven by an induction motor at 900 rev/min. The pump/motor assembly is rigidly mounted to a concrete pad 20 cm (8 in.) thick. The supporting floor vibrates at a peak-to-peak displacement of 1.0 mm (0.04 in.) at 15 Hz while the pump is running. During the test, the unit was operated without fluid in the connected piping, and it was found that the presence of the fluid in the system had little effect on the structural vibrations of the floor.

Design an isolation system for the pump-motor assembly that reduces the structural floor vibrations to meet the safe limit for the structure with a safety factor of 2 and provides reasonable equipment vibration levels. The total mass of the pump–motor assembly is 800 kg (1760 lb_m), and the manufacturer states that the out-of-balance force should not exceed that of an eccentric mass of 1 kg (2.204 lb_m) with an eccentricity of 25 mm (0.984 in.).

Solution: The recommended safe limit for structures is 25 mm/s (1 in./s) peak-to-peak. Compute the required efficiency of the isolation system. Begin with Equations 22-1 and 22–8.

SI Units

$$d = \frac{25}{2\pi 15} = 0.265\,\text{mm (peak to peak)}$$

$$d_{\text{desired}} = \frac{0.265}{2} = 0.132\,\text{mm}$$

$$\varepsilon = [1 - (0.32/1.0)] \times 100 = 87\%$$

U.S. Customary Units

$$d = \frac{1}{2\pi 15} = 0.0106\,\text{in. (peak to peak)}$$

$$d_{\text{desired}} = \frac{0.0106}{2} = 0.0053\,\text{in.}$$

$$\varepsilon = [1 - (0.0053/0.04)] \times 100 = 87\%$$

From Figure 22-7, the required isolator static deflection is about 12 mm (0.47 in.). This deflection cannot be achieved with most pad-type isolators but is easily obtained with conventional steel spring isolators. By definition, the spring constant equals force divided by deflection so the spring constant ($k = F/d$) for each of four springs must be about

$$k = \left(\frac{800}{4}\right)\text{kg} \times 9.81\,\frac{\text{m/s}^2}{0.012\,\text{m}} = 163{,}500\,\text{N/m} \qquad k = \frac{(1760/4)}{0.47} = 936\,\text{lb/in.}$$

From Equation 22-6 ($F_o = me_o\omega^2$), the maximum out-of-balance force for a mass of 1 kg at 25 mm (2.204 lb_m at 0.984 in.) is

$$F_o = (1)(0.025)(2\pi 15)^2 = 222.1\,\text{N} \qquad F_o = \frac{2.204}{32.17}\left(\frac{0.984}{12}\right)(2\pi 15)^2 = 49.9\,\text{lb}$$

From Equation 22–7, the natural frequency of the isolation system is $f_o = (1/2\pi)\sqrt{k/M}$, where M is the pump mass supported by each spring and k is the stiffness of each spring:

SI Units

$$f_o = \frac{1}{2\pi}\sqrt{\frac{163{,}500}{(800/4)}} = 4.55\,\text{Hz}$$

U.S. Customary Units

$$M = W/g = \frac{(1760/4)}{32.17 \times 12} = 1.140\,\text{lb} \cdot \text{s}^2/\text{in.}$$

$$f_o = \frac{1}{2\pi}\sqrt{\frac{936}{1.140}} = 4.56\,\text{Hz}$$

The ratio of driving frequency to natural frequency is $f_d/f_o = 15/4.56 = 3.29$. From Equation 22-10, $x = F_o/k[(1 - f_d^2/f_o^2)^2 + (2\delta_r f_d^2/f_o^2)^2]^{-1/2}$ and assuming a 2% damping ratio, the maximum displacement amplitude of the pump is

$$x = \frac{222.1}{163{,}500}[(1 - 3.29^2)^2 + (2 \times 0.02 \times 3.29)^2]^{-1/2}$$
$$= 1.38 \times 10^{-4}\text{m}$$
$$= 0.138\,\text{mm peak to zero}$$

$$x = \frac{49.9}{936}\left[\left(1 - 3.29^2\right)^2 + (2 \times 0.02 \times 3.29^2)\right]^{-1/2}$$
$$= (0.053)/92.69 + 0.017$$
$$= 0.0054\,\text{in. peak to zero}$$

From Equation 22-1 ($x = v/2\pi f$),

$$v = 2\pi f x = 2\pi(15)(0.138) = 13.0\,\text{mm/s (peak)}$$

$$v = 2\pi(15)(0.0054) = 0.51\,\text{in./s (peak)}$$

The computed value is almost double that allowable for clear liquid pumps given in Table 22-3. Hence, the vibration level of the machine must be reduced, so an inertia base must be added. If the inertia base has an added mass of 1000 kg (2204 lb_m) and if the spring constant is nearly doubled to 300,000 N/m (1713 lb/in.), the new static deflection of each of the four isolators will be

$$d = \frac{(1000 + 800)}{4} \times \frac{9.81}{300{,}000}$$
$$= 0.0147\ \text{m} = 14.7\ \text{mm}$$

$$d = \frac{(2204 + 1760)}{4 \times 1713} = 0.579\,\text{in.}$$

which still meets the 12-mm (0.47-in.) minimum. From Equation 22-7, $f_o = (1/2\pi)\sqrt{k/M}$, the new frequency would be

$$f_o = \frac{1}{2\pi}\sqrt{\frac{300{,}000}{1800/4}} = 4.1\,\text{Hz}$$

$$f_o = \frac{1}{2\pi}\sqrt{\frac{1713 \times 386}{(2204 + 1760)/4}} = 4.1\,\text{Hz}$$

The new dynamic displacement is calculated from Equation 22-10. Note that $f_d/f_o = (900/60)/4.1 = 3.659$.

$$x = \frac{222.1}{300{,}000}\left[\left(1 - 3.659^2\right)^2 + (2 \times 0.02 \times 3.659)^2\right]^{-1/2}$$
$$= 5.98 \times 10^{-5}\,\text{m}$$
$$= 0.060\ \text{mm peak to zero}$$

$$x = \frac{49.9}{1713}\left[(1 - 13.385)^2 + 0.04(3.659)^2\right]^{-1/2}$$
$$x = 0.00233\ \text{in. peak to zero}$$

From Equation 22-1 ($x = v/2\pi f$), the peak velocity is

$$v = 2\pi 15(0.060) = 5.63\,\text{mm/s (peak)}$$

$$v = 2\pi(15)(0.00233) = 0.220\,\text{in./s (peak)}$$

which meets the criteria listed in Table 22-3.

22-10. Piping Vibration

The piping system attached to any pump vibrates as long as the pump is moving fluid inside the piping. The amount of vibration present in the pipe depends on many factors, including (1) the size, type, and rotational speed of the pump; (2) the fluid characteristics; and (3) the size, construction, support, and layout of the piping system. The purpose of this section is to present and discuss the basic design information needed to avoid vibration problems in piping systems.

To understand the nature of pipe vibrations, it is important to realize that vibration can occur in many forms; the most common are shown schematically in Figure 22-10. In general, all three modes—compression, pure bending, and radial (as well as other, less important modes)—coexist in varying degrees in all piping systems.

In the compressional mode, the vibration displacement of the pipe wall is along the length of the pipe, and the instantaneous density of the pipe wall material oscillates slightly with time and distance along the pipe. This form of vibration does not tend to stress the pipe unduly, and, consequently, it is of less importance.

In the pure bending mode, the entire pipe displaces in a direction perpendicular to the length of the pipe, much like that of a violin string when plucked. This action can put significant stress on the pipe wall if the vibration amplitude is large enough.

In the radial mode of vibration, the pipe wall motion is perpendicular to the length of the pipe, but the pipe wall at a given location moves radially inward and outward. This mode of vibration is naturally induced by pumps that discharge the fluid into the piping system as a series of pulses that cause a sinusoidal variation of fluid pressure in the piping.

These three forms of vibration, called the "natural modes of free vibration," occur only at particular discrete frequencies, called "resonance frequencies."

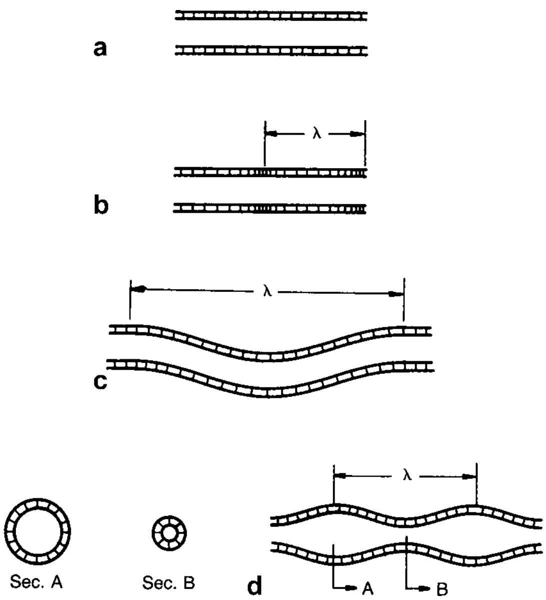


Figure 22-10. Modes of pipe vibration. (a) Pipe at rest; (b) compression; (c) bending; (d) radial.

Resonance frequencies are determined by the physical characteristics of the piping system.

Although each section of pipe has an infinite number of resonances in each of several forms of vibration, only a few of these are of concern to the designer because of the following:

- Vibrational energy from the pump is usually concentrated at the low-order pump frequencies, which are usually less than 100 Hz.
- Resonances occurring at low frequencies are the only ones that might cause large structural displacements and corresponding high stress.

In most situations, it is adequate to consider only the first two resonance orders for piping systems. The first-order resonance is called the "fundamental frequency." The second-order resonance is defined in Figure 22-11.

Bending Modes in Piping Systems

At each bending mode resonance frequency of the piping system there is a characteristic shape that the pipe assumes. The pipe oscillates back and forth resuming the same shape at periodic time intervals equal to the inverse of the resonance frequency. The exact shape of the pipe for a given mode is slightly different from the theoretical shape because the pipe also simultaneously vibrates (to a lesser degree) at other modes.

The mode shape for a given resonance is determined by the boundary conditions (i.e., how the pipe section is supported) and the order of the mode. The lowest resonance frequency for a given system is called the "fundamental" or "first-order resonance." This resonance is usually the most important because it has the greatest potential for large pipe displacements. The second-order resonance occurs typically at a frequency two to four times the fundamental and, therefore, has the potential for only about one-half to one-fourth of the displacement of the fundamental mode of vibration if equal energy input into each mode is assumed.

The boundary conditions describe how the pipe is supported (restrained from motion) at the ends of a pipe section. There are numerous ways of supporting pipe sections, but four of the more common are given in Table 22-7.

Mode shapes and corresponding natural frequencies for various combinations of the first three boundary conditions have been determined analytically [5, 22], and some are shown in Figure 22-11. The analysis of the vibration-isolated support does not seem to

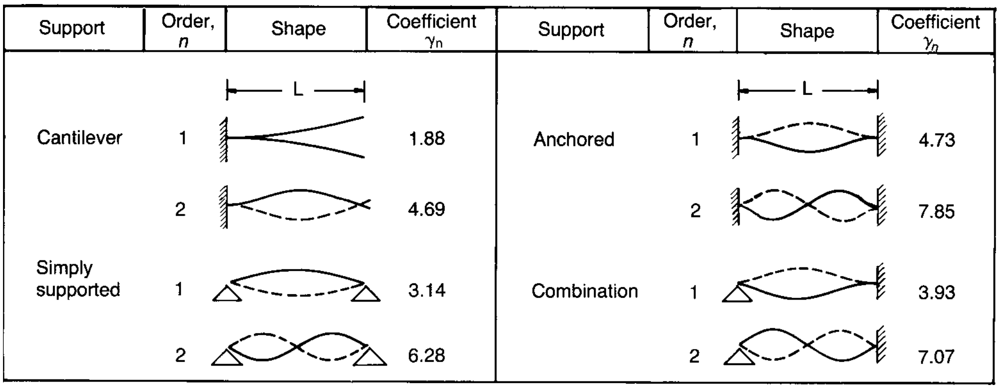


Figure 22-11. Bending mode resonances.

have been published to date, but the mode shapes and natural frequencies are expected to fall somewhere between the free and simply supported conditions.

The general equation for computing bending mode resonance frequencies for a piping system of uniform diameter and physical properties can be expressed as

$$f_n = \frac{\gamma_n^2}{2\pi L^2}\sqrt{\frac{EIg}{w}} \qquad (22\text{-}11)$$

where f_n is the resonance frequency in hertz, γ_n is the coefficient of bending resonance, L is the length of pipe between supports in meters (inches), E is the modulus of elasticity in Pascals (pounds force per square inch), I is the moment of inertia in meters to the fourth power (inches to the fourth power), g is the acceleration due to gravity [9.81 m/s^2 (386 in./s^2)], and w is the weight (including fluid) per unit length in Newtons per meter (pounds force per inch). In Equation 22-11, it is assumed that the pipe is straight and has a constant diameter. Values for γ_n are given in Figure 22-11 for the fundamental and second-order resonance for various boundary conditions. The moment of inertia for pipe is

$$I = \frac{\pi}{64}\left[(OD)^4 - (ID)^4\right] \qquad (22\text{-}12)$$

where OD and ID are the outside and inside diameters of the pipe. Weights of pipe are readily available from the pipe manufacturers (see also Tables B-1 to B-4).

Another condition of interest to designers is a straight section of uniform pipe with an added center mass, which may be a valve or some other device supported by the pipe. Blevins [5] has published an expression for the approximate natural frequency for simply supported boundaries:

$$f_1 = \frac{2}{\pi}\sqrt{\frac{3EI}{L^3(M_o + 0.49M_p)}} \qquad (22\text{-}13)$$

where M_o is the central concentrated mass in kilograms (pounds mass) and M_p is the total mass of the pipe and fluid between the support points. The other terms are as defined previously.

An approximate solution for clamped boundaries and noncentered masses is also given by Blevins, but it is too lengthy to reproduce here. Techniques for computing the natural frequencies of piping systems containing elbows and other complexities are addressed by the Kellogg Company [22].

Table 22-7. Boundary Conditions for Pipe Bending Modes

Support	Boundary conditions
Free	Pipe is unrestrained to any motion (moment and shear loads are zero)
Simply supported	Pipe is supported to prevent displacement but allow rotation
Clamped	Pipe is clamped to prevent both displacement and rotation
Vibration isolated	Pipe is resting on a compliant support (e.g., a spring) that allows both displacement and rotation

Forced Vibrations

The source of vibrational energy in a typical piping system is usually the pump (sometimes augmented by

an engine drive). The energy can be transmitted to the pipe either through the coupling between the two or through the fluid in the pipe. The coupling between the pump and the pipe wall transmits vibrations from the pump housing, which is usually dominated by the rotational frequency of the pump shaft and the pump frequency (the shaft frequency times the number of blades, teeth, or pistons). The vibration of a pipe due to a continuous energy source (e.g., a pump) is called "forced vibration." The term "forced vibration" is used because the system vibrates at the driving frequency of the source in addition to its own natural frequencies.

The amplitude of the forced vibration of a pipe depends on the amount of energy and frequencies introduced by the pump as well as the physical characteristics of the piping system. If the forced vibration occurs at one of the natural frequencies of the pipe, system vibration can be amplified to extremely high levels. This condition is called "resonance." The amplification of vibration near resonance follows the same basic laws discussed in Section 22-7.

Most metallic pipes possess values of δ_r that are much less than 0.05 and even as low as 0.02. Consequently, during resonance ($f_d = f_n$) the vibration amplitudes, depending on the damping factor, can be as much as 10 to 25 times that of the nonresonant condition. Most vibration-induced piping failures are due to resonance. Resonance can best be avoided by designing the piping support system so that the driving frequency is either less than $f_1/2$ or greater than $3f_1/2$.

The problems introduced by variable-speed pumps are demonstrated in Example 22-2. The variable-speed pump can also excite structural resonances of the building, a problem discussed in detail in Section 22-12.

The spacings between pipe supports should be approximately equal, but not necessarily exactly equal. Unequal support spacings simply add additional natural frequencies to the system, which reduces the frequency spacing between the driving frequency and the natural frequencies of the piping system. However, there is some merit to varying the pipe support spacing slightly (e.g., 10% or less) in an irregular manner, particularly if the system must operate near resonance. Adjacent sections of piping are more likely to reinforce vibration at resonance if the support spacings are identical. The damping factor is so low for most pipes that spacing variations as small as 2% should be adequate to minimize this problem.

If it is necessary to minimize bending mode vibrations in piping systems, observe the following design guidelines:

- Anchor the pipe to the structure at interval spacings that avoid natural frequencies in close proximity to forcing frequencies.
- Eliminate sharp elbows in the piping system—especially in the vicinity of pumps—since the fluid acceleration due to the change in direction at elbows helps transfer the energy of fluid pressure pulsations into bending mode vibrational energy in the pipe.
- Install flexible connections between the pump and piping at the inlet and discharge.

Example 22-2
Pipe Vibration with a Variable-Speed Centrifugal Pump

Problem: An end-suction centrifugal pump discharges clean water into a Class 53 ductile iron pipe 600 mm (24 in.) in diameter. The pump has a twin vane impeller that rotates at speeds between 560 and 800 rev/min. Design a piping support system free of bending mode resonance problems.

Solution: The rotating frequency of the pump shaft and impeller varies from $f_{min} = 560/60 = 9.3$ Hz to $f_{max} = 800/60 = 13.3$ Hz, and the fundamental blade passage frequency varies from $f_{min} = 2(9.3) = 18.6$ Hz to $f_{max} = 2(13.3) = 26.6$ Hz. The second and third harmonics (two and three times the fundamental, respectively) of the blade passage frequency vary as follows: $f_2 = 37 - 53$ Hz and $f_3 = 56 - 80$ Hz.

Support spacing requirements. Select the supporting spacing to avoid the resonance condition (within 50% as recommended previously). Because a variable-speed pump is used, virtually all frequencies above 18.6 Hz are possible driving frequencies or harmonics thereof. There is an available window between 13.3 and 18.6 Hz, but the window is not wide enough to meet the 50% clearance requirement. Because there is no practical method of completely

avoiding the pipe resonances, select a support spacing that puts the fundamental pipe resonance above 100 Hz. The resonance problem would thus be moved outside of the frequency region where vibration problems are most common, thus reducing the amplitude of vibration because displacement is inversely proportional to the frequency.

Resonance frequencies. Compute these by using Equation 22–11, which requires the determination of I (from Equation 22–12 and Tables B-1 and B-2), E (from Tables 4-1 or A-10), w (from Tables B-l and B-2), γ_n (from Figure 22–11), and L.

SI Units

$$I = \frac{\pi}{64}\left[(0.655)^4 - (0.629)^4\right] = 0.00120\,\text{m}^4$$

$$E = 166 \times 10^6\,\text{kPa} = 1.655 \times 10^{11}\,\text{N/m}^2$$

SI Units

$w = w_{\text{water}} + w_{\text{pipe}}$

From Tables B-1 and A-8, for 1 meter of length:

$$w_{\text{water}} = (0.308\,\text{m}^2)(998.2\,\text{kg/m}^3)(9.81\text{m/s}^2) = 3020\ N/m$$

$$w_{\text{pipe}} = (170\,\text{kg/m})(9.81\,\text{m/s}^2) = 1670\ N/m$$

$$w_{\text{liner}} = (2.38 \times 10^{-3}\,\text{m})\pi(0.627\,\text{m}) \times (2400\,\text{kg/m}^3)(9.81\,\text{m/s}^2) = 110\ N/m$$

w per meter $=$ 4800 N

$\gamma_n = 3.14$

Try $L = 3\,\text{m}$

$$f_1 = \frac{(3.14)^2}{2\pi(3)^2}\sqrt{\frac{(1.655 \times 10^{11})(0.00120)(9.81)}{4800}} = 111\,\text{Hz}$$

$$f_2 = \left(\frac{\gamma_2}{\gamma_1}\right)^2 f_1 = \left(\frac{6.28}{3.14}\right)^2 \times 111 = 444\,\text{Hz}$$

U.S. Customary Units

$$I = \frac{\pi}{64}\left[(25.8)^4 - (24.9)^4\right] = 2880\,\text{in.}^4$$

$$E = 24 \times 10^6\,\text{lb/in.}^2$$

U.S. Customary Units

$w = w_{\text{water}} + w_{\text{pipe}}$

From Tables B-2 and A-9, for 1 in. of length:

$$w_{\text{water}} = (3.32\,\text{ft}^2)(62.3\,\text{lb/ft}^3)(1\,\text{ft})/12 = 17.2\,\text{lb/in.}$$

$$w_{\text{pipe}} = (114\,\text{lb/ft})(1\,\text{ft})/12 = 9.5\,\text{lb/in.}$$

$$w_{\text{liner}} = (3/32\,\text{in.})\pi(24.7\,\text{in.})(150\,\text{lb/ft}^3)(1/1728) = \underline{0.7\,\text{lb/in.}}$$

w per inch $=$ 27.4 lb

$\gamma_n = 3.14$

Try $L = 118.1$ in.

$$f_1 = \frac{(3.14)^2}{2\pi(118)^2}\sqrt{\frac{(24 \times 10^6)(2880)(386)}{27.4}} = 111\,\text{Hz}$$

$$f_2 = \left(\frac{\gamma_2}{\gamma_1}\right)^2 f_1 = \left(\frac{6.28}{3.14}\right)^2 \times 111 = 444\,\text{Hz}$$

If these guidelines are followed, the worst situation that could result is a pipe system resonance excited by one of the higher harmonics of the pump. The displacement amplification may reach 20 or even 50, but the displacement of the pipe would also be reduced by the increase in vibration frequency and the decrease in energy available from the pump at higher harmonics. By increasing the natural frequency of the piping system from 15 Hz (the center of the available window between 13.3 and 18.6 Hz) to 112 Hz, the displacement amplitude is reduced by a factor of $^{15}\!/_{112}$ or 0.134 if equal energy in all harmonics is assumed. If harmonics above f_3 are assumed to have less than $^{1}\!/_{10}$ the energy of the fundamental (a safe assumption for centrifugal pumps), then, even with a displacement amplification of 50 at 112 Hz (assuming the worst condition), the amplitude of the pipe vibration would be less than 0.134 times 50 divided by 10. This displacement is less than 70% of the value that would be expected at 15 Hz if a spacing of 10 m (30 ft) had been selected to fit into the 13.3- to 18.6-Hz window.

Another solution to Example 22-2 would be to switch to a constant-speed pump. If the specified pump were changed to a triple vane impeller centrifugal pump driven at a constant speed of 1200 rev/min, the driving frequencies would be 20 Hz (the shaft speed), 40 Hz (two times the shaft speed),

60 Hz (the pump frequency), 120 Hz, 180 Hz, and so on. A pipe spacing of 3 m (118 in.) would not be satisfactory because its resonance frequency (112 Hz) is close to the second harmonic of the blade passage frequency (120 Hz). With this pump, pipe resonance should ideally be about 90 Hz. Although the 90-Hz pipe resonance would provide only 30% clearance to the nearest driving frequencies, it would be preferred because the nearest driving frequencies are higher harmonics (which typically have less energy than the fundamental) and because the higher frequency would produce lower displacement amplitudes. Equation 22-11 can be rearranged to solve for L, which yields 3.33 m (10.9 ft) for 90 Hz.

Acoustic Resonances

Periodic pressure fluctuations caused by the pump can also excite acoustic resonances within the piping system. Just as there are an infinite number of bending mode resonances in a piping system, there are also an infinite number of acoustic resonances. An acoustic resonance is created when a discontinuity in the piping system (e.g., a termination, elbow, or branch) reflects a pressure wave. When a pressure wave propagating down the pipe reaches a discontinuity, a portion of the energy in the wave is reflected back toward the source while the rest of the energy in the wave continues down the pipe. If the length of the pipe and the speed of wave propagation in the fluid are such that the initial and reflected waves are in phase, the system operates at an acoustic resonance. In actuality, the pressure wave reflects back and forth at each discontinuity several times, creating what is called a "standing wave" in the pipe. The number of times a wave is reflected before it dissipates depends on the type of discontinuity (i.e., the strength of the reflection at each end) and the viscous losses in the system. Terminations or abrupt changes in area provide the strongest reflections. Reflections from small branches and 45-degree elbows are generally weak enough to be disregarded.

What occurs in the fluid when a system operates at an acoustic resonance is graphically illustrated in Figure 22-12. The instantaneous fluid pressure (with the movement in the pipe wall greatly exaggerated) is shown in a fixed section of pipe at four different times. At time $t = 0$, the pressure is unevenly distributed along the pipe in a sinusoidal manner with the peak pressure locations separated spatially by a distance, γ, the wavelength of the acoustic resonance in the fluid. At $t = f_n/4$, the pressure distribution and pipe shape are uniform. At $t = f_n/2$, the uneven distribution returns, except that the location that was at high pressure (at $t = 0$) is now a low-pressure zone, and vice versa. It is important to note that the average pressure in the pipe (averaged over a time interval of $1/f_n$ or its multiple) is the same at all points.

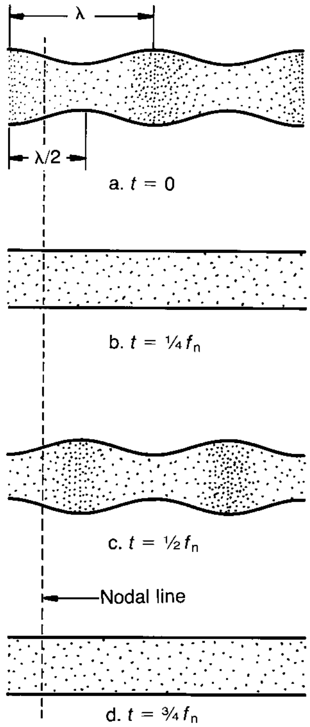


Figure 22-12. Internal pressure distribution at acoustic resonance.

There are fixed locations along the pipe where the internal pressure does not oscillate about the average pressure. These points are called "nodes," and they also are separated by a distance, $\lambda/2$, along the pipe. Note that the nodes and antinodes (which are regions of high-pressure fluctuations) of acoustic resonances are at fixed locations that are determined by the pipe system geometry. They do not move with the fluid in the pipe.

The natural acoustic frequencies of a straight, uniform length of pipe are determined by the pipe length, the boundary conditions at each end, and the compressional wave propagation speed in the fluid (speed of sound). Depending on the impedance presented by the discontinuity, the acoustic wave reflected by a discontinuity may be reflected with or without a 180-degree phase reversal. In general, a pipe opening into a large volume with an abrupt termination creates a large reflection with a 180-degree phase shift.

This termination is called an "open-end" condition. At the resonance frequency of the open-end condition, the system node is located at the termination. On the other hand, a pipe section that terminates with a blocked end cap, a tee connection, or a 90-degree elbow is said to have a "closed-end" condition, which provides no phase reversal for the reflected wave. Consequently, the dynamic pressure is greatest at the termination and the nodes occur at other positions along the pipe.

For straight runs of uniform pipe, the acoustic natural frequencies for pipes open at both ends or closed at both ends can be computed as follows:

$$f_n = \frac{nc}{2L} \tag{22-14}$$

where n is the resonance order ($n = 1, 2, 3, \ldots$), c is the wave propagation speed of fluid, and L is the length of the column of fluid between discontinuities where a reflection is anticipated.

For pipes open at one end and closed at the other end,

$$f_n = (2n - 1)\frac{c}{4L} \tag{22-15}$$

Note that L has nothing to do with the spacing of pipe supports or the spacing of couplings.

In addition to individual sections of water columns, it is also possible to encounter acoustic resonances that incorporate one or more individual water columns, nonuniform sections of pipe, and other complexities. The equations for computing these frequencies are very complex—well beyond the scope of this text (see *Design of Piping Systems* [22]).

Speed of Sound

The speed of sound in liquids is

$$c_o = \sqrt{\frac{\gamma B}{\rho}} \tag{22-16}$$

where c_o is speed of sound in a free liquid in meters per second (feet per second), γ is the ratio of specific heats (1.0 for fresh water and 1.01 for sea water), B is the isothermal bulk modulus of the fluid in pascals (pounds force per square foot), and ρ is the density of the fluid in kilograms per cubic meter (slugs per cubic foot). In general, these quantities vary somewhat with temperature and pressure in a complex manner as shown in Tables A-8 and A-9. γ is further defined as c_p/c_v, where c_p is specific heat capacity of the fluid at constant pressure and c_v is the specific heat capacity at constant volume. In SI units, an empirical expression for the speed of sound in fresh water is

$$\begin{aligned} c_o = 1403 + 5.02T - 0.055T^2 \\ + 0.0003T^3 + 0.00173P \end{aligned} \tag{22-17a}$$

where c_o is the speed of sound in meters per second, T is the temperature in degrees Celsius, and P is the gauge pressure in kilopascals.

In U.S. customary units,

$$\begin{aligned} c_o = 4248 + 13.23T - 0.072T^2 \\ + 0.00016T^3 + 0.0392P \end{aligned} \tag{22-17b}$$

where c_o is speed of sound in feet per second, T is the temperature in degrees Fahrenheit, and P is the gauge pressure in pounds force per square inch.

The speed of sound in salt water is slightly higher (by approximately 3%) than it is in fresh water due to the effects of salinity. The speed of sound can be lowered dramatically by air bubbles in the fluid. As long as the concentration of air is less than 0.01% by volume, the effect is insignificant. However, when the air concentration reaches 0.1%, the speed of sound is reduced by nearly 50% (see Streeter and Wylie [23] for more information on the effects of air bubbles).

In addition to its dependence on temperature and pressure, the speed of sound is also influenced by the pipe. The finite mass and compliance of the pipe wall decreases the speed of sound as the pipe wall thickness decreases and the pipe diameter increases. The expression for the speed of sound within a pipe is

$$c = \frac{c_o}{1 + B(ID/Ee)} \tag{22-18}$$

where c_o is given by Equation 22-17, B is the bulk modulus of the fluid, ID is internal pipe diameter, E is the modulus of elasticity of the pipe, and e is the thickness of the pipe wall. The speed of sound in fresh water within several Class 53 ductile iron pipes is shown in Figure 22-13 as a function of water temperature.

Acoustic resonances do not necessarily cause visible or perceptible vibration in the wall of the pipe, but they can cause very high dynamic pressures inside the pipe at the antinodes. It should be emphasized that the pressure rating for piping is based on internal static (steady) pressures. Safe limits for dynamic

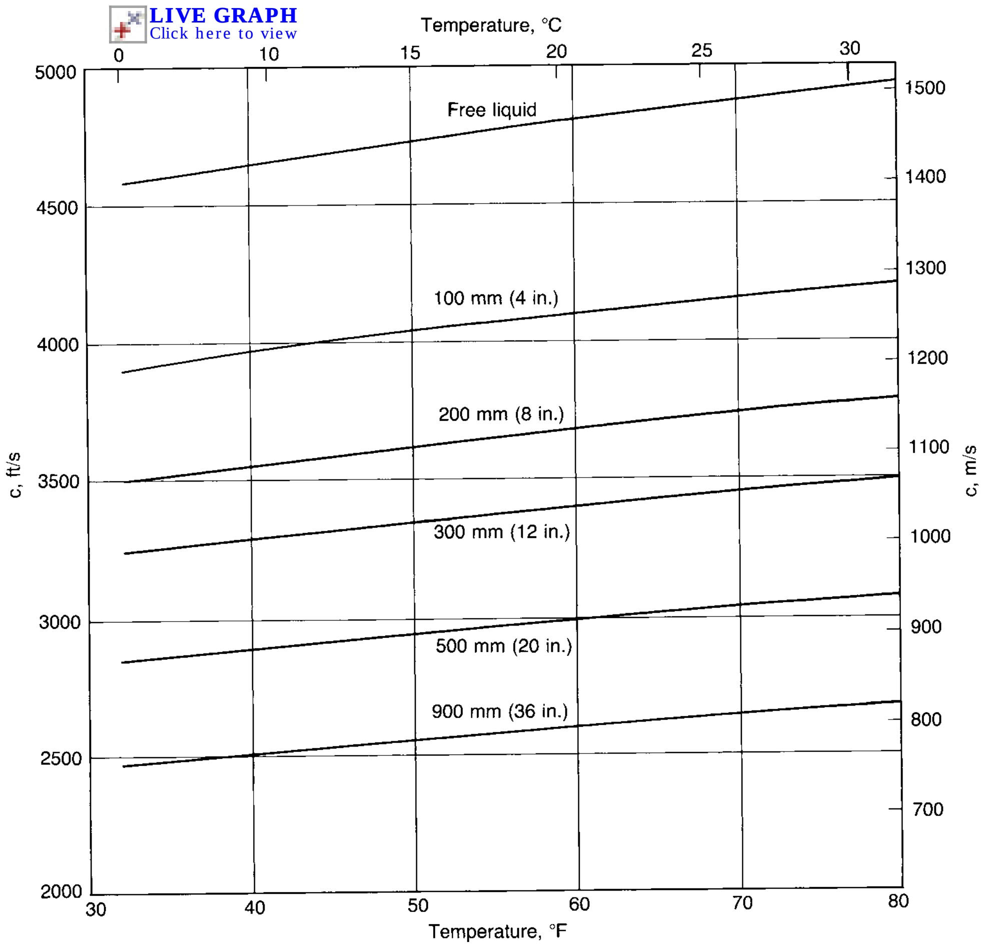


Figure 22-13. Speed of sound in fresh water at atmospheric pressure in Class 53 ductile iron pipe.

pressures are typically less than one-half of the static limits because of pipe fatigue. For most piping systems, this limits safe dynamic pressures to less than 690 kPa (100 lb/in.^{2}) peak-to-peak.

Acoustic resonance may also increase the noise level radiated by the pipe if the frequency is high enough. Depending on the ambient noise level in the plant, and if the frequency is lower than 250 Hz (as it would be for most pumping station systems), radiated noise from an acoustic resonance should not be perceptible.

Suppression Techniques

Once a potential or existing piping vibration or acoustic resonance problem is discovered, one is usually interested in finding a remedy—or at least a method of improving the situation with as little effort and expense as possible. The solution, of course, requires knowledge of the nature of the problem as well as some specifics concerning the frequencies and levels of vibration. For existing problems, this information is best obtained by direct measurement with an accelerometer and a spectrum analyzer. These measurements usually require the expertise of a qualified vibration consultant. Once the specifics of the problem are identified, a proper suppression technique can usually be applied.

Detuning

Because most vibration problems are associated with the mechanical resonance of one or more elements in the system, detuning is the simplest and most effective

method of solving a piping vibration problem. Detuning merely implies changing either the driving frequency of the pump or the natural frequency of the piping system. Prior to detuning any problem system, it is important to determine the exact driving frequency and natural frequency of the installed piping system, particularly if the anticipated solution is to make a modest change in pump speed. Without this information, it is possible to make the situation even worse if the new pump speed turns out to be closer to the piping's natural frequency than the original pump speed.

Flexible Connections

A flexible connection between the pump and the piping can be an effective means of reducing pipe vibration if the pump is vibration-isolated from the structure and if the flexible connection is truly flexible. When the pump and piping are not isolated from the structure, it is possible for this energy to get back into the piping via the common structure. As shown in Figure 22-14, the most effective flexible connectors are made from molded neoprene rubber. A series of "accordion" corrugations is even more effective. In addition to isolating the piping from the structural vibrations of the pump, this particular design also effectively reduces pulsations in the fluid by virtue of the expansion capabilities of the rubber walls. The use of neoprene rubber is limited to temperatures below 100°C (212°F) and pressures below 1000 kPa (150 lb/in.^{2}).

The conduit for the motor should likewise be flexible to prevent transmitting vibration into the structure via the electrical conduit.

Resilient Supports

Resilient supports for piping systems are used in some industries to provide for expansion and contraction of the piping system as well as to reduce structure-borne noise from the pipe. In these devices a resilient element (e.g., neoprene rubber or fiberglass) is placed between the pipe wall and the pipe clamp, which allows the pipe to flex slightly and independently of the structure. Theoretically, these clamps also add some additional damping to the bending modes of pipe vibration, but they cannot reduce pressure pulsations in the fluid. Because structure-borne noise is not usually a big problem in pumping stations, the use of these devices is probably not warranted.

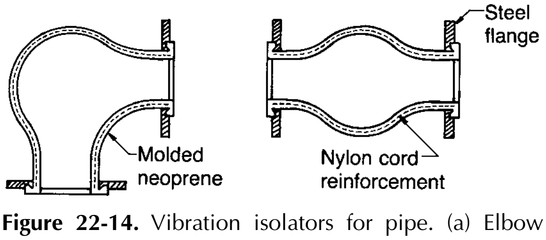


Figure 22-14. Vibration isolators for pipe. (a) Elbow coupling; (b) straight coupling.

Dynamic Absorbers

A dynamic absorber is a passive mass attached to a vibrating element or system with a resilient component (e.g., a spring). The auxiliary mass and spring are attached to the vibrating element and are designed (or "tuned") to vibrate out of phase with the primary system. This results in significant vibration, which is significantly reduced at the resonance frequency. The tuning of the auxiliary mass and spring is critical because these auxiliary elements create a two-degree-of-freedom system that may cause an increase of vibration level at other nearby frequencies. The use of a dynamic absorber is applicable only for a system operating at or near resonance and should be attempted only with the assistance of a qualified vibration consultant.

Expansion Chambers

Expansion chambers are simply enlarged sections of pipe with abrupt discontinuities at each end, as shown in Figure 22-15. The expansion chamber provides a loss of dynamic pressure (i.e., reduced pressure pulsations) primarily through the interaction of reflections within the chamber. The dynamic pressure losses occur at all frequencies except at 0 Hz and at the resonance frequencies of the expansion chamber (when the length, L, equals an integer multiple of one-half of the wavelength) where no losses occur. The expansion chamber cannot amplify the fluid pulsations in the downstream pipe.

The effectiveness of an expansion chamber can be described by the ratio of the outlet to inlet dynamic pressures:

$$P_{\text{out}}/P_{\text{in}} = \frac{1}{\{1 + [0.25(S_2/S_1 - S_1/S_2)^2]\sin^2(2\pi fL/c)\}} \tag{22-19}$$

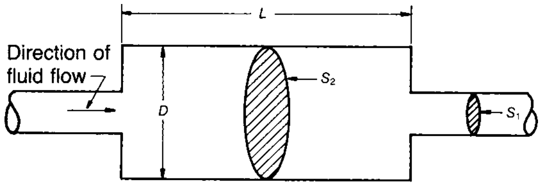


Figure 22-15. Expansion chamber.

where S_1 and S_2 represent the cross-sectional area of the inlet pipe and the expansion chamber, respectively. Note that from this expression there is no pulsation reduction whenever the argument of the $\sin^2$ function is an integer multiple of π. The maximum pulsation reduction occurs at the frequency where the argument of the $\sin^2$ function is equal to $\pi/2$, or

$$f = (2n - 1)c/4L \qquad (22\text{-}20)$$

Thus, maximum reduction occurs when the length of the expansion chamber equals one-fourth of the wavelength, three-fourths of the wavelength, five-fourths of the wavelength, and so on.

In most pumping applications, the concern is with low-frequency performance where the expansion chamber length is only a fraction of the wavelength of the pump frequency. Under these circumstances, the expansion chamber is designed to ensure that the ratio S_2/S_1 and the pump frequency are high enough to produce the necessary pulsation reduction.

Tuned Resonators

Tuned resonators are sections of pipe that are added to the system in order to remove dynamic pressures in a very narrow frequency band. The simple example of a side branch resonator (a closed-end tube attached to the main pipe) is shown in Figure 22-16. If the effective length of the side branch resonator is exactly equal to one-fourth of the wavelength of the pump frequency, it can reduce the dynamic pressure in the downstream piping dramatically—by as much as 99%. It is important to realize that this device works only at the following particular frequencies:

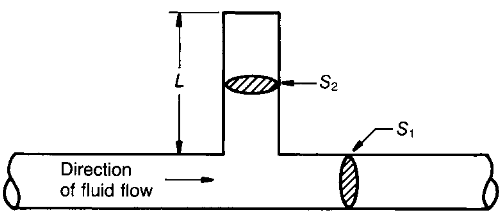


Figure 22-16. Side branch resonator.

$$f_n = (2_n - 1)c/2L \qquad (22\text{-}21)$$

where $n = 1, 2, 3, \ldots$. The length of the resonator must equal one-half of the wavelength, three-halves of the wavelength, and so on. These lengths differ from those of the expansion chamber discussed previously, because the side branch resonator is closed at one end, whereas the expansion chamber is open at both ends. The sound wave in the expansion chamber can propagate the distance L in a single pass. But the sound wave in the side branch resonator travels the distance L in one direction, is reflected and travels back again, and gives rise to the different factors ($2L$ versus $4L$) in Equations 22-20 and 22-21.

At other frequencies the resonator is transparent to the system; that is, it acts as though it were absent or "invisible." Therefore, consider this device only for single-speed pumps moving fluids with temperatures that remain constant. Side branch resonators are not economically feasible for low-speed pumps because the resonator length becomes very long.

Pulsation Dampers

A pulsation damper reduces the dynamic pressure fluctuations within a piping system, and it can also reduce the bending mode vibration levels of the piping system if the pipe vibration is induced by the pulsating fluid. Two types are commonly used. The most common pulsation damper is a fluid-filled spherical vessel mounted in line with the piping at the discharge of the pump. A sphere is commonly utilized to achieve maximum volume and maximum pressure rating with minimum weight and cost. The general design concept is the same as that of the expansion chamber discussed above. Typically, pulsation dampers reduce dynamic pressures by 75% and more.

The other type of pulsation damper is the air chamber (see the hydropneumatic tank in Chapter 7), which also functions as an accumulator. This device is essentially a pressurized expansion tank (partially filled with gas) connected to the piping near the pump discharge. The accumulator adds and accepts fluid from the piping through an orifice as the dynamic pressure fluctuates. The accumulator is not an in-line device; therefore, its effectiveness at reducing high-frequency (50 Hz and above) dynamic pressures in the downstream piping is limited. Neither the accumulator nor the pulsation damper has a sufficient volume of gas to control water hammer in most instances. Pulsation control devices should be considered for both the suction and discharge flow lines because these units protect the pump as well as the piping; usually, they also improve pump performance.

Example 22-3
Dynamic Pressure from a Plunger Pump

Problem: A reciprocating (single-acting plunger duplex) pump delivers fresh water at 3.15 L/s at 2413 kPa (50 gal/min at 350 lb/in.^{2}) working pressure from a 5°C (50°F) reservoir at 400 rev/min into a Class 53 ductile iron pipe 300 mm (12 in.) in diameter that terminates in a large holding tank. Assume that the peak dynamic pressure generated by the pump is 25% of the static head (50% peak-to-peak). Compute the acoustic resonance frequencies and determine if the system is in danger of fatigue fracture if the pipe is relatively straight, has a total length of 64 m (210 ft), and has a yield strength of 290,000 kPa (42,000 lb/in.^{2}).
Solution: The speed of sound in the pipe can be computed to be 1007 m/s (3303 ft/s) from Equations 22-17 and 22-18. If the effect of pressure is ignored (because it is so small), the speed of sound in the pipe can also be estimated from Figure 22-13. The driving frequency of the pump is $f_d = (400)(2)/60 = 13.3\,\text{Hz}$, and the harmonics of the driving frequency are 26.6 Hz, 40 Hz, etc. The pipe is open at one end, and the pump itself closes the other end. From Equation 22-15 [$f_n = (2n - 1)c/4L$], the acoustic resonances are as follows:

SI Units	**U.S. Customary Units**
$f_1 = (2 \times 1 - 1)1007/(4 \times 64) = 3.93\,\text{Hz}$	$f_1 = (2 \times 1 - 1)3303/(4 \times 210) = 3.93\,\text{Hz}$

The second- and third-order acoustic resonances are

$f_2 = (2 \times 2 - 1)1007/(4 \times 64) = 11.8\,\text{Hz}$	$f_2 = (2 \times 2 - 1)3303/(4 \times 210) = 11.8\,\text{Hz}$
$f_3 = (2 \times 3 - 1)1007/(4 \times 64) = 19.7\,\text{Hz}$	$f_3 = (2 \times 3 - 1)3303/(4 \times 210) = 19.7\,\text{Hz}$

The second-order acoustic resonance (11.8 Hz) is close (within about 10%) of the fundamental driving frequency of the pump ($f_o = 0.89 f_d$), which could excite the first-order acoustic resonance within the pipe. A pulsation damper is required in this installation to reduce the dynamic pressure, but the pulsation damper will not change the resonance match between the driving frequency of the pump and the water column in the pipe.

Effect of damping. If the pulsation damper reduces the dynamic pressures in the discharge pipe by, for example, 75% and the pressure amplification for the acoustic resonance is estimated to be about 5 (from Figure 22-6, for 2 to 10% of critical damping), the maximum dynamic pressure can be computed as working pressure times 25% times (1 – pulsation damping reduction) times the acoustic resonance amplification factor, or

$(2413)(0.25)(1 - 0.75)(5) = 754\,\text{kPa}$	$(350)(0.25)(1 - 0.75)(5) = 109\,\text{lb/in.}^2$

These pressures meet the 50% of rated value criterion, but they are higher than the 690-kPa (100-lb/in.^{2}) guideline. If the damping ratio were as low as 0.02, amplification factors could reach 20, which could bring the dynamic pressures inside the pipe to as high as 3020 kPa (440 lb/in.^{2}), and the total (working plus dynamic) pressure would be 5440 kPa (790 lb/in.^{2}). From Equation 4-2 ($s = PD/2e$), the hoop tensile stress in the pipe at peak pressure would be

$$s = \frac{5440 \times 0.335}{2 \times 0.0102} = 89{,}300\,\text{kPa} \qquad s = \frac{790 \times 13.2}{2 \times 0.40} = 13{,}000\,\text{lb/in.}^2$$

This seems safe enough, but the maximum yield strength of 290,000 kPa (42,000 lb/in.^{2}) is applicable only to static stress levels. Dynamic stress levels must be substantially lower to avoid fatigue failure.

Changing operating speed. In addition to the pulsation damper, the operating speed of the pump should be changed to avoid operation near the acoustic resonance frequencies and, thus, reduce the internal dynamic pressure. A suitable choice might be a smaller duplex pump running at 480 rev/min, which would yield driving frequencies of 16 Hz, 32 Hz, etc., and

would result in an estimated pressure amplification of less than about 2.0 for all values of the damping ratio (see Figure 22-6) for $f_d/f_o = 16/11.8 = 1.356$. The maximum dynamic pressure at the driving frequency of the pump would then be $(2420)(0.25)(1 - 0.75)(2) = 302$ kPa (44 lb/in.^{2}), which is safe by both criteria.

22-11. Vibration of Drive Shafts

Drive shaft vibration is a complicated phenomenon composed of two independent forms of vibration—translational and torsional. The approximation methods presented here are intended to provide only (1) a reasonable idea of safety margins, and (2) sufficient background for pumping station designers to judge properly the results presented by the shaft, engine, or (usually) pump manufacturer, who should be responsible for the design of the entire rotating system.

Translational Vibration

Translational vibration (which is sometimes called "lateral vibration") involves the oscillatory motion of the drive shaft about its ideal "at rest" centerline. The shaft has natural frequencies and assumes related mode shapes just as piping systems do, and all of the equations and analyses on piping vibration in Section 22-10 also apply to drive shafts.

Drive shafts are much more sensitive to vibration problems than are piping systems, because the drive shaft rotates and any unbalance in the drive shaft or misalignment in the coupling causes the shaft to vibrate or deflect laterally, which results in "shaft whip." This vibration is very difficult to measure directly, but it can be measured indirectly via sensors at the shaft support bearings. Avoiding rotational speeds that match or excite the natural or resonance frequencies of the translational vibration of the drive shaft is imperative. These speeds are called "critical speeds," and they are determined entirely by the physical properties of the shaft.

If the drive shaft is assumed to be a uniform, straight, steel tube with a circular cross-section (or a solid cylinder if $ID = 0$), Equation 22-11 can be rearranged into a somewhat simpler form for a quick evaluation of the lowest natural frequency.

$$f_1 = \frac{\beta}{L^2}\sqrt{(OD)^2 + (ID)^2}$$
$$\beta = \frac{\gamma_1^2}{8\pi}\sqrt{\frac{E}{\rho}} \qquad (22\text{-}22)$$

where f_1 is the first-order shaft resonance in hertz, OD is the outer diameter of the shaft in millimeters (inches), ID is the inner diameter of the shaft in millimeters (inches), L is the effective length of the shaft in millimeters (inches), and β is 2.03×10^6 mm/s in SI units for universal (pinned) joint couplings or 4.6×10^6 mm/s for direct-coupled (fixed) shafts. In U.S. customary units, β is 7.9×10^4 in./s for pinned joint couplings and 1.79×10^6 in./s for fixed shafts. A drive shaft also has other modes of lateral vibration at higher frequencies, but these are usually not problem frequencies because the rotational speed of shafts is usually below the first resonance.

When using Equation 22-22 to compute the resonance frequencies of the drive shaft, the length, L, of the shaft should be measured from the center of the universal joints on both the driving and the driven ends (see Figure 22-17). Steady bearings may be used as intermediate supports to reduce L and increase the resonance frequency out of the region of concern.

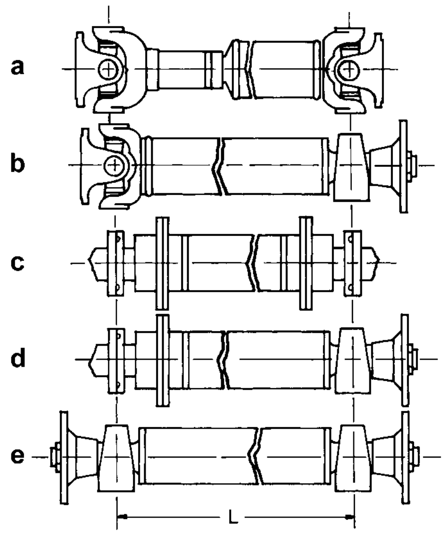


Figure 22-17. Typical drive shaft couplings. (a) Universal joint drive shaft. Vibration is accentuated if the two shafts are not exactly coaxial; (b) universal joint coupling shaft with right end flanged to connect with another shaft; (c) solid coupled shaft; (d) solid coupled coupling shaft; (e) middle shaft (with two support bearings) in a series of three shafts.

To avoid operating at a critical speed, the drive shaft should be designed to ensure that the highest rotational shaft speed is less than 75% of the lowest natural frequency. Furthermore, rotational speeds that are nearly equal to one-half of the lowest bending mode frequency of the shaft should be avoided because secondary moments from universal joints produce an excitation frequency at twice the shaft speed. Combining these two criteria gives two acceptable ranges for the lowest shaft frequency: (1) 1.4 to 1.6 times the shaft rotational speed, or (2) at least 2.5 times the shaft rotational speed.

Although the diameter of a drive shaft can be determined by the length of the shaft and the maximum operating speed, it is good design practice to limit the length to no more than 30 times the shaft diameter. Extremely long shafting must be divided into sections supported by steady bearings mounted on cross beams at the joints. If multiple sections are required, each section should, if possible, be of approximately equal length. The resonance frequencies of, say, four sections of nearly equal length are almost the same, but, if the lengths are substantially different, the additional resonance frequencies complicate the problem.

Support of Steady Bearings

When a drive shaft is divided into more than one section, the intermediate bearings must be supported adequately to resist not only the static load but also the dynamic load induced by the shaft rotation. Auxiliary steel members attached directly to the building structure usually support the intermediate bearings. A common rule of thumb is to make sure that the lowest natural frequency of the support structure is at least four times the shaft rotational speed. If the lateral support of the steady bearings does not meet this criterion, the boundary conditions will not approximate the "pinned" or "fixed" conditions and the first resonance frequency of the shafts will be lower than the calculated value.

The natural frequency of a single span support beam is given by Equation 22-11. The values of γ_n for clamped–clamped and pinned–pinned beams are given in Table 22-8 and, for modes 1 and 2, in Figure 22-11.

The length of the beam is often such that its lowest natural frequency is less than the recommended minimum of four times the shaft rotational speed. If so, it may be necessary to provide intermediate supports for the beam. If the main beam is supported at the center of the span, the same equation can be used (with modified values of γ_n) to compute the new resonance frequencies. Blevins [5] gives modified values of γ_n for various boundary conditions and positions of the intermediate support. In the special case of a center support with pinned boundary conditions (no displacement, but unrestrained rotation), the modified values of γ_n are given in Table 22-9. Most real-world boundary conditions for "rigidly" attached beams are such that some displacement and rotation occur at the ends of the beam no matter how they are fastened to the structure. For this reason, a more suitable choice for the frequency coefficient, γ_n, is a number somewhere between the theoretical values for the clamped–clamped and pinned–pinned boundaries.

Table 22-8. Frequency Coefficients for Bending Modes of Straight Beams

Mode (n)	Clamped–clamped (γ_n)	Pinned-pinned (γ_n)
1	4.73	3.14
2	7.85	6.28
3	11.00	9.42
4	14.14	12.57

Table 22-9. Frequency Coefficients for Bending Modes of Straight Beams with Center Supports[a]

Mode (n)	Clamped–clamped (γ_n)	Pinned-pinned (γ_n)
1	7.8	6.4
2	9.5	7.8
3	14.2	12.6
4	15.7	14.2

[a]The distance L is the full length of the beam—not the distance from the end to the center support.

Torsional Vibration

Torsional vibration problems occur less frequently than translational vibration problems, but they can be just as troublesome and sometimes catastrophic. Because they are not detected by our senses until failure occurs, it is important to select equipment components that preclude this possibility.

Torsional vibration can be magnified significantly if a torsional exciting frequency is close to the torsional natural frequency. The disturbing force in Figure 22-18 is the dynamic torque that oscillates about the average torque. At all frequencies less than 1.3 times the natural frequency, some magnification of the disturbance occurs, whereas at greater frequencies the disturbing force is damped. The maximum magnification occurs at resonance, and the value 3.3 shown is an arbitrary choice

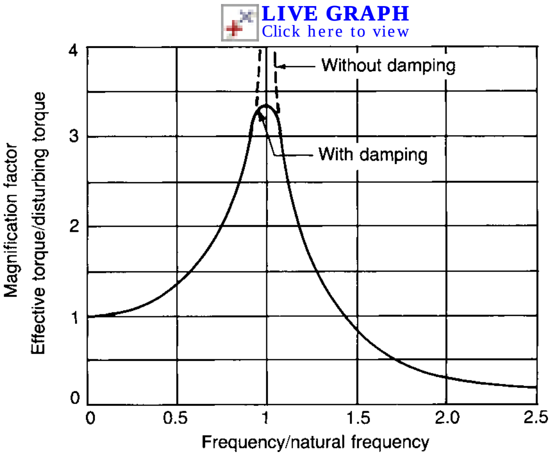


Figure 22-18. Magnification of a disturbing force in torsional vibration.

that is nevertheless typical. The maximum magnification can be calculated, but it depends on the damping factor, which is difficult to ascertain. The simplest and best procedure for designing shafts is to position the torsional natural frequency well outside any exciting frequencies. Typical exciting frequencies include: (1) the shaft rotational speed, (2) the pump vane passage frequency and its harmonics, and (3) the universal joints, which generate a frequency twice that of the shaft speed.

To avoid torsional resonances, keep the fundamental exciting frequencies less than 50% of torsional natural frequency. At times, it is necessary to run the system at speeds where harmonic exciting frequencies are above the natural frequency. This mode of operation is not a problem (1) if the exciting frequencies are more than 1.5 times the natural frequency, and (2) if the pass through the natural frequency range is accomplished quickly (which is normally true in electric motor pumping installations). If the exciting frequency is at least 1.5 times the torsional natural frequency, the effective disturbing system forces are less than the driver input disturbing forces. If the exciting frequency is less than 0.5 times the torsional natural frequency, the magnification factor will not exceed 1.3. Steel drive shafts typically have a minimum torsional elastic limit of 235 kPa (34,000 lb/in.^{2}) or an endurance limit (infinite life) of 117 kPa (17,000 lb/in.^{2}). The 1.3 stress magnification factor combined with the endurance limit means that a maximum allowable design torsional stress of $117/1.3 = 90$ kPa (13,000 lb/in.^{2}) ensures safety if torsional exciting frequencies are less than 0.5 or more than 1.5 times the torsional natural (resonance) frequencies.

In the simplest of systems (a pump driven by an electric motor with intermediate shafting), the torsional natural frequency of the system can be computed in SI units as follows:

$$f_o = \frac{1}{2\pi}\sqrt{\frac{K(J_m + J_p)}{J_m J_p}} \tag{22-23}$$

where f_o is the torsional resonance frequency in hertz, K is the torsional rigidity of the shaft in Newton-meters per radian, J_m is the mass moment of inertia of the motor in meters squared times kilograms, and J_p is the mass moment of inertia of the pump in meters squared times kilograms.

In U.S. customary units, K has units of pounds force-inches per radian and J_m and J_p have units of inches squared times pounds mass. The mass moment of inertia of a motor rotor is approximated by that of a cylinder of equal weight and diameter in which

$$J = \frac{D^2 M}{8} = \frac{D^2 W}{8_g} \tag{22-24}$$

where D is the diameter in meters (inches), M is the mass in kilograms (pounds mass), and W is the weight in Newtons (pounds force). Approximating the mass moment of inertia of the impeller and the water within the casing is neither easily done nor likely to be accurate, so obtain it from the equipment manufacturer. Accurate drive shaft torsional rigidity values can be obtained from computer programs (usually proprietary) or calculated directly by analyzing the shaft in sections and combining the rigidity constant, K, for each individual section to obtain the torsional rigidity of the entire shaft. The torsional rigidity of the stubs (the connecting link between two shafts), however, depends on the design—which is different for each manufacturer—so the gross torsional rigidity of shaft and stub must also be obtained from the maker.

The torsional rigidity of a uniform, hollow, circular shaft of length L is expressed as

$$K = \frac{\pi G}{32L}\left[(OD)^4 - (ID)^4\right] \tag{22-25}$$

where G is the modulus of elasticity of steel in shear, which is 82.7×10^9 Pa(12×10^6 lb/in.^{2}), and linear dimensions are in meters (inches). For solid shafting, use $ID = 0$. If a shaft is made up of two or more sections of shafting with different physical properties, the torsional rigidity of the entire shaft can be computed from the torsional rigidity of the individual sections as

$$K = \frac{1}{\sum_{i=1}^{n}(1/K_i)} \tag{22-26}$$

Example 22-4
Torsional Vibration in a Shaft

Problem: A two-vane, end-suction centrifugal pump is driven by a 75-kW (100-hp), variable-speed, direct-drive electric motor at speeds varying from 840 to 1200 rev/min (14 to 20 Hz). The power shafting is a universal joint, 3.353-m (132-in.) drive shaft (see Figure 22-17a) with 2.67 m (105 in.) of 101.6-mm (4-in.) OD tubing and a wall thickness of 2.108 mm (0.083 in.). The manufacturer gives the mass moment of inertia as $J_p = 1.729\,\text{kg}\cdot\text{m}^2(15.3\,\text{lbm}\cdot\text{in.}^2)$ and $J_m = 1.345\,\text{kg}\cdot\text{m}^2(11.9\,\text{lbm}\cdot\text{in.}^2)$. The torsional rigidity of the drive shaft stubs (excluding the tubing between the stubs) is given as 31,400 N · m/rad (277,802 lb · in./rad).

Evaluate the translational and torsional resonances of the shafting and determine if the system is within acceptable guidelines. If intermediate support bearings are required, size the structural members within recommended guidelines.

Solution:

Lateral resonance. The rotating frequencies of the shaft are 14–20 Hz (fundamental) and 28–40 Hz (second harmonic and pump vane passage frequency). From Equation 22-22, the first-order lateral resonance of the shaft is

SI Units

$$f_1 = \frac{2.03\times10^6}{(3353)^2}\sqrt{(101.6)^2+(97.384)^2} = 25.4\text{ Hz}$$

U.S. Customary Units

$$f_1 = \frac{7.99\times10^4}{(132)^2}\sqrt{(4)^2+(3.834)^2} = 25.4\text{ Hz}$$

The maximum shaft speed is 20 Hz, which is 78% of the fundamental resonance of the shaft. Ideally, the shaft speed should not exceed 75% of the first resonance, so this shaft speed is a marginal condition.

Torsional resonance. To compute the torsional resonance frequency, first determine the shaft rigidity, K_s, from Equation 22-25.

$$K_{\text{tubing}} = \frac{\pi(82.7\times10^9)\left[(0.1016)^4-(0.097384)^4\right]}{32(2.67)} = 50{,}526\ \text{N}\cdot\text{m/rad}$$

$$K_{\text{tubing}} = \frac{\pi(12.0\times10^6)\left[(4)^4-(3.834)^4\right]}{32(105)} = 447{,}935\ \text{lb}\cdot\text{in./rad}$$

From Equation 22-26:

$$K_{\text{shafting}} = \left[(31{,}400)^{-1}+(50{,}526)^{-1}\right]^{-1} = 19{,}368\ \text{N}\cdot\text{m/rad}$$

$$K_{\text{shafting}} = \left[(277{,}802)^{-1}+(447{,}935)^{-1}\right]^{-1} = 171{,}463\ \text{lb}\cdot\text{in./rad}$$

Determine the torsional resonance frequency from Equation 22-23:

$$f_o = \frac{1}{2\pi}\sqrt{\frac{(19{,}368)(1.729+1.345)}{(1.729)(1.345)}} = 25.4\text{ Hz}$$

$$f_o = \frac{1}{2\pi}\sqrt{\frac{(171{,}463)(15.3+11.9)}{(15.3)(11.9)}} = 25.4\text{ Hz}$$

This is below the recommended minimum of 1.5 times 20 Hz (the highest shaft speed) and is, thus, unacceptable.

Increasing the shaft OD to 127 mm (5 in.) and the ID to 120 mm (4.72 in.) would increase the first-order lateral resonance to 31.6 Hz and the torsional resonance to 29.7 Hz. This modification would nearly meet the torsional criteria, but the lateral resonance may be excited by the second harmonic of the shaft speed at 948 rev/min (or 15.8 Hz). An alternative solution would be to retain the original shaft diameters and split the shaft into two equal sections with

an intermediate support bearing. The fundamental lateral shaft resonance then becomes 101.8 Hz, which is acceptable, because $f_1/f_d = 5.09$. The new torsional spring constant of each section is $101{,}052\,\mathrm{N \cdot m/rad}$, which yields a total spring constant of $23{,}956\,\mathrm{N \cdot m/rad}$ and a torsional resonance of 58.2 Hz, which is also acceptable.

If the intermediate shaft bearing is supported by a section of steel angle 20.3 cm × 20.3 cm× 1.9 cm (8 in. × 8 in.× ¾ in.) thick that weighs 568 N/m (38.9 lb/ft), has a moment of inertia about both axes of 2.9×10^{-5} m^4 (69.7 in.^{4}), and spans 6.10 m (20 ft) with fully restrained end points, the fundamental resonance frequency of this beam is, from Equation 22-11 and Figure 22-11,

SI Units

$$f_1 = \frac{(4.73)^2}{2\pi(610)^2}\sqrt{\frac{(2 \times 10^{11})(2.9 \times 10^{-5})(9.81)}{568}} = 30.3\,\mathrm{Hz}$$

U.S. Customary Units

$$f_1 = \frac{(4.73)^2}{2\pi(240)^2}\sqrt{\frac{(29 \times 10^6)(69.7)(386)}{(38.9/12)}} = 30.3\,\mathrm{Hz}$$

The computed resonance frequency of the beam is insufficient to meet the conservative criterion of four times the maximum operating speed, but if a center support (with pinned boundary conditions) were added to this beam, the lowest beam frequency would be increased by a factor of $(7.85/4.73)^2 = 2.75$. The fundamental beam frequency would be increased to $2.75 \times 30.4 = 83.6$ Hz, which is 4.18 times the maximum operating speed of 20 Hz and does meet the recommended criterion. If the bearing were to be located at the center of the main beam (where the center support beam is attached), this problem would not arise because the vibrating force would be applied at a node of this first beam resonance, and it would not be excited.

22-12. Vibration of Structures

Pumping station equipment can cause structures to vibrate at levels exceeding normally acceptable criteria [19], particularly if the equipment is operating at a speed that matches one of the low-order resonance frequencies of the structure. In the design phase, it is too difficult to predict accurately the natural frequencies of supporting structures (building floor slabs, pedestals, etc.), but there are some simple calculations that can be made to determine whether a potential problem exists.

The lowest natural frequency of a supporting structure can be estimated from basic information concerning the static load-carrying capability of the structure. This information is often available directly from the structural engineer. Assuming that the structural member undergoes a static deflection directly proportional to the dead load of the equipment, the dynamic behavior of the structural member is similar to that of a steel spring (for small deflections). If the maximum static deflection of the structure (due to the weight of the equipment), d, is expressed in millimeters, the natural frequency of the structure in hertz is approximately

$$f_o = 3.13\sqrt{1/d} \tag{22-27a}$$

If d is expressed in inches,

$$f_o = 15.8\sqrt{1/d} \tag{22-27b}$$

This expression is approximately valid (±25%) only when $d < L/500$, where L is the smallest cross-dimension (width or height) of the structure, but this condition should be satisfied by most pumping station buildings. Although the correlation between the vibration of a point mass on a spring and the distributed mass of a concrete floor may seem unreasonable, the two systems vibrate at nearly the same frequency, because the use of deflection in Equation 22-27 tends to compensate for the differences between the two systems.

Much more accurate determinations of structural resonance frequencies can be obtained by modal analysis and other more elaborate techniques, but these methods are usually beyond the scope of most pumping station projects. Determining the maximum structural deflection caused by the static weight of the equipment is sometimes complicated, but approximations that provide an upper limit for d are usually adequate. Practical solutions lie in the close cooperation between the project leader and the structural engineer, which illustrates why a project leader must be much more than an administrator who assigns tasks and responsibilities to others.

Ideally, vibrating equipment should not operate at a rotational speed near the fundamental frequency of the supporting structure. Although problems are unlikely if the rotating speed is 5% greater than or 5% less than the resonance frequency, a 25% safety margin on each side is recommended.

Natural frequencies of a variety of structural members are given by Blevins [5]. In general, the natural frequency of any structure is proportional to the square root of the composite moment of inertia, I, divided by the total mass of the system.

$$f_o \propto \sqrt{I/M} \tag{22-28}$$

Any efforts to change the natural frequency of an existing system, therefore, should be devoted to these two parameters. Notice that doubling the mass of the system (equipment plus structure) reduces the natural frequency by 29% $(1 - \sqrt{0.5})$ if it is assumed that mass can be doubled with no change in the moment of inertia. However, structural modifications usually affect both M and I, so the modifications should be evaluated carefully prior to making the change.

Example 22-5
Vibration of a Floor

Problem: A 300-kVA (400-hp) variable-speed induction motor (900 to 1200 rev/min) is to be mounted on a 100-mm- (4-in.-) thick housekeeping pad over a 200-mm- (8-in.-) thick concrete slab that is continuously supported by the perimeter wall as well as by a concrete beam that divides the slab in the center. The motor is mounted with the shaft vertical and is rigidly bolted to the housekeeping pad. The drive shaft penetrates the floor beside the beam and in the center of the pad. The structural engineer estimates the static deflection of the floor slab (due to the weight of the motor) to be 1 mm (0.04 in.). Evaluate the potential for a floor vibration problem and provide a solution if necessary.

Solution: The driving frequency range of the motor is 15 to 20 Hz for the fundamental and 30 to 40 Hz for the second harmonic. Note that because vibrating machinery produces excitation forces in all directions, the orientation of the motor is of little practical significance.

From Equation 22–27 the estimated natural frequency of the floor is as follows:

SI Units

$$f_o = 15.8\sqrt{\frac{1}{1}} = 15.8\,\text{Hz}$$

U.S. Customary Units

$$f_o = 3.13\sqrt{\frac{1}{0.04}} = 15.7\,\text{Hz}$$

which is in the operating range of the pump's fundamental frequency.

The natural frequency of the floor should be at least 25 Hz (1.25 times the maximum fundamental driving frequency of 20 Hz). Solving for d in Equation 22-27, the maximum allowable deflection is

SI Units:

$$25 = 15.8\sqrt{\frac{1}{d}}$$
$$d = 0.4\ \text{mm}$$

U.S. Customary Units:

$$25 = 3.13\sqrt{\frac{1}{d}}$$
$$d = 0.016\ \text{in.}$$

This would require a reduction of more than 50% from the original conditions, and it might be very difficult for the structural engineer to design for so small a deflection. As an alternative, consider reducing the natural frequency of the floor so that the lowest driving frequency is about 25% above the natural frequency of the floor. This modification would require a floor

frequency of 15 Hz/1.25 = 12 Hz, which (again from Equation 22–27) corresponds to a static deflection of

$d = 1(15.8/12)^2 = 1.73\,\text{mm}$ $\qquad$ $d = 0.04(15.8/12)^2 = 0.07\,\text{in.}$

By directing the structural engineer to alter the supporting structure (decreasing the slab thickness, modifying the beam supports, etc.) to meet the desired static deflection, the floor resonance problem can be avoided. It should be noted that the floor would have other natural frequencies higher than the fundamental frequency approximated by Equation 22-27, and these natural frequencies may become resonant with the higher harmonics of the pump. Evaluating such a problem is so complex that it is beyond the scope of this chapter.

22-13. Noise

The intent of this section is to provide project leaders with some elementary tools to evaluate the severity of sound propagation through the air, including transmission through barriers. A discussion of structure-borne sound (i.e., structural vibrations resulting in subsequent noise radiation from solid surfaces) is of little importance in most pumping stations and, hence, is not included. Heed OSHA regulations and local ordinances, as explained in Section 22-6.

Outdoor Sound Propagation

Noise radiates from a source in all directions, and sound propagates in air at a nominal velocity of 344 m/s (1130 ft/s) at 20°C (70°F). The actual velocity varies somewhat with temperature and barometric pressure, but for practical purposes this dependence can usually be ignored. The intensity of sound decreases with the square of the distance from a source (much like the intensity of light from a light bulb) as the acoustical energy spreads out over a larger and larger surface area. In addition to this "spreading loss" there is also a slight loss due to atmospheric absorption (conversion of acoustical energy into heat), but this can be neglected in most applications.

The sound pressure level, L_p, at a point can be expressed as a function of the distance to the source, x, and the sound pressure level from the source at a reference distance, x_o, in the same direction. This spreading loss can be expressed as

$$L_p(x) = L_p(x_o) + 20\log_{10}(x_o/x) \qquad (22\text{-}29)$$

The effects of partial or full barriers, the reflections from surrounding surfaces, and the directivity of the source are not considered in Equation 22-29, and in an enclosed room the equation may be valid only within a few feet of the source. For an outdoor source, the expression may be reasonably accurate for hundreds of meters provided that (1) there are no barriers or major reflecting surfaces (such as buildings), and (2) the direction of the reference measurement is the same as the direction of interest. In the special case of $x = 2x_o$ (where the location of interest is twice as far from the source as the reference position), the difference in the noise levels at the two positions is 6 dB. Thus, whenever the distance to the source is doubled, the sound intensity drops by 6 dB. Likewise, whenever the distance to the source is reduced by 50%, the sound intensity increases by 6 dB. These general rules are only strictly valid in a free field (i.e., an environment without reflective or absorptive surfaces), but they are approximately valid in many typical situations.

Indoor Sound Propagation

In an enclosed or partially enclosed room, noise radiating from equipment is reflected from the room boundaries and builds up to a higher noise level than if the equipment were located outside. This buildup of noise caused by acoustical energy being contained within the room creates a "reverberant area or field" in the room.

The reverberant field concept is illustrated in Figure 22-19, where noise level is plotted as a function of distance from a noise source. In the region near the source (the direct field), the noise level falls off at the 6 dB per double distance rate because the acoustical energy radiating directly from the source overpowers the reverberant field. As the listener moves farther from the source, the noise level reaches a sound pressure level that is relatively constant. It is this region that is called the reverberant field. In SI

LIVE GRAPH
Click here to view

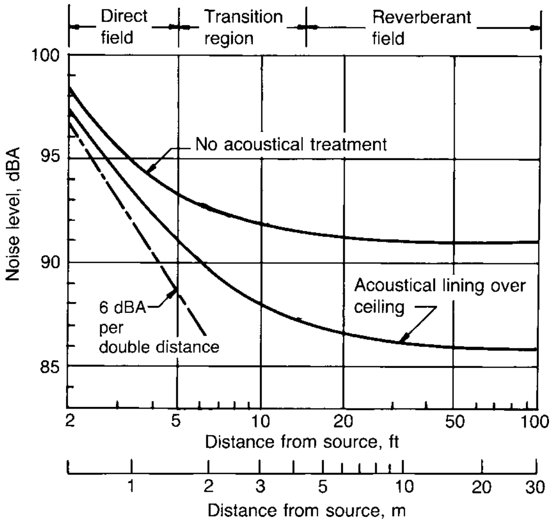


Figure 22-19. Noise level in a room.

units, the noise level in the reverberant field can be approximated by the following expression:

$$L_{p(rev)} = L_p(1\,\text{m}) - 10\log_{10}(NRC \times S) + 17 \quad (22\text{-}30a)$$

where $L_p(1\,\text{m})$ is the sound pressure level 1 m from the source, S is the total surface area of the interior surfaces of the room in square meters, and NRC is the average noise reduction coefficient of the room surfaces exposed to the noise.

In U.S. customary units, the expression is

$$L_{p(rev)} = L_p(3\,\text{ft}) - 10\log_{10}(NRC \times S) + 27 \quad (22\text{-}30b)$$

where S is in square feet.

The noise reduction coefficient is a single number (between 0 and 1.0) that approximates the ratio of the absorbed to the incident acoustical energy. For example, a material with an NRC rating of 0.60 would absorb 60% of the incident sound and reflect the other 40% of the sound back into the room. A list of common building materials and approximate NRC ratings is shown in Table 22-10. Note that most hard, smooth surfaces have low NRC ratings, while most soft surfaces have high NRC ratings.

The noise level in the reverberant field can be reduced by adding materials with high NRC values. The noise reduction achievable depends primarily on the product of the surface area and the NRC rating of the added acoustical material, but it also depends on the average NRC rating and the total surface area of the existing room. For pumping stations with little or no existing acoustical materials, the noise reduction by adding acoustical materials can be approximated by

$$NR = 10\log_{10}[1 + 10(S_a/S)NRC_a] \quad (22\text{-}31)$$

where S_a and NRC_a are the surface area and NRC rating of the added acoustical material. When a material is placed over an existing material (shielding it from the noise field), the NRC_a value used in Equation 22-31 should be the net increase in NRC. For example, if a 5-cm- (2-in.-) thick fiberglass insulation board is placed under a steel deck roof, the NRC_a should be $0.95 - 0.05 = 0.90$ (see Table 22-10). Hanging baffles (or unit absorbers) are sometimes rated in terms of metric or English sabins per unit rather than by NRC. These numbers can easily be converted to NRC values by dividing the sabins per unit by the total surface area of the absorber to yield NRC. Equation 22-31 is represented in graphical form by Figure 22-20.

Table 22-10. Values of *NRC* for Common Building Materials Based on Total Surface Area (All Sides of the Absorber)

Material	Approximate *NRC*
Concrete (poured or sealed)	0.05
Steel (and other hard metals)	0.05
Concrete block (porous)	0.05–0.10
Wood (all types)	0.05–0.10
Glass	0.05–0.10
Gypsum board	0.05–0.10
Floor tile (ceramic, vinyl, etc.)	0.05–0.10
Acoustical masonry blocks	0.35–0.75
Acoustical ceiling tile (mineral fiber)	0.45–0.75
Carpet	0.45–0.65
Acoustical roof decking	0.45–0.85
Fiberglass insulation board 25 mm (1 in.) thick	0.75–0.85
Fiberglass insulation board 50 mm (2 in.) thick	0.85–1.00
Acoustical baffles (hanging)	0.65–0.95
Acoustical panels (wall or ceiling mounted)	
12 mm (½ in.) thick	0.35–0.60
25 mm (1 in.) thick	0.65–0.80
50 mm (2 in.) thick	0.85–1.00
Spray-on acoustical system	
12 mm (½ in.) thick	0.35–0.60
25 mm (1 in.) thick	0.65–0.80
37 mm (1-½ in.) thick	0.85–0.95

LIVE GRAPH
Click here to view

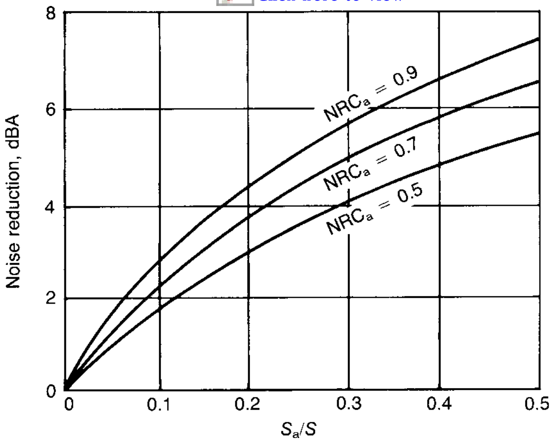


Figure 22-20. Reverberant field noise reduction with acoustical treatment on walls and ceiling.

Noise Sources

There are numerous noise-generating devices typically found in pumping stations, but the most common noise problems stem from engine generators, air compressors, fans, electric motors, and pumps.

Equipment manufacturers should be able to provide reasonably accurate noise data for a particular piece of equipment. The noise (sound pressure) levels in Tables 22-11 and 22-12 apply to the normal operation of equipment in good repair, but they are approximate and should be used only when manufacturer's data are unavailable or, for some reason, unreliable. Equipment with worn bearings or damaged internal parts may generate higher noise levels. Avoid comparing noise data in different formats. For example, sound power levels are not equivalent to sound pressure levels, even though they may appear to have the same units. (Actually, they do not have the same units.) Also be aware that sound pressure levels may be A-weighted, B-weighted, C-weighted, D-weighted, or even unweighted. They may also be octave band, one-third-octave band, or even narrow band. To be applicable to the expressions presented in this chapter, all noise levels must be A-weighted sound pressure levels at some identified direction and distance from the source.

Table 22-11. Typical Noise Levels from Diesel Engines[a]

		A-weighted noise level, [dBA at 1 m (3.28 ft)]	
Horsepower	(rev/min)[b]	Casing radiated	Exhaust radiated
under 50	1200	94	107
50–100	1200	96	109
100–200	1200	99	112
200–400	1200	102	115
400–800	1200	105	118
800–1600	1200	108	121

[a]Use the same data for spark-ignition engines. They are quieter, so these data are more conservative.

[b]For other engine speeds add $30 \log_{10}$ [(rev/min)/1200].

Table 22-12. A-weighted Sound Pressure Levels of Electric Motors [dBA at 1 m (3.28 ft)]

		Speed (rev/min)	
Power (kVA)	Horsepower	450–900	900–1800
7.5–37	10–50	88	92
37–75	50–100	92	96
75–150	100–200	95	99
150–300	200–400	98	102
300–600	400–800	101	105

Engine-Powered Generators

Most of the noise generated by engine-powered generators is caused by the engine firings within the individual cylinders. This acoustical energy escapes via direct transmission through the engine block and the exhaust stack into the surrounding environment. Although some noise exits the combustion air intake, it is usually much lower in intensity because it is blocked off during the ignition while the gases in the cylinder are exhausting. Mechanical noise from the internal rotating parts in the engine and the generator are usually insignificant. The radiator fan is the only other significant noise source on the machine (unless it is remotely cooled). Noise levels from these fans can be found in the subsection entitled "Fans," below.

Approximate A-weighted sound pressure levels for diesel engines are given in Table 22-11. Casing-radiated and exhaust noise levels are shown separately because the exhaust stack outlet is usually remote from the engine. The conditions for the exhaust noise data are predicated on (1) no muffler in the exhaust pipe, (2) an exhaust directed upward, and (3) a measurement point 1 m (3.28 ft) to the side (90 degrees from the exhaust direction) at the elevation of the opening. Under these assumptions there is very little directionality to the source (i.e., the noise level is independent of azimuth).

The noise levels given in Table 22-11 are for steady operation at fixed speed. Start-up noise may be higher, particularly if compressed-air starters are used. Compressed-air starters should be avoided in residential areas because noise levels inside the plant can exceed 120 dBA.

Silencers are nearly always required to bring the exhaust stack noise levels down to meet environmental noise ordinances. Depending on the silencer design, the engine, and the operating speed, silencers usually reduce noise by 10 to 40 dBA. Most manufacturers provide exhaust noise data for their engines, with a variety of silencers, matched to the engine operating at fixed-governed speeds. In general, it is best to locate the silencer close to the engine exhaust with a flexible coupling between the two, as shown in Figure 22-21. This location minimizes the length of exhaust pipe with high internal noise, some of which radiates outward through the pipe wall. The preferred technique of silencing a diesel generator is illustrated in the figure.

Electric Motors and Pumps

Electric motors can be a significant source of noise in pumping stations. In most applications, the noise radiated by a motor-driven pump is generated by the electric motor, not the pump. There are some instances where the pump can be noisier than the motor, but these are usually limited to high-speed (pump frequencies greater than 200 Hz), high-pressure, low-volume pumps such as gear pumps, which are not common in pumping stations. Approximate A-weighted noise levels from induction motors are shown in Table 22-12.

Air Compressors

Noise levels produced by air compressors vary widely with design and application. Very large machines that are designed to deliver volumes of compressed air greater than 50 standard L/s (100 scfm) can generate noise levels in excess of 120 dBA near the source. Air compressors common in pumping stations require only a fraction of this capacity and, as a result, are much quieter (but still rather noisy). Typical noise levels from these machines at 1 m (3.28 ft) are in the 90- to 100-dBA range.

Fans

The noise levels produced by ventilating fans are a function of the fan blade and housing design, the number of blades, the tip speed of the blades, and the volume flow and static pressure created by the fan. The most popular types for use in pumping stations are propeller fans and centrifugal fans. Propeller fans are commonly used in an open condition (no housing or ductwork connected to the fan) to ventilate spaces through an opening in the wall or roof of the space. Under these conditions the static pressure requirements of the fan are minimal—usually less than 25 mm (1 in.) of water. Centrifugal fans, on the other hand, are commonly used in ducted applications where static pressure requirements are more substantial. Noise levels radiated from centrifugal fans are computed as shown in the ASHRAE handbook [24].

The approximate noise levels from propeller fans of various diameters and speeds are given in Figure 22-22. The blade design (airfoil, flat, etc.) also has some effect on the resulting noise level, but the noise

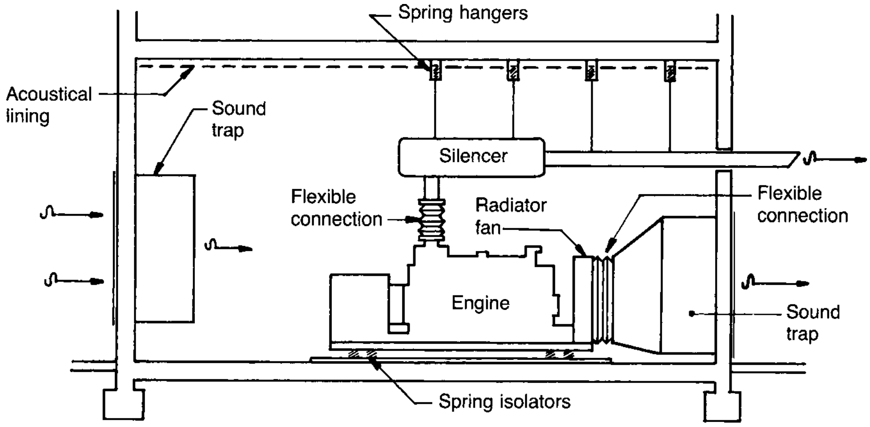


Figure 22-21. Noise control for a diesel generator.

LIVE GRAPH
Click here to view

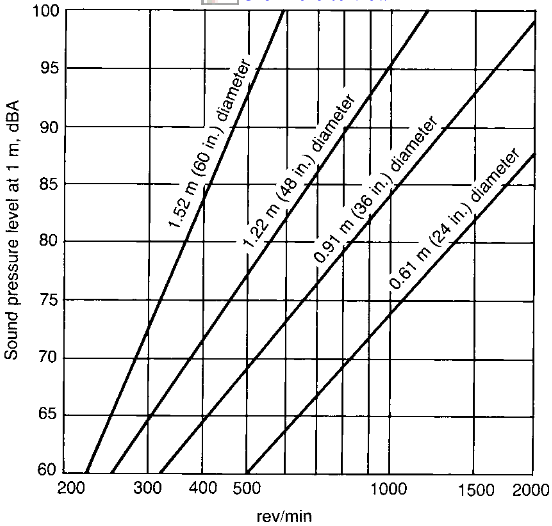


Figure 22-22. Noise levels of propeller fans.

produced by most propeller fans should fall within 5 dBA of these curves.

Multiple Sources

When two or more sources of noise operate simultaneously, the total noise level at a point from all sources can be computed from the following expression:

$$L_p = 10 \log_{10} \sum_{i=1}^{m} 10^{N_i} \qquad (22\text{-}32)$$

where L_p is the total noise level in dBA, N is $L_i/10$, L_i is the noise level of the ith source, and m is the total number of sources. This expression is valid for multiple sources in a free field (i.e., outdoors) or within a room as long as the noise level from the direct field and the reverberant field are each evaluated independently as separate sources. If all sources have the same noise level at a point, Equation 22-32 reduces to

$$L_p = L_i + 10 \log_{10} m \qquad (22\text{-}33)$$

Noise Reduction by Enclosures

Noise levels from mechanical equipment can be reduced by (1) modifying the source, (2) adding acoustical materials to the room, or (3) isolating the source within a complete or partial enclosure. Techniques for modifying the source (e.g., reducing the speed or changing the equipment design) can often be very effective, but they may be costly and are usually undesirable for one reason or another. Adding acoustical materials to the room is somewhat expensive and only partially effective because it does not reduce the noise near the source. Usually a 5- to 8-dB noise reduction is the most that can be achieved (see Figure 22-20). Consequently, the most common method of reducing noise from machinery is to enclose it within partitions and/or barriers.

The noise reduction, NR, provided by a partition or wall separating two spaces is defined as the difference between the noise level on the source side, L_{p1}, and the receiver side, L_{p2}, of the partition. In general, the noise reduction is primarily a function of the transmission loss of the partition, the total area of the common partition through which the noise radiates, the physical size and acoustical properties of the rooms in question, and the frequency distribution of the noise on the source side. A complete description of the equations required to compute the noise reduction accurately is well beyond the scope of this text. However, the following simple equation is usually accurate to within ±8 dBA:

$$NR = L_{p1} - L_{p2} = \text{STC} - 5 \qquad (22\text{-}34)$$

The STC term in Equation 22-34 refers to the sound transmission class, STC, of the partition. The STC rating of a partition is a single number descriptor determined from laboratory measurements of the sound transmission loss as a function of frequency. The STC ratings of some common building materials are listed in Table 22-13.

For materials not listed in the table, the STC ratings can be obtained from data in Harris [25] or approximated from the surface weight of solid materials using the following formula:

$$\text{STC} = 8 + 17 \log_{10} w \qquad (22\text{-}35a)$$

where w is the surface mass of the material in kilograms per square meter.

In U.S. customary units, the expression is

$$\text{STC} = 20 + 17 \log_{10} w \qquad (22\text{-}35b)$$

where w is the surface weight in pounds mass per square foot.

Table 22-13. Approximate STC Ratings for Common Building Materials

Material	STC[a]
Poured concrete, 150–300 mm (6–12 in.) thick	50
Hollow concrete block, 200 mm (8 in.) thick, unpainted	40
Steel acoustical panel, 100 mm (4 in.) thick	40
Drywall partition,[b] 150 mm (6 in.) thick	35
Drywall partition,[c] 150 mm (6 in.) thick	40
Safety glass, 6.35 mm (¼ in.) thick	30
Hollow metal door, without seals	20
Metal louver, more than 50% open area	2

[a]Note that the STC values are *not* additive for composites: two partitions each with an STC of 35 *do not* yield an STC of 70. Each composite must be separately tested.

[b]Metal studs with one layer of gypsum board on each side.

[c]Batt insulation between metal studs and gypsum board on each side.

In most applications, an enclosure is made from more than one material. For example, the door and roof are usually constructed differently than the walls, and there may be openings for ventilation. To evaluate this complex situation, calculate the composite sound transmission class, STC_c. The composite STC can be determined from substituting the exposed areas and STC of each component of the barrier system into the expression

$$STC_c = -10\log_{10}\left[1/s\sum_{i=1}^{n} A_i 10^{-N_i}\right] \tag{22-36}$$

where N is $STC_i/10$, A_i and STC_i are the exposed surface area and STC rating of the ith material, and s is the total exposed surface area of all components in the direction of the listener. In most situations, the isolation provided by a composite system is only as good as its weakest component, unless that component has a surface area less than one-tenth of the other components. A unique but common example is the enclosure with an opening because openings have an STC of 0. The dependence of the composite STC on opening size is illustrated in Figure 22-23. Note that even small leaks can cause serious degradation of performance, particularly for partitions with high STC ratings.

Sound Traps

In most practical applications, enclosure openings are required for ventilation. Louvered openings are typ-

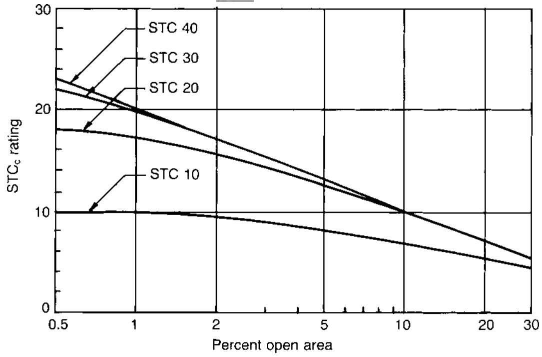


Figure 22-23. Dependence of composite STC rating on openings in partitions or enclosures.

ically more than 50% open area, so these openings present a major acoustical problem—a problem usually solved with sound traps. A sound trap or duct silencer is usually a rectangular steel box (but sometimes round for circular ducts) open at both ends with a packing of acoustical materials that create parallel baffles, as in Figure 22-24. The air passes straight through most units without changing direction. These devices are prefabricated in a variety of sizes and are furnished with laboratory-tested acoustic and aerodynamic performance data. The STC of an opening can, therefore, be significantly increased by inserting one or more sound traps into the opening in the barrier and sealing the perimeter airtight with acoustical (nonhardening or resilient) caulking.

The acoustical performance of a sound trap depends on its size and internal design. The length (the dimension in the direction of air flow) of the sound trap is the most important factor affecting performance. Sound traps are available in lengths ranging from 0.3 to 3 m (1 to 10 ft). Within each length, most manufacturers produce several designs that essentially vary in the size of the air passage (i.e., the percentage of open area). Units with larger air passages have lower acoustical ratings and lower aerodynamic pressure drops. Sound traps are rated by the dynamic insertion loss (in decibels) at a variety of frequencies (and not by STC values). The dynamic insertion loss is the difference in noise level with and without the sound trap in the system (with air flowing through the silencer). Aerodynamic and acoustic performance data for a few typical sound traps are given in Table 22-14. The STC of a sound trap can be approximated roughly by the manufacturer's dynamic insertion loss rating in the frequency band nearest the primary frequency of the noise source. The primary frequency would be the blade passage frequency for a fan or the firing rate frequency for a

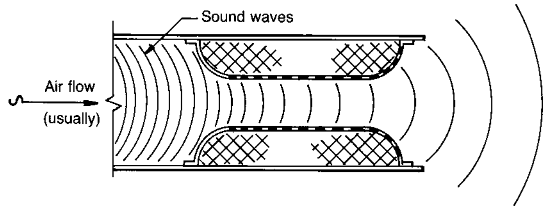


Figure 22-24. A typical duct silencer.

diesel engine. If the primary frequency of the noise is unknown, the insertion loss at 250 Hz may be used to approximate the STC of the sound trap.

Noise Reduction of Barriers

It is often neither practical nor economical to enclose a noise source completely. An often-used compromise is to erect a barrier between the noise source and the receiver to block the direct transmission of noise to the listener. For the barrier to be effective, it must (1) be solid (nonporous), and (2) completely block the straight-line path between the source and the listener.

Table 22-14. Performance Characteristics of Sound Traps. Face Velocity of 5.08 m/s (1000 ft/min).

Length m	Pressure drop ft	Approximate mm Hg	in. H_2O	STC rating
0.3	1	1.40	0.75	6
1.0	3	0.09	0.05	10
1.0	3	0.19	0.10	15
1.0	3	0.37	0.20	25
1.5	5	0.15	0.08	18
1.5	5	0.28	0.15	25
1.5	5	0.37	0.20	30
2.1	7	0.19	0.10	25
2.1	7	0.28	0.15	30
2.1	7	0.48	0.25	35

Chain-link fences, shrubs, or rows of trees are virtually useless as noise barriers, but earth berms and concrete walls have been used successfully. Barriers do not provide as much noise reduction as enclosures because the noise tends to go around the barrier.

The noise reduction provided by a barrier is primarily dependent on (1) the material of construction, (2) the height of the barrier, and (3) the presence of other adjacent reflecting surfaces. For an infinitely long solid wall blocking the line-of-sight transmission between two points (as shown in Figure 22-25) with no nearby reflecting surfaces, the noise reduction is approximated by the following dimensionless equation from Beranek [26]:

$$\mathrm{NR} = 20\log_{10}\left(\frac{\sqrt{2\pi N}}{\tan\mathrm{h}\sqrt{2\pi N}}\right) + 5 \qquad (22\text{-}37)$$

where N is the dimensionless Fresnel number, which is defined by

$$N = \frac{2f}{c}(A + B - r) \qquad (22\text{-}38)$$

in which f is the primary frequency of the noise source in Hz and c is the speed of sound in air in meters per second (feet per second). See Figure 22-25 for the definitions of A, B, and r, which are in meters (feet). Although very high barriers might reduce noise by more than 20 dBA, they usually do not because of sound transmission either through the barrier material or by another flanking path. In general, the barrier material should have an STC rating about 15 dB greater than the noise reduction predicted by Equation 22-37. Note that the noise reduction given by Equation 22-37 represents the change in sound level at the receiving position with and without the barrier.

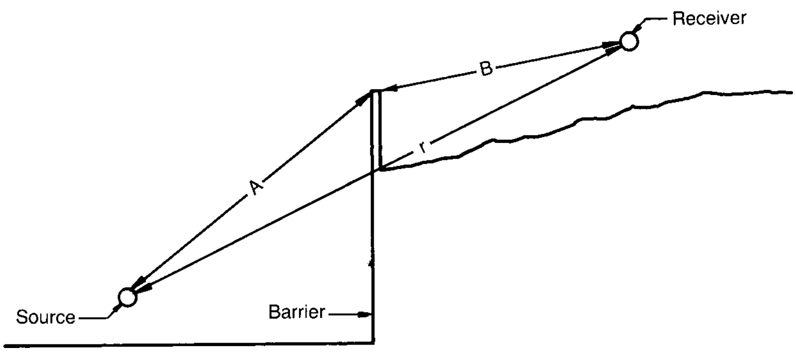


Figure 22-25. Noise reduction of barriers.

22-14. Reducing Exterior Noise

As shown in Example 22-6, the exterior noise radiating from a pumping station may stem from several sources, such as motors, pumps, engine generators, and cooling fans.

Noise is attenuated by (1) walls, (2) partitions or other enclosures (Table 22-13), (3) distance, and (4) silencers and sound traps. Reducing reverberant noise by acoustical treatment [acoustical tile, masonry blocks (if specially manufactured to do so), or thick fiberglass] also reduces radiating noise unless the source is near an opening in a wall. Placing motors below grade or using submersible pumps and motors is very effective. (But engines should not be below grade.) Fan noise can be reduced by using (1) larger diameter blades at lower rotational speeds, (2) more effective sound traps, and/or (3) exhausting through the roof. Reducing fan noise by using larger blades and lower air velocity carries the penalty of a larger louver, which decreases the STC of the wall (or roof), but the penalty can be overcome by permitting a higher pressure drop in the sound trap, thus making it more effective for suppressing both fan noise and generator-casing noise. In some situations, it may be desirable to provide cooling by refrigeration—an expensive solution.

When designing for noise suppression, contributing sources with noise levels that are 10 dBA or more below the total noise level allowed can usually be ignored because (from Equation 22-32) the effect on total noise level is less than 1 dB.

An additional margin of safety is possible, but it may become quite expensive. Generally speaking, noise control measures tend to follow the law of diminishing returns; that is, the cost of the first 10 dBA of noise reduction is generally much less expensive than the second 10 dBA, and so on. In situations where noise is important, it is usually cost-effective to hire a consultant who can take the responsibility of performing a detailed frequency analysis and specifying exactly what is required to meet code requirements without excessive safety margins.

Example 22-6
Reducing the Sound Level at a Pumping Station

Problem: A pumping station with concrete exterior walls 200 mm (8 in.) thick houses two 500-kVA (670-hp), 1800-rev/min standby diesel generators that supply power to six 200-kVA (270-hp), 600-rev/min direct-drive centrifugal pumps, two of which are back-ups. The layout of the pumping station is such that the nearest residential property is 60 m (197 ft) from the edge of the building, as shown in Figure 22-26. Compute the expected noise levels inside and

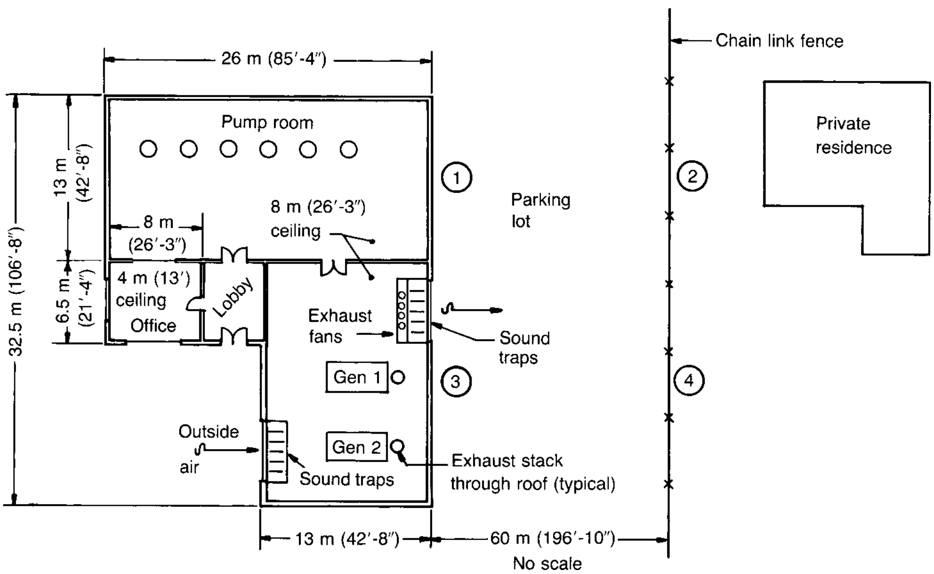


Figure 22-26. Pumping station for Example 22-6.

outside of the building with no special acoustical considerations, then determine the appropriate measures needed to reduce the acoustical impact to an acceptable level. Assume that the NRC for the untreated interior spaces is 0.10. An NRC of 0.05 is not realistic for real-life conditions because of absorption from miscellaneous fixtures, etc.

Solution:

Interior noise levels in pump room and office. The main noise sources in the pump room are the six electric motors that power the pumps. From Table 22-12, the expected noise level from each pump is 98 dBA at 1 m (98.8 dBA at 3 ft). The noise level in the reverberant field from each pump is evaluated from Equation 22–30. Assume the room dimensions to be 13 × 26 × 8 m (43 × 85 × 26 ft). The total surface area is $S = 1300\,\mathrm{m}^2 (14{,}000\,\mathrm{ft}^2)$. Use Equation 22-30 and Table 22-10.

SI Units

$$L_{p(rev)} = 98 - 10\log_{10}[(0.10)(1300)] + 17 = 93.8\,\mathrm{dBA} \simeq 94\,\mathrm{dBA}$$

U.S. Customary Units

$$L_{p(rev)} = 98.8 - 10\log_{10}[(0.1)(14{,}000)] + 26.5 = 93.8\,\mathrm{dBA} \simeq 94\,\mathrm{dBA}$$

The total noise level in the pump room is evaluated by using four noise sources for m in Equation 22-33. (Two of the six pumps are always off.)

$$L_p = 94 + 10\log_{10} 4 = 100\,\mathrm{dBA} \qquad L_p = 94 + 10\log_{10} 4 = 100\,\mathrm{dBA}$$

The noise level in the office can be computed by estimating the sound transmission through the $32\,\mathrm{m}^2 (345\,\mathrm{ft}^2)$ gypsum board partition with a $3\,\mathrm{m}^2 (32.4\,\mathrm{ft}^2)$ safety glass window separating the office from the pump room. Assuming the partition has an STC of 40 and the window has an STC of 30 (from Table 22-13), the composite STC is determined from Equation 22-36

$$\mathrm{STC_c} = -10\log_{10}\frac{1}{32}\left[(29)(10)^{-4} + (3)(10)^{-3}\right] = 37.3 \qquad \mathrm{STC_c} = -10\log_{10}\frac{1}{325}\left[(313)(10)^{-4} + (32)(910)^{-3}\right] = 37.3$$

From Equation 22-34, the noise reduction is

$$NR = 37.3 - 5 = 32.3 \simeq 32 \qquad NR = 37.3 - 5 = 32.3 \simeq 32$$

and the noise level in the office is approximately

$$L_p = 100 - 32 = 68\,\mathrm{dBA} \qquad L_p = 100 - 32 = 68\,\mathrm{dBA}$$

Interior noise in generator room. Noise data from the manufacturer indicate that the casing-radiated noise level from each unit (including the radiator fan) is 98 dBA at a distance of 3 m (9.84 ft), which can be converted to 1 m (3 ft) by Equation 22-29.

$$98 = L_p(1\,\mathrm{m}) + 20\log(1/3) \qquad 98 = L_p(3\,\mathrm{ft}) + 20\log(3/9.84)$$

$$L_p(1\,\mathrm{m}) = 98 + 9.5 = 107.5\,\mathrm{dBA} \qquad L_p(3\,\mathrm{ft}) = 98 + 10.3 = 108.3\,\mathrm{dBA}$$

The reverberant field level is computed using $S = 13 \times 19.5 \times 2 + (13 + 19.5)\,2 \times 8 = 1027\,\mathrm{m}^2$ $(11{,}055\,\mathrm{ft}^2)$, which yields (from Equation 22-30)

$$L_{p(rev)} = 107.5 - 10\log_{10}(0.10 \times 1027) + 17 = 104.4\,\mathrm{dBA} \text{ for each unit.}$$

$$L_{p(rev)} = 108.3 - 10\log_{10}[(0.1)(11{,}055)] + 26.5 = 108.3 - 30.4 + 26.5 = 104.4\ \mathrm{dBA} \text{ for each unit.}$$

When both units are operating, the noise level (from Equation 22-33) increases to

$$L_p = 104.4 + 10\log_{10}(2) = 107.4\,\text{dBA} \qquad L_p = 104.9 + 10\log_{10}(2) = 107.4\,\text{dBA}$$

The other major noise sources in the generator room are the six propeller fans 1 m (36 in.) in diameter located in the louvered opening. These fans rotate at 600 rev/min, and each fan generates a noise level of 75 dBA at 1 m, as shown by Figure 22-22. (For U.S. customary units, first use Equation 22-29 to obtain the noise level at 3 ft.) From Equation 22-30 the reverberant field noise level from each fan is

$$L_{p(\text{rev})} = 75 - 10\log_{10}(0.10 \times 1027) + 17$$

$$L_{p(\text{rev})} = 75 - 20.1 + 17 = 71.9\,\text{dBA} \simeq 72\,\text{dBA}$$

$$L_p(3\,\text{ft}) = 75 + 20\log_{10}(3.28/3) = 75.8$$

$$L_{p(\text{rev})} = 75.8 - 10\log_{10}[(0.1)(11{,}055)] + 27$$

$$L_{p(\text{rev})} = 75.8 - 30.4 + 26.5 = 71.9 \simeq 72\,\text{dBA}$$

and from all six fans, Equation 22-33 gives

$$L_p = 72 + 10\log_{10} 6 = 79.8\,\text{dBA} \qquad L_p = 72 + 10\log_{10} 6 = 79.8\,\text{dBA}$$

From Equation 22-32, the total noise from the six fans and the two generators is

$$L_p = 10\log_{10}[(10)^{7.98} + (10)^{10.74}] = 107.4\,\text{dBA} \qquad L_p = 10\log_{10}[(10)^{7.98} + (10)^{10.74}] = 107.4\,\text{dBA}$$

Exterior noise levels. Independently compute the noise levels from each source type and add them together by using Equation 22-32. First, consider the pump room. The interior noise level is 100 dBA (the reverberant field level) because the wall is not within the direct field of any of the pumps (see Figure 22-19). The STC rating of the partition is 50 (see Table 22-13), so if there are no penetrations or openings in the wall, the exterior noise level at position 1 (Figure 22-26) is, from Equation 22-34,

SI Units

$$L_p = 100 - NR = 100 - (50 - 5) = 55\,\text{dBA}$$

U.S. Customary Units

$$L_p = 100 - NR = 100 - (50 - 5) = 55\,\text{dBA}$$

Estimating the average distance from position 1 to the four motors as 12 m (40 ft) and from position 3 to the four motors as 72 m (235 ft), the pump noise level at position 3 is, from Equation 22-29,

$$L_p(1) = 55 + 20\log_{10}(12/72) = 39\,\text{dBA} \qquad L_p(2) = 55 + 20\log_{10}(40/235) = 39\,\text{dBA}$$

The generator room noise is evaluated in a similar manner. The main difference here is the fact that the generator room has a louvered opening containing six propeller fans in the wall facing the receiving property. The total area of the opening is $14\,\text{m}^2(151\,\text{ft}^2)$, or approximately 9% of the $156\,\text{m}^2(1680\,\text{ft}^2)$ exterior wall area. From Equation 22-36, the STC-50 partition is reduced to a composite (effective) STC_c rating of

$$\text{STC}_c = -10\log_{10}\left\{\frac{1}{156}\left[14(10)^{-0.2} + 142(10)^{-5.0}\right]\right\} = 12.5\,\text{dBA}$$

$$\text{STC}_c = -10\log_{10}\left\{\frac{1}{1680}\left[151(10)^{-0.2} + 1529(10)^{-5.0}\right]\right\} = 12.5\,\text{dBA}$$

The STC_c rating from Figure 22-23 is 11, a satisfactory check. From Equation 22-34 again, the generator casing noise level at position 3 with both generators running is

$$L_p(3) = 107.4 - NR = 107.4 - (12.5 - 5) = 99.9 \simeq 100\,\text{dBA}$$ $$L_p(3) = 107.4 - NR = 107.4 - (12.5 - 5) = 99.9 \simeq 100\,\text{dBA}$$

Assuming the distance from position 3 to the generators to be 5 m (16.4 ft), the generator casing noise level at position 4 is, from Equation 22-29,

$$L_p(4) = 99.9 + 20\log_{10}(5/65) = 77.6\,\text{dBA}$$ $$L_p(4) = 99.9 + 20\log_{10}(16.4/213) = 77.6\,\text{dBA}$$

The propeller fan noise is evaluated from Figure 22-22. As shown, each of the fans is expected to generate a noise level of 75 dBA at a distance of 1 m (3 ft). Thus, at position 4, each fan yields a noise level of

$$L_p(4) = 75 + 20\log_{10}(1/60) = 39.4\,\text{dBA}$$ $$L_p(4) = 75 + 20\log_{10}(3.28/197) = 39.4\,\text{dBA}$$

The noise level from all six fans (from Equation 22-33) would be

$$L_p = 39.4 + 10\log_{10}(6) = 47\,\text{dBA}$$ $$L_p = 39.4 + 10\log_{10}(6) = 47\,\text{dBA}$$

The manufacturer lists the exhaust stack noise from each unit to be 62 dBA at a distance of 15 m (49 ft) when equipped with the standard residential-grade silencer. Using Equation 22-29, the noise level from one of the silenced exhausts at position 4, which is 63 m (207 ft) distant, would be

$$L_p(4) = 62 + 20\log_{10}(15/63) = 49.5\,\text{dBA}$$ $$L_p(4) = 62 + 20\log_{10}(49/207) = 49.5\,\text{dBA}$$

From Equation 22–33, the total noise level from both stacks would be

$$L_p(4) = 49.5 + 10\log_{10}(2) = 52.5\,\text{dBA}$$ $$L_p(4) = 49.5 + 10\log_{10}(2) = 52.5\,\text{dBA}$$

The total exterior noise level at the residential property from all sources originating from the pump station property is greatest at position 4; from Equation 22–32, it is

$$L_p = 10\log_{10}[10^{7.76} + 10^{4.7} + 10^{5.25}] = 77.6\,\text{dBA}$$ $$L_p = 10\log_{10}[10^{7.76} + 10^{4.7} + 10^{5.25}] = 77.6\,\text{dBA}$$

which is well above the maximum allowable noise standards for residential communities (see Table 22-5).

Potential interior noise reduction measures. Noise levels in both the pump and generator rooms may exceed OSHA standards [20] for 8 h/d employee noise exposure. Assume that on an average day an employee is expected to spend time as follows:

Time, (h)	Place	Noise level, dBA	Allowable time per Table 22-4	D_n from Eq. 22-5
1.0	Pump room	100	2.0	0.5
0.25	Generator room	108	0.66	0.38
6.75	Office	68	No limit	0.00
Total				0.88

The computed D_n value is below the allowable limit of 1.0, and there should be no problem complying with OSHA noise laws provided that all employees spend a considerable amount of

time in the quiet areas. But the anticipated noise levels would make vocal communication difficult between co-workers in the pump room and the generator room.

The generator room has a total surface (floor, ceiling, and walls) of 1027 m^2(11,055 ft^2), and if the ceiling, which has an area of 253.5 m^2(2728 ft^2), were entirely covered with 5-cm- (2-in.-) thick fiberglass insulation board, the reverberant field noise level would be reduced by (from Equation 22-31 and Table 22-10)

$$NR = 10\log_{10}[1 + 10(253/1027)(0.95)] = 5.2\,\text{dBA} \qquad NR = 10\log_{10}[1 + 10(2728/11{,}055)(0.95)] = 5.2\,\text{dBA}$$

This 5-dBA noise reduction would (from Figure 22-3) nearly double the distance for which voice communication is possible. If the pump room ceiling were also covered with fiberglass insulation board, the D_n noise dose would be reduced as follows:

$$NR = 10\log_{10}[1 + 10(13 \times 26/1300)(0.95)] = 5.4\,\text{dBA} \qquad NR = 10\log_{10}[1 + 10(42.7 \times 85.3/14{,}000)(0.95)] = 5.4\,\text{dBA}$$

Time, (h)	Place	Noise level, dBA	Allowable time per Table 22-4	D_n from Eq. 22-5
1	Pump room	95	4.0	0.25
0.25	Generator room	102	1.5	0.17
Total				0.42

Example 22-7
Reducing Exterior Noise from a Pumping Station

Problem: The pumping station of Example 22-6 must meet a community noise ordinance that limits the maximum noise level at adjacent residential property to 55 dBA. Find and evaluate the modifications necessary to comply with the local ordinance.

Solution: All three major noise sources (generator casing, engine exhaust, and fan noise) must be reduced. From Example 22-6, the noise level at position 3 (Figure 22-26) emanating from the pump room is only 39 dBA, so further reduction of the noise from this source is not necessary.

The following noise levels at position 4 are obtained from Example 22-6, for which there are no special noise control features:

Generator casing:	77.6 dBA
Propeller fans:	47.0 dBA
Generator exhaust:	52.5 dBA
Total noise level:	77.6 dBA

To meet the maximum noise level of 55 dBA requires a reduction of 23 dB. Try STC-30 sound traps in the louver. The pressure drop for these 1.5-m- (5-ft-) long traps is about 0.37 mm Hg (0.20 in. WC) at 5.08 m/s (1000 ft/min). To reduce this pressure drop to the recommended 0.09 mm Hg (0.05 in. WC), the velocity must be reduced. Because the pressure drop is proportional to the square of the velocity,

$$\frac{\text{new pressure drop}}{\text{original pressure drop}} = \left(\frac{\text{new velocity}}{\text{original velocity}}\right)^2$$

SI Units	**U.S. Customary Units**
$\dfrac{0.09\,\text{mm Hg}}{0.37\,\text{mm } Hg} = \left(\dfrac{v}{5.08\,\text{m/s}}\right)^2$	$\dfrac{0.05\,\text{in.}}{0.20\,\text{in.}} = \left(\dfrac{v}{1000\,\text{ft/min}}\right)^2$
from which $v = 2.51$ m/s	from which $v = 500$ ft/min

The original 14-m^2 (151-ft^2) louver sized for a face velocity of 3.81 m/s (750 ft/min) must be enlarged to

$A = 14\,(3.81/2.51) = 21\,\text{m}^2$	$A = 151(750/500) = 226\,\text{ft}^2$

The net wall area is

$A = (19.5\,\text{m})(8\,\text{m}) - 21\,\text{m}^2 = 135\,\text{m}^2$	$A = (64\,\text{ft})(26.25\,\text{ft}) - 226\,\text{ft}^2 = 1454\,\text{ft}^2$

From Equation 22-36,

$\text{STC}_c = -\,10\log_{10}[(1/156)(21\times10^{-30/10} + 135\times10^{-50/10})] = 38.4\,\text{dB}$	$\text{STC}_c = -\,10\log_{10}[(1/1680)(226^{-30/10} + 1454^{-50/10})] = 38.4\,\text{dB}$

Recompute the noise level at position 4. Noise from the generator casing is reduced by an STC$_c$ 38.4-dB barrier instead of the original STC$_c$ 12.5-dB wall (without sound traps). The net noise reduction is 38.4 – 12.5 = 25.9 dB.

The propeller fan noise is reduced by the STC rating of the sound traps (30 dB) minus the STC rating of the louver (2 dB) for a net noise reduction of 28 dB. The components at position 4 are, therefore,

Generator casing (77.6 – 25.9)	= 51.7 dBA
Propeller fans (47 – 28)	= 19 dBA
Generator exhaust	= 52.5 dBA

and the total noise from Equation 22-32 is

$L_p = 10\log_{10}[10^{51.7/10} + 10^{19/10} + 10^{52.5/10}] = 55.1\,\text{dBA}$	$L_p = 10\log_{10}[10^{51.7/10} + 10^{19/10} + 10^{52.5/10}] = 55.1\,\text{dBA}$

This sound level is just slightly above the 55-dBA limit. To make further reductions, choose critical-grade exhaust silencers for an additional 5-dB noise reduction on the exhaust noise. Note that both engines must be silenced with critical-grade silencers to achieve this result.

With the critical-grade exhaust silencers and the STC-30 sound traps, the noise level at position 4 is

Generator casing:	51.7 dBA
Propeller fans:	19.0 dBA
Generator exhaust:	47.5 dBA
Total noise level:	53.1 dBA (from Equation 22-32)

This meets the noise ordinance with a small margin of safety. For an increased margin of safety, STC-35 sound traps could be used for the louver opening. This application would require sound traps that are 2.1 m (7 ft) long, which would require considerable floor space—one of the several reasons for avoiding retrofit in favor of planning for noise reduction during the first stages of layout.

22-15. References

1. Karassik, I. J., J. P. Messina, P. Cooper, and C.C. Heald, *Pump Handbook*, 3rd ed., McGraw-Hill, New York (2001).
2. Reichert, F. E., J. R. Hall, and R. D. Woods, *Vibrations of Soils and Foundations*, Prentice-Hall, Englewood Cliffs, NJ (1970).
3. Corbo, M. A. and S. B. Malanoski, "Pump rotordynamics made simple," *Pumping Technology* (June 1998).

4. Marscher, W. D., "Determination of pump rotor critical speeds during operation through use of modal analysis." *Proceedings ASME 1986 WAM Symposium on Troubleshooting Methods and Technology*, Anaheim, CA (December 1986).
5. Blevins, R. D., *Formulas for Natural Frequency and Mode Shape,* Robert Krieger Publishing Co., Malabar, FL (1984).
6. Corbo, M. A. and S. B. Malanoski, "Practical design against torsional vibration," *Proceedings of the Twenty-Fifth Turbomachinery Symposium* sponsored by the Turbomachinery Laboratory, The Texas A&M University System (September 1996).
7. Marscher, W. D., "The effect of variable frequency drives on vibration problems in vertical pumps," *Proceedings of the Water & Wastewater 1990 Conference,* Barcelona, Spain (April 24, 1990). Published by Walpole Productions, Houston, TX.
8. Rathbone, T., "Vibration tolerance," *Power Plant Engineering,* 43:721–724 (November 1939).
9. Blake, M., "New vibration standards for maintenance," *Hydrocarbon Processing and Petroleum Refining,* 43: 111–114 (January 1964).
10. Baxter, R. L., and D. L. Bernard, "Vibration tolerances for industry," American Society of Mechanical Engineers, 67-PET-14:1–8 (1967).
11. Hancock, W. P., "How to control pump vibration," *Hydrocarbon Processing,* 53:107–113 (March 1974).
12. Marscher, W. D., "The relationship between pump rotor system tribology and appropriate vibration specifications for centrifugal pumps," Proceedings Institute of Mechanical Engineers (London), IMechE 3rd European Congress on Fluid Machinery for the Oil and Petrochemical Industries, The Hague, Netherlands (May 1987).
13. Wachel, J. C., and C. L. Bates, "Techniques for controlling piping vibration and failures," 76-Pet-18, American Society of Mechanical Engineers, New York (1976).
14. Marscher, W. D., "How to use impact testing to solve pump vibration problems," *Proceedings Electrical Power Research Institute Power Plant Pumps Symposium*, Tampa, FL (June 1991).
15. Dodd, V. R., *Total Alignment,* The Petroleum Publishing Co., Tulsa, OK (1974).
16. Agostinelli, A., D. Nobles, and C. R. Mockridge, "An experimental investigation of radial thrust in centrifugal pumps," *Journal Engineering for Power Transmission,* American Society of Mechanical Engineers 82:120–126 (1960).
17. Black, H. F., "Effects of fluid-filled clearance spaces on centrifugal pump vibrations," 8th Turbomachinery Symposium, Texas A&M University (1979).
18. Marscher, W. D., "Analysis and test of multistage pump at critical speeds," Society of Tribologists and Lubrication Engineers/American Society of Mechanical Engineers Joint Tribology Conference, Ft. Lauderdale, FL (October 1989).
19. Harris, C. M., and C. E. Crede, *Shock and Vibration Handbook,* 2nd ed., McGraw-Hill, New York (1976).
20. OSHA, "Safety and health standards for federal supply contracts," (Walsh Healey Public Contracts Act, U.S. Department of Labor), *Federal Register*, 34, 7948, Washington, DC (1969).
21. Miller, J. D., "Effects of noise on people," *Journal of the Acoustical Society of America,* 56: 746 (1974).
22. M. W. Kellogg Company, *Design of Piping Systems,* John Wiley, New York (1956).
23. Streeter, V. L., and E. B. Wylie, *Fluid Transients,* FEB Press, Ann Arbor, MI (1982).
24. *ASHRAE Handbook, Systems Volume,* American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, GA (1984).
25. Harris, C. M. (Ed.)., *Handbook of Noise Control,* McGraw-Hill, New York (1979).
26. Beranek, L. L. (Ed.), *Noise and Vibration Control,* McGraw-Hill, New York (1971).

Chapter 23

Heating, Ventilating, and Cooling

EARL L. HECKMAN
PERRY L. SCHAFER
PHILIP WOLSTENHOLME

CONTRIBUTORS
Richard E. Pustorino
Ashok Varma

The needs, criteria, and design procedures for ventilating, heating, and cooling pumping stations are discussed in this chapter. Worked examples are used to illustrate the principles presented. Because these examples are drawn from practice, only U.S. customary units are used. The design of air conditioning systems is complex enough to require a specialist, so the discussion of air conditioning is limited to elementary considerations.

References to codes and standards are given in abbreviated form—code letters and numbers only (such as ASHRAE Standard 90-80).

23-1. Need for Heating, Ventilating, and Air Conditioning

Heating and ventilating (H&V) with either air conditioning or cooling (HVAC) are included in pumping stations for the following reasons:

- To facilitate the safe and efficient performance of operating and maintenance personnel
- To minimize the deterioration of the equipment, controls, and structure
- To promote community acceptance of the station by helping to control noise and odor emissions.

Spaces within a pumping station are divided into two broad categories:

- The "wet well" or "sump" is a pump intake basin containing the fluid. In wastewater pumping stations, the term often includes wastewater channels, grit and screen rooms, and the pump suction chamber. If the fluid is potable water, it is also called a "clear well," "forebay," or "suction well."
- The "dry well" or pump room is a term also loosely used to include ancillary spaces for pumps, motors, and auxiliary equipment.

Hazardous Environments

Hazardous environments can be defined differently in different codes. In general, it is any environment that can be dangerous because of the possible presence of toxic, flammable, or explosive gases or liquids. As defined by NFPA 820, it is an environment that could contain explosive gases. Toxic gas is not covered. Regarding pumping stations, NFPA 70 classifies hazardous environments for electrical equipment as:

- Class 1, Group D, Division 1 where there is a *high likelihood* of an explosive hazard. Group D environment is one where hydrocarbon vapors may be present. There are about a dozen hazardous environment groups. All equipment in Group D must be explosion-proof—an expensive requirement for polyphase motors.

- Class 1, Group D, Division 2 where there is the *possibility* of an explosive hazard. Only single-phase equipment (switches, lights, motors) must be explosion-proof.

A Division 1 classification is assigned to a wet well that is either not ventilated or only intermittently ventilated in accordance with NFPA 820 criteria. If the pump intake basin is well-ventilated at all times, the likelihood of an explosive atmosphere is greatly reduced and the space could be reclassified as Division 2.

Personnel Safety

There have been too many incidents of death and injury to ignore good practice for workers entering any room containing wastewater or the possibility that sewer gases could enter a room. Some jurisdictions require permanently installed meters for explosive gas (hydrocarbons), hydrogen sulfide, and low oxygen levels. However, the chance that such equipment will not be calibrated with adequate frequency or that detectors will become fouled with hydrogen sulfide poses a risk. It is better that workers be equipped with portable monitors regularly calibrated and tested. The monitors are expensive, but there is no good substitute for the protection they offer.

Confined Spaces

OSHA defines confined spaces as follows:

- A "confined space" is an area that is large enough and so configured that an employee can bodily enter and perform assigned work, has limited or restricted means for entry or exit [e.g., tanks, vessels, silos, storage bins, hoppers, vaults, trenches greater than 1.2 m (4 ft) in depth, and pits] and is not designed for continuous occupancy.
- A "permit required confined space" is defined as a confined space with one or more of the following characteristics: (1) contains or has the potential to contain a hazardous atmosphere such as lack of oxygen (less than 19.5%), or explosive or toxic gases; (2) contains a material that has the potential for engulfing an entrant; (3) has an internal configuration such that an entrant could be trapped or asphyxiated by inwardly converging walls or a floor that slopes downward and tapers to a smaller section; or (4) contains any other recognized serious safety or health hazard.

Merely providing ventilation per codes, such as NFPA 820 or the Ten-State Standards, does not by itself change the classification of a confined space to a nonconfined space. The criteria regarding entry, exit, and continuous occupancy must also be addressed. In practical terms, it would seem that the following features must be present in a wastewater pumping station wet well to avoid classification as a confined space:

- Stairway access. Ladder access probably would not be considered to meet the criteria of eliminating "limited or restricted means for entry or exit."
- Permanently installed, continuous ventilation sufficient to control the accumulation of any hazardous (toxic or explosive) gases and prevent oxygen depletion below concentrations necessary to support life.
- Permanently installed lighting.
- Permanently installed or (if always used) portable detectors for combustible gas, hydrogen sulfide, and oxygen concentrations.

Even with the above features, there are some sanitation agencies that do not allow individuals to enter wet wells alone. They must always be accompanied by an observer who does not go with them into the wet well.

Hazards in Wet Wells

Proper ventilation of pumping stations is an often misunderstood and neglected subject. Enclosures below grade, such as wet wells and vaults, are considered by OSHA to be confined spaces, and there are stringent requirements for access to them and elaborate protective measures for persons entering them. Many deaths in both water and wastewater pumping stations could have been avoided with good ventilation. Although ventilation by itself does not change the classification of a "confined space" to a "nonconfined space," good ventilation goes a long way in improving the safety of wastewater pumping stations. In an accident in England, methane gas leaked from the forebay into a water pumping station and collected over a weekend. The station was not continuously ventilated. When a party of visitors entered the station, a spark from the lighting system caused an explosion that killed 14 people. In another incident, an operator on his normal rounds breathed a fatal concentration of hydrogen sulfide gas in a wet well inadequately ventilated by an inappropriate design. In subsequent air sampling, hydrogen sulfide concentrations up to 20 times greater than the recommended

limit were found. These examples, along with many others, underscore the need for proper ventilation design. In a matter of life and death, there should be no compromise to save costs. Decision makers and designers should be well aware of the hazards (see especially the 1979 and 1984 NIOSH reports [1, 2]) and the methods to overcome them.

Good ventilation is also important in maintaining a benign, dry environment for mechanical and electrical equipment. The cost of the ventilating equipment may be partially or completely offset by the reduced deterioration of equipment and improved efficiency of personnel due to better working conditions. Without ventilation, explosive gases (from illegally dumped flammable liquids) may accumulate in wastewater wet wells and result in devastating explosions. The formation of explosive concentrations of methane through bacterial action is also possible.

Bacterial growth under anaerobic conditions also can produce hydrogen sulfide, which is more poisonous than cyanide. At trace concentrations it has the smell of rotten eggs, but at high and dangerous concentrations the gas overwhelms the olfactory senses. Because it then cannot be smelled, it is the more insidious and deadly. Victims overcome but not killed outright often suffer irreparable brain damage. The depletion of oxygen in an enclosed space is also a possible danger.

Hazardous conditions in the clear wells of water pumping stations are less frequent than in the wet wells of wastewater stations, but they nevertheless exist (as cited at the beginning of this section). Ventilation is necessary, and heating may be required to protect equipment from corrosion and freezing.

Summary of Good Ventilation Practice

- Intermittent ventilation systems are inherently hazardous and may be deadly, because workers will not wait for the lengthy scavenging period even of high-speed fans. For example, at 60 air changes per hour (AC/h), it takes two hours to reduce 2000 ppm H_2S to 50 ppm at 100% efficiency, because an air change only dilutes the air and does not actually change it.
- NFPA 820 recommends 12 AC/h continuously for wastewater wet wells and 6 to 10 AC/h continuously for pump rooms.
- Some states require considerably greater ventilation rates—up to 30 AC/h.
- Prudent engineers design for 20 to 45 AC/h in wet wells and 10 AC/h in dry wells.
- Continuously force air in at the ceiling through industrial diffusers, and force air out at the floor with fans at both inlets and outlets.
- Supply enough air to cool the motors.
- Use either stainless-steel or fiberglass fans and shafts with permanently lubricated bearings.
- Use fiberglass or aluminum ducts.
- Use run-around circuits and heat exchangers to recover heat.

An Alternative to Ventilation

Neither OSHA nor NFPA 820 requires ventilation in wet wells. If ventilation is not provided, however, the wet well must be classified as a "permit required confined space" per OSHA and as Class I, Group D, Division 1 location per NFPA 820. Entry into such spaces requires:

- A permit
- Atmospheric testing, purging, and monitoring for oxygen, flammability, and toxicity
- Personnel training
- Established procedures for working, standby, and rescue
- Safety equipment and clothing to protect the head, body, hands, and feet plus safety harnesses and lifelines
- Self-contained breathing apparatus
- Record keeping
- Competent persons at the site as observers and supervisors.

Augment these requirements by consulting the literature [2] and local authorities. Include the required safety procedures and measures in the operating and maintenance (O&M) manual.

Temperature, Humidity, and Condensation

The life expectancy of equipment and controls decreases at excessive temperatures and humidities. Most electrical gear and electronics are rated for use in surrounding air temperatures up to 40°C (104°F). High humidity increases the likelihood of moisture condensation, which occurs when air is cooled below its dew point; rust, mildew, and deterioration of electrical and thermal insulation follow, particularly if corrosive gases are present. Blowing warm air over cooler pipe and wall surfaces can raise their surface temperature sufficiently to prevent condensation, thus preserving them.

Dry Wells

Heating and ventilating requirements for pump rooms and dry wells are similar for both water pumping stations and wastewater pumping stations. Maintaining a reasonable minimum temperature in cold weather and removing excess heat in warm weather are the principal reasons for ventilating a dry well or pump room. The following are sources of unintentional heat gain:

- Motors of pumps and other equipment
- Variable-speed drives and inverters
- Motor controls
- Solar gain and lights
- Personnel (negligible).

The sum of unintentional heat gains often entirely offsets building heat losses due to transmission through the structure and infiltration of colder outside air. Any remaining net heat loss must be made up by additional heat supplied to the space. In cold climates, building insulation and double glazing can be used to minimize structure heat losses and reduce condensation.

Heat gain in pumping stations usually exceeds their heat loss in moderate weather. The excess heat must be removed by ventilating with cooler outdoor air; if the ambient air temperature is too high, it must be removed by (1) direct evaporative cooling, (2) indirect evaporative cooling, or (3) refrigeration. Direct evaporative cooling is generally ineffective if the relative humidity of outdoor air exceeds about 75%. Even then, the nearly saturated air supply may promote rusting. Spaces with either high heat gains or a noise level that must be contained may justify the higher costs of refrigerated cooling or the use of some of the pumped water as the heat transfer medium in an air cooling system.

Aesthetics

Pumping stations, especially those in residential neighborhoods, must be acceptable to the community. Acceptability may require (1) noise control, (2) a building in harmony with other structures in the area, and (3) for stations pumping wastewater, control of odors. The superstructure of some stations has been designed to resemble a residence. Zoning ordinances may affect station height, setback, and appearance. Local noise ordinances may require sound attenuation at openings in the building envelope, particularly if an emergency generator is installed (see Chapter 22 for a discussion of noise control).

HVAC Complexity Related to Station Type and Location

The required extent of HVAC, its controls, and its cost increase when:

- Pumping capacity and station size increase
- Auxiliary spaces are added
- Both heating and cooling are required
- Distance to a residential area decreases
- Variable-speed pumps are used
- Stations are automated
- Continuous attendance is required.

23-2. HVAC Design Criteria

Codes and guidelines for good engineering practice are presented in this section, with emphasis on safety for workers.

Codes, Ordinances, and Standards

Confer with the jurisdictional authorities to learn which codes, ordinances, or adopted standards apply at the site. Examples of documents that may govern pumping station design include:

- City ordinances
- State or local plumbing codes
- State or local fire codes
- State or local building construction codes
- State or local mechanical codes
- State or local energy codes
- Federal or state EPA regulations
- Occupational Safety and Health Act (OSHA) [3]
- National Electric Code (NEC)
- National Fire Protection Association (NFPA) codes and standards
- Ten-State Standards [4]
- ASHRAE 90.1-2001 energy code, which includes recommended heat-transfer coefficients for buildings and rules for energy expenditure and recovery.

Codes and standards, whether mandatory or not, generally represent minimum requirements. Good engineering practice and safety considerations often require adherence to more stringent criteria.

Wet Well Design Guidelines

There is a profound distinction between wastewater wet wells designed for frequent entry (accessible) and those designed for nonentry (sealed).

Accessible Wastewater Wet Wells

Wastewater wet wells containing either bar screens or mechanical equipment must be accessible to workers for maintenance and servicing, so they must be ventilated mechanically. Continuous ventilation with air forced in and forced out is the best safeguard against the buildup of hazardous gases. Section 32.7 of the Ten-State Standards [4], however, only requires forcing the air into the wet well; multiple inlets and outlets for wet wells over 5 m (15 ft) deep are also recommended. Of course, interconnections between wet well and dry well ventilating systems are not allowed.

Fan wheels and shaft seals should be rated nonsparking in accordance with standards of the Air Movement and Control Association. Electric motors, wiring, controls, and monitors must meet NEC requirements for Class 1, Group D, Division 1 hazardous areas. Both automatic heating and dehumidification should be considered for worker comfort and safety as well as for protection from corrosion.

According to the Ten-State Standards, ventilation can be either

- continuous with 12 complete air changes per hour (based on wet well volume above high water level), or
- intermittent with 30 air changes per hour with the high-speed fan ventilation switch interlocked to the wet pit lighting system. However, intermittent ventilation is prohibited by NFPA 820 if the space is to be a Division 2 location.

Ventilation at a low rate can be automatically increased by gas sensors that (1) detect the presence of combustible gases or hydrogen sulfide, and (2) either increase fan speed or start an auxiliary fan. (The incremental airflow is normally unheated.) Methane detectors are usually set for 20% or less of the lower explosive limit (LEL), which is the lowest concentration of a combustible gas in air at which an explosion can occur (approximately 5% by volume for methane). Hydrogen sulfide gas detectors can be set for 10 ppm, which is considered a safe level for 8-h exposure [5]. If such detectors are to be dependable, they must be recalibrated and tested at least monthly, and this maintenance should be specified in the O&M manual. Consider the possibility of intermittent ventilation design giving a false sense of security because maintenance may become unreliable and sensors may fail. Personnel should always carry portable detectors when entering potentially hazardous enclosures.

Good ventilation is not easily achieved because the purging of gases (which may be heavier than air) is by dilution and not by a "clean sweep," as is suggested by the phrase "air changes." Thus, even with the best supply air distribution, the purging of dangerous gases from odd-shaped areas is far from perfect. Opening the door to a wet well ventilated only with forced air inflow with gravity relief defeats the ventilation system because the airflow escapes through the door. The best practice for hazardous spaces, such as wastewater wet wells, is to blow air into the chamber at or near the ceiling and to exhaust air at or near the lowest level at a rate of approximately 5% more than the air intake rate, thus producing a partial vacuum of 30 to 60 Pa ($\frac{1}{8}$ to $\frac{1}{4}$ in.) of water column (WC). Even though permitted by some codes, such practices as providing air supply or exhaust alone at a continuous, but minimum, airflow rate that is to be switched to a high rate only at the time of entry should be avoided. It compromises safety for a relatively small reduction of cost. Minimum or intermittent air changes may not prevent explosions and do not adequately protect impatient workers who may not wait for the required period of high-speed scavenging.

According to NFPA 820, the *minimum* recommendations for ventilation are as follows:

- All ventilated spaces are to be served by both supply and exhaust fans and powered from two independent sources to ensure operation during power failure of a single source.
- Continuous ventilation at the rate of 12 air changes per hour with combustible gas detectors is required. A two-speed ventilation system is recommended, with high-speed operation initiated at the warning level of gas concentration.
- Equipment rooms and other spaces below grade containing gas piping are to be (1) ventilated at the rate of 12 air changes per hour, and (2) equipped with combustible gas detectors and, preferably, two-speed fans.
- Galleries and tunnels are to be treated the same as below-grade spaces with a 0.38 m/s (74 ft/min) air velocity.
- Below-grade spaces without gas piping are to be ventilated at the rate of 10 air changes per hour; galleries are to be ventilated at that rate or by an air velocity of 0.25 m/s (50 ft/min), whichever is greater.

The airflow pattern can be controlled only by using a combination of supply and exhaust ducts. Without ducts, air follows the path of least resistance

Table 23-1. Air Velocities and Friction Losses in Duct Design

Item	Air velocity		Friction loss	
	m/s	ft/min	mm WC/100 m	in. WC/100 ft
Supply duct	5.1–9.1	1000–1800	6.7–10	0.08–0.125
Exhaust duct	4.1–7.6	800–1500	5.0–8.3	0.06–0.10
Registers, grilles	3.0–5.1	600–1000	4.2–6.7	0.05–0.08 net
Intake louvers	1.3–2.0	250–400	2.5–5.0	0.03–0.06 net

and leaves stagnant areas. Suggested duct design velocity and friction ranges are given in Table 23-1. Ducts must be routed to clear equipment access and removal space, hoist rails, and hoistways. The air quantity supplied should be based either on the recommended air change rate needed or on the required heat removal, whichever is larger. Outside ventilation air should be filtered for cleanliness and insect control. Arrange insect screens for easy removal for cleaning. In the O&M manual, alert maintenance workers to the need for filter replacement. Consider installing a filter gauge to show the pressure drop across the filter or, preferably, use a differential pressure switch to energize a "clogged filter" signal at the control panel. In cold climates, select a storm louver type and net air velocity to prevent the entry of snow; and provide heat at louvers to keep them from being clogged with ice.

Sealed Wet Wells

"Sealed" (not easily entered) wet wells require an adequate vent (perhaps only a manhole cover) to accommodate air displacement due to changes in the liquid level. Except in very cold climates, heating or heat tracing is rarely required because the temperature of the moving water is sufficient to prevent freezing. However, if the water temperature is near freezing (especially if it contains frazil ice) and the pumps run only intermittently, some form of heating is needed. Explosion-proof electrical equipment is required for sealed wastewater wet wells, just as it is for accessible wet wells.

Heat

If no equipment is located in a wet well, heating the airstream is frequently omitted, which allows the wet well temperature to reach an equilibrium between the water temperature and outdoor air temperature. Pipes that are filled with motionless water for long periods and exposed to subfreezing air can be "traced" (wrapped) with electric heat tape beneath their insulation to prevent freezing. Panel heating of stairs and walkways can be used to prevent ice formation in cold climates. Wet wells with screens or other equipment that require inspection and maintenance should be heated to approximately 10°C (50°F) for worker comfort, safety, and efficiency as well as for corrosion protection. Any heat source open to wastewater wet well air must not have either an open flame or a temperature above 260°C (500°F) to prevent the ignition of combustible gases.

Explosion-proofing

All electrical equipment (including motors) in or open to the wet well must be explosion-proof and should be located above the flood level. Even submersible pump motors should be explosion-proof because they may not always be submerged. Their control panels should not be in the wet well but in a nonhazardous location. Some engineers recommend explosion-proof equipment regardless of location.

23-3. Odor Control

Odors at wastewater pumping stations constitute one of the worst difficulties that agencies encounter. If not resolved quickly, they can take inordinate amounts of staff time in dealing with irate neighbors. A more advisable and cost-effective approach is to conduct appropriate evaluation of any potential odor problems during planning and design of the pumping station, and to install all necessary odor control measures along with the construction of the pumping station itself. There are sufficient valid engineering and scientific tools available today to allow fully workable solutions to be determined during the project design.

Odor Control Evaluations

Odor control evaluations need to be conducted within the broadest context possible. Too many en-

gineers believe odor control is synonymous with "foul air treatment." Actually, foul air treatment is often the most costly type of odor control and should be avoided unless absolutely necessary. Other types or categories of odor control should normally be evaluated first to determine if foul air treatment can be avoided.

Considerable information is needed to conduct a proper odor control evaluation, and information about the wastewater entering the pumping station is crucial. The details of the upstream collection system (including operation of other pumping stations); the sources, kinds, and amounts of wastewater; industrial contributions; and other information are all vital. Specific sampling and testing of existing wastewater flows in the same area as the future pumping station are likely to be helpful. Several chapters in WEF Manual 22 [6] are helpful in conducting such an evaluation. See also ASCE Manual 69 [7].

Four odor control strategies are defined and discussed herein. In order of effectiveness, they are:

- Minimizing or preventing production of odorous compounds
- Treating odorous compounds within the liquid phase
- Containing and treating foul air
- Enhancing atmospheric dispersion of foul air.

A well-organized evaluation of these categories of odor control almost always results in a successful project. Operation and maintenance costs as well as first cost must be evaluated for each strategy. Odorous substances include a large variety of compounds. The reduced sulfur family of compounds is the major problem in most wastewater systems, and hydrogen sulfide is the most common offender. Various effects and standards relating to hydrogen sulfide concentrations are described in Table 23-2. But other sulfides, disulfides, and mercaptans are also frequent problem compounds because their odor thresholds are almost all in the part-per-billion range or less. Reduced sulfur compounds, amines, aldehydes, ketones, and various organic acids can cause problems. Ammonia is only rarely a problem because its concentration is typically low compared with its odor threshold. The concentration of odorants can be measured in the liquid phase as well as the gas phase for almost all potential odorous compounds. Liquid-phase analysis is often easier to complete and is more accurate, whereas gas phase testing can be costly, especially when scanning for large numbers of potentially odorous organic compounds.

Minimizing Odorous Compounds

The first line of defense against odor problems is to design the entire system to produce the absolute minimum quantity of odorous compounds allowed to enter the pumping station via the influent water. Upstream controls may be beyond the purview of the pumping station designer, but they may need to be explored because it could be less costly to solve the problem upstream. Control measures could include the following:

- Further pretreatment of specific industrial discharges to the system.

Table 23-2. Effects and Standards of Hydrogen Sulfide Gas in the Atmosphere[a]

Concentration, ppm by volume	Effect
0.0005	Olfactory detection threshold
0.003	Max concentration for electronic equipment per ISA
0.002–0.008	Practical odor threshold range
0.010	Max concentration for electrical equipment per NEMA
0.030	Ambient Air Quality (odor-based) Standard in California
1	Offensive odor; rotten egg smell
5	Deadens olfactory senses
10	Max 24-h exposure per OSHA
15	Max 8-h exposure per OSHA
10–50	Headache, nausea, and eye, nose, and throat irritation
50	Max 30-min exposure per OSHA
50–300	Eye and respiratory injury
300–500	Life threatening; pulmonary edema
700	Immediate death for everyone
46,000	Lower explosive limit

[a]Adapted from ASCE Manual 69 [7] and industry standards.

- Minimizing slug loads of wastewater from industries or other point sources.
- If hydrogen sulfide is the significant problem, keeping wastewater pH well above 7 to minimize hydrogen sulfide off-gassing. A pH of 8 would usually be adequate, but pH 9 might be required sometimes.
- Designing upstream sewers to maintain aerobic conditions in the wastewater—usually by keeping velocities above 1.5 m/s (5 ft/s).

At the pumping station itself, there should be minimum turbulence of wastewater because turbulence promotes off-gassing of odorous compounds. Drop inlets into the wet well can and should be avoided. In stations with constant-speed pumps, use a sloping approach pipe with its invert at or slightly below the low water level (even though its crown may be submerged at the high water level; see Section 12-7). However, if the inlet pipe crown remains submerged for an extended period, foul air will be trapped in the influent pipe, and the foul air will be forced out of manholes upstream from the pumping station. Variable-speed pumping is highly desirable (especially in larger stations where odor problems are more likely and the investment more easily justified), because matching water elevations in the sewer and wet well allows smooth, nonturbulent entry into the wet well. Refer to Sections 12-6 and 12-7 for design details.

Wet wells should be kept small enough to minimize stagnation and the settling of solids. These deposits are anaerobic and produce odorous compounds that diffuse into the liquid above and thence into the air. Slime layers form on submerged walls of wet wells and also produce odors. Such wall areas should also be minimized. Wet wells should be frequently cleaned (say, weekly). Refer to Section 12-7.

The velocities needed to keep domestic wastewater aerobic, promote scour, and eliminate odor-producing deposits in pipes are given in ASCE Manual 69 [7]. In general, force main velocities of 1.1 to 1.2 m/s (3.5 to 4.0 ft/s) occurring at least once per day are advisable (depending on pipe diameter) to minimize problems.

Treating Odorous Compounds in the Liquid Phase

There is a host of chemicals that can be added to wastewater to inhibit or treat odorous compounds, thus minimizing off-gassing and subsequent odor problems. Only the broad categories of such chemicals are defined here:

- Oxidants, such as chlorine, sodium hypochlorite, and hydrogen peroxide. These chemicals oxidize sulfide (and perhaps other compounds) that already exist in solution, and they minimize additional generation for a limited time downstream.
- Precipitants (such as ferrous or ferric chloride) that precipitate sulfide as insoluble, dark-colored compounds. Reactions usually take 15 to 20 min.
- Inhibitors, such as high-pH slugs (pH greater than 12 for 20 min) or anthroquinone slugs, greatly inhibit sulfate-reducing bacteria densities for a one- to two-week period. Continuous addition of nitrate compounds is sometimes advantageous to promote nitrate reduction and minimize sulfate reduction.
- Aerobic conditioning by the addition of air or high-purity oxygen. If more than about 0.5 to 1.0 mg/L of dissolved oxygen is kept in the wastewater, little anaerobic activity can take place to form the reduced, offensive compounds. In gravity sewers it is difficult to add dissolved oxygen except by natural re-aeration of wastewater flowing at velocities in excess of 1 m/s (3.3 ft/s). In some force mains (with certain rising profiles, sufficient pressure, and compatible pipe materials), it is feasible and cost-effective to add high-purity oxygen.
- Bases such as lime or caustic added continuously to keep pH above 8.5 minimizes hydrogen sulfide production and off-gassing.

Chemicals, their reaction times, and transit times through the facilities must be evaluated carefully [6, 7, 8]. Chemicals can be added far upstream, at the wet well, or at the discharge of the force main.

Containing and Treating Foul Air

The first step in designing a foul air treatment system is to develop a reliable containment and ventilation system that brings all foul air to the treatment device. Containment of foul air is not always easy, with sewers bringing gases into the wet well and doors or hatches being opened. But regulating fans on both inlet and exhaust to maintain the slight vacuum described in Section 23-2 helps, as does proper dispersal of incoming air at the ceiling and collection of foul air at the floor.

Various ventilation, corrosion, safety, and related issues are discussed separately in this chapter. A large variation in pollutant concentration over the course of the day sometimes results in poor treatment unless the system is specifically designed for it. Therefore,

details about upstream system characteristics and operation are critical.

Biological Foul Air Treatment

Of the various types of biological foul air treatment systems, the one most applicable to pumping stations is bulk media biofiltration. A typical bulk-media biological filter reduces odors by passing foul air through a mixture of constituents including compost, bark, soil, wood chips, peat, carbon, and other media. Odorous compounds are removed through a biological oxidation process, facilitated by microorganisms attached to the media. The main products of the oxidation process are carbon dioxide and water, with other products including salts, acids, and a biomass. Hydrogen sulfide in the foul air produces sulfuric acid and causes the media to become acidic, so underdrains and leachate piping must be corrosion-resistant. Maintenance costs are relatively low, although the media must be changed from time to time. The exhausted media is not classified as a hazardous material. For further information, see the literature [6, 7, 8].

Biofilters can be constructed, or purchased as a vendor package. A constructed biofilter consists of perforated pipes installed in gravel or other dispersion layers and overtopped with a layer of the media and is designed to treat $0.3\,m^3/min$ of gas per m^2 ($1\,ft^3/min$ of gas per ft^2) of media surface area, so the area requirements can be significant. The moisture content of the media is critical for the biomass, and the retention time of the foul air in the media must be sufficient—usually 30 to 120 s of empty bed contact time. To sustain the appropriate level of biomass, humidification of the inlet air or irrigation of the media is required. With proper design and operation, biofilters can provide good removal of a wide variety of odorous compounds. However, a biofilter has limited ability to remove high concentrations of hydrogen sulfide and organics simultaneously. In foul air streams with high concentrations of hydrogen sulfide that also contain organics, it may be necessary to add a polishing stage to remove the organics to sufficiently low levels ($\ll 0.1$ ppm).

Activated Carbon

Activated carbon units are used as the primary odor control elements for foul streams containing low levels of organics and hydrogen sulfide, particularly at remote locations. When the adsorptive material becomes exhausted, it must be either regenerated or replaced. Regenerable media can be regenerated in-place by flooding the vessel with water; however, in-place regeneration of activated carbon has proven difficult and the adsorptive capacity of the media is not fully recovered upon regeneration. Disposal of spent material has been a problem at some locations.

There are several types of granular activated carbon commercially available, including virgin, caustic impregnated, and catalytic carbon.

Virgin Carbon

If the concentrations of odorous compounds are low, the foul air can be effectively treated with virgin activated carbon. However, when concentrations of odorous compounds, particularly hydrogen sulfide, are greater than several ppm, large amounts of carbon could be required and quickly exhausted, which creates substantial operating and maintenance costs. Spent virgin carbon is usually nonhazardous and can be disposed of at most landfills. At pumping stations where the foul air has a high humidity ($> 50\%$), virgin carbon provides adequate hydrogen sulfide removal, but removal of organics may be inhibited due to water adsorption.

Caustic Impregnated Carbon

Granular activated carbon can be treated with a caustic solution to produce increased hydrogen sulfide capacity. However, caustic-impregnated carbon has a potential for bed fires. Bed fires are caused due to the low ignition temperature of the carbon and the natural exothermic adsorption process. Due to the threat of bed fires, impregnated carbon is not recommended for unattended locations. The disposal of the spent media can be more costly than the disposal of virgin carbon due to its low pH (caused by its increased hydrogen sulfide content).

Catalytic Carbon

There are several types of catalytic carbon available, such as U.S. Filter's Midas™ and Calgon's Centaur™. This catalytic carbon has the same benefits as virgin carbon in adsorbing both volatile organic compounds (VOCs), and reduced sulfur odorants, but offers a higher hydrogen sulfide adsorption capacity than virgin carbon. Catalytic carbon is used to treat air streams with moderate hydrogen sulfide and organics. Although it may be several times as expensive as virgin carbon, catalytic carbon is recommended for

remote areas or in hard-to-access areas, because the media need not be changed as frequently.

Wet Chemical Scrubbing

Wet chemical scrubbing is in widespread use at wastewater treatment facilities for odor control, and is space-conserving for large foul air flow rates. The scrubbing solution is maintained at a high pH with caustic, to which an oxidant such as sodium hypochlorite or hydrogen peroxide is added. The solution is sprayed over the packing media, and blowdown to the sewer is required.

Demisting of exhaust removes most mist particles from the airstream, but some can remain in the treated gas discharge. This method is not recommended for pump stations unless there is a clean water supply and high air flow rates ($\gg$ 10,000 cfm), and should not be installed at a remote pumping station. The units are effective at treating foul air but require chemical storage tanks, recycling pumps, and frequent monitoring.

Dry Chemical Scrubbing

Similar to wet chemical scrubbers in removal mechanism, dry chemical units operate by chemical oxidation. However, dry chemical scrubbers operate using engineered media designed to remove specific groups of odorous compounds. The media consists of a high surface area support, such as activated alumuna, coated with a chemical. Typical chemicals used in these scrubbers include potassium permanganate, chlorine dioxide, and iron oxide. Dry chemical scrubbing units are available as manufactured packages in low to medium airflow ranges, or the media can be purchased in bulk quantities. Many of the chemicals used in dry chemical scrubbers are specifically directed at hydrogen sulfide and, consequently, may not treat other odorous contaminants. It is therefore important that the proper media is selected to remove the desired compounds. To treat a variety of odorous compounds, hybrid chemical systems are frequently used. The hybrid systems contain a mix of different dry chemical media and activated carbon. The different types of media can be mixed, layered, or staggered in multiple beds.

Other Treatment Systems

High-temperature oxidation of foul airstreams is rarely used at pumping stations because of its high cost. Achieving consistent performance with ozone systems has proven difficult. Neutralizing chemicals and counteractants are also sometimes used. Two-stage systems are occasionally used when a very high degree of foul air treatment is required.

Enhancing Atmospheric Dispersion of Foul Air

Occasionally, elevating the foul air discharge point to about 30 or 40 ft above ground level can solve a local odor problem. A discharge stack may be aesthetically objectionable, but with small foul airflows, a hollow light pole 200 to 250 mm (8 to 10 in.) in diameter can be used as a disguised stack.

Discharging gas at the roof line of the pumping station building should be avoided because downwash from wind blows the discharge to ground level. Carefully evaluate how local wind conditions will disperse odors. In critical situations, atmospheric odor modeling may be necessary to determine downwind effects [6, 7].

Sealing wet wells, with the intention of avoiding any gas discharge, is not normally recommended. Maintenance personnel need access to the wet well on occasion and resealing is difficult. Also, so-called gas-tight structures often have leaks. Safety and corrosion issues normally demand some degree of ventilation. Small wet wells need only a ventilation pipe to allow gases to be discharged either close to the ground or at an elevation of at least 3 m (10 ft), as the situation requires (see OSHA requirements [3] and NFPA 820).

Controlled Atmospheres for Sensitive Equipment

Motor control centers should be placed in a separate room with an atmosphere controlled for temperature and contaminant level.

Sensitive electrical and electronic equipment (such as PLCs or AF controllers) require highly controlled atmospheres to avoid corrosion and to keep temperatures at least below 30°C (85°F). ISA 71.04 (1986) defines four "Severity Levels": G1 is mild, G2 is moderate, G3 is harsh, and GX is severe. Environments in wastewater pumping stations are typically G3 (harsh), an environment in which there is a high probability that corrosive attack will occur. HVAC systems for electronic equipment such as AF controllers should be designed and specified to produce a G1 environment, as partially described in Table 23-3.

Table 23-3. ISA G1 Environmental Conditions

Atmospheric contaminant	Maximum concentration
Relative humidity	50%
H_2S	3 ppb
SO_2, SO_3	10 ppb
Cl_2	1 ppb
NO_X	50 ppb
HF	1 ppb
NH_3	500 ppb
O_3	2 ppb

In addition, temperature should be maintained as low as possible [22°C (72°F)] or lower (if possible). Rooms or cabinets are often pressurized to 2.5 mm (0.1 in.) of WC to minimize leaks of outside contaminated air. Rooms or cabinets need to be sealed to achieve this limitation. Clean, filtered air is required to eliminate dust and other airborne particles.

To achieve these standards, air cooling is almost always required, along with filtration for particulate control. At wastewater stations, hydrogen sulfide is the primary contaminant of concern, although ozone concentrations in urban areas are often excessive, and various hydrocarbons can also be present in the ambient air. Use three-stage filters (prefilter at 10 μm, followed by a carbon filter, followed by a 1-μm, high-efficiency filter) on air supplied from the upwind side of the pumping station. Relative humidity control is also required for many locations. Chemical filtration media are often used to adsorb pollutants, and sometimes to oxidize them. Activated carbon media, along with potassium permanganate, are often used [9].

Equipment for integration into HVAC systems, or separate, stand-alone equipment is available. Equipment is also available that performs these functions on a small scale for individual cabinets or for banks of cabinets. These facilities pressurize the cabinet to allow it to operate within a room containing harsh environmental conditions, and they include filtration, pollutant adsorption filters, air conditioning, and, sometimes, relative humidity control. Airflow capacities in the 2.8 to 5.7 m^3/min (100 to 300 ft^3/min) range are typical for cabinet-type systems.

23-4. Dry Well Design Guidelines

The following guidelines are recommended for designing the heating, ventilating, and cooling systems of dry wells. They supplement those briefly outlined in the Ten-State Standards [4]. Other suggestions can be obtained from *Design of Municipal Wastewater Treatment Plants* [10].

Heating

There are three basic heating techniques:

- *Space heaters (gas, oil, or electric) with a thermostat and a summer fan-only switch*. Heater locations and airflow patterns should be chosen to produce thorough circulation through the space. In milder climates this may be the simplest solution. A separate fresh-air heater may also be provided. This approach has the *least* temperature control, but usually the stations do not require precise temperature control.
- *Infrared radiant heating*. With either gas or electricity, radiant heating is a simple and effective way of heating large, spacious working areas. Comfort conditions are achieved at a lower air temperature with radiant heat than with any other heating method.
- *Ducted systems with temperature controls*. Mixing dampers can be used to control the proportion of outdoor air to recirculated air in response to room temperature. A supply-air low-limit control is recommended.

Building heat loss in cold climates should be minimized by using wall and roof insulation, double glazing, and air dampers properly installed with blade and jamb seals to limit unwanted air infiltration. Follow a locally adopted energy conservation code or ASHRAE Standard 90-80. Maintain adequate space temperatures to facilitate essential activities and prevent freezing (see Table 23-4). Install a manual-reset, capillary-type, low-temperature thermostat downstream from the preheating or heating coils and set it at 3°C (37°F) to stop the fan and energize a remote warning for 100% outside air units in unattended stations. For systems with a recirculating air damper, an automatic-reset low-temperature thermostat may be used. In addition to energizing the warning signal, it repositions the dampers for recirculation

Table 23-4. Recommended Temperatures

	Minimum		Maximum	
Space type	°C	°F	°C	°F
Continually occupied areas	20	68	26	78
Occasional work areas	13	55	35	95
Unoccupied areas	>4	>40	43	110

only. It may either stop the fan or leave the fan running at the designer's option. The fan and dampers automatically resume their normal operation when the freeze potential is no longer sensed. Include extra heating capacity controlled by a thermostat (or use portable heaters) that can raise the temperature in the work area to a reasonable, comfortable level when maintenance and repairs must be done in the coldest expected weather. Locate air outlets and inlets to take advantage of the natural upward convection currents from hot equipment as well as to offset the heat losses at their source. Introduce heat near the bottom of exposed walls. If heated air is supplied from overhead (either from ducts or from unit heaters), its downward velocity must be sufficient to force it down to the occupied level. Consider central heating systems if heated ventilation air is to be supplied to multiple spaces totaling more than about 300 m^2 (3000 ft^2) of floor area. Specify industrial-quality equipment for long service with low maintenance.

Ventilating

The primary function of dry well ventilation is the removal of excess heat for the protection of equipment (specifically, motors and controls) and for the comfort of personnel. Secondary functions are moisture removal, corrosion prevention, and odor reduction. The outdoor air ventilation rate can be varied in cold, dry weather by using thermostatically controlled mixing dampers to blend sufficient outdoor air with return air to match the need for heat removal. In SI units, the required ventilation is

$$Q = \frac{G - L}{(1208)\Delta T} \tag{23-1a}$$

where Q is the airflow rate in cubic meters per second, G is the heat gain in watts, L is the heat loss in watts, $G - L$ is the net heat gain, and ΔT is the temperature difference in degrees Celsius.

In U.S. customary units,

$$Q = \frac{G - L}{(1.08)\Delta T} \tag{23-1b}$$

where Q is the airflow rate in cubic feet per minute, G is the heat gain in British thermal units per hour, L is the heat loss in British thermal units per hour, $G - L$ is the net heat gain, ΔT is the temperature difference in degrees Fahrenheit, and $1.08 = (0.075\ \mathrm{lb/ft^3})$ $(0.2395\ \mathrm{Btu/lb \cdot deg})(60\ \mathrm{min/h})$.

Cold Weather

The recommended minimum air temperature in an occasionally occupied space is 13°C (55°F) (see Table 23-4). If the supply air is thoroughly mixed with warmer room air near the ceiling, the supply air temperature can be reduced to as low as 7°C (45°F) when necessary for heat removal.

Heat recovered from the dry well exhaust air can be used to help heat the wet well by using any of the heat exchangers discussed in Section 23-5. There must, however, be complete separation of the two airstreams to preclude any leakage of toxic or explosive gases from the wet well to the pump room. The heat pipe coil is the safest exchanger in this regard. The heat recovery device should be protected against corrosion.

Hot Weather

The required summer air-handling capacity of ventilating equipment for heat removal is determined for a given heat load by the difference between the outside air temperature and the allowable maximum indoor temperature (ΔT in Equation 23-1).

The heat gain to the space includes:

- Heat generated by pump motors, drivers, and controls
- Heat from other miscellaneous motors
- Heat from lights and occupants
- Inward heat transmission through walls, windows, and roof
- Direct solar heat gain through windows and skylights, and indirect solar gain through walls and roof previously heated by the sun
- Heat from emergency generators (obtain the data on heat lost to the room and the recommended airflow rate for this equipment from the engine manufacturer).

In hot climates the rate of outdoor ventilation needed to remove excess heat may be exorbitant due to the small difference between the outdoor temperature and the allowable indoor temperature. First try to reduce the required ventilation rate by reducing the heat gain to the space. For example, cooling air that leaves a pump motor is well above room temperature. If that warm air can be immediately removed by a

hood or by direct-ducted exhaust before mixing with room air, the total room heat gain and required ventilation are less. Of course, hoods and ducts must not unduly interfere with pump motor access and removal.

Evaporative Cooling

In dry climates, cooling the air supply by evaporation is effective. Direct evaporative cooling occurs when unsaturated air passes through a water spray or a wet filter. The heat required to evaporate part of the water is extracted from the air and, thus, lowers its dry bulb (sensible) temperature. The added moisture increases the total heat content of the air as measured by its increased wet bulb temperature. Less of the more humid but cooler air is required to remove the sensible heat from the space at a given rate.

Indirect evaporative cooling occurs when evaporatively cooled air is used in turn to cool warmer air by passing both airstreams through an indirect heat exchanger.

"Indirect–direct" evaporative cooling units are even more effective. The supply air is first cooled by passing it through an indirect heat exchanger in counterflow to a second (waste) airstream previously cooled by direct evaporation. The supply airstream is then further cooled by direct evaporation. In this way, both the dry bulb and wet bulb temperatures of the original airstream are reduced.

Due to the small ΔT usually available between supply and room air temperatures, the evaporatively cooled supply air rate is usually in the range of 15 to 30 air changes per hour. Such rates require larger fans and ducts and better air diffusion than a refrigerated air conditioning system with a larger ΔT. Recirculation of evaporatively cooled air is impractical because of the resulting buildup of humidity.

Be warned that evaporative cooling systems increase the humidity, may cause condensation, make rusting more likely, and increase maintenance. Unless winterized by complete draining in severe climates, the evaporative equipment can be badly damaged by freezing.

Refrigerated Cooling

Refrigerated cooling may be necessary where outdoor temperature and humidity are high and where internal heat gains are relatively large, as in a control room containing inverters for variable-speed motors. Other cooling methods should be investigated first because of the comparatively high operating cost of refrigeration compressors.

Chlorination Rooms

Chlorination and chlorine storage rooms require special ventilation. Both the *Chlorine Manual* [11] and the Ten-State Standards [4] recommend that these rooms be heated to 16°C (60°F) and have the exhaust fan capacity required for 60 air changes per hour. The exhaust fan should be energized by an automatic door switch and a manual switch (located outside the door) that simultaneously opens an air intake damper. Ventilation air should be introduced at the ceiling and exhausted at floor level because chlorine is heavier than air.

23-5. Energy Use and Conservation

The high energy prices of the recent past years reinforce the importance of energy conservation. The adoption of energy-saving methods is at least partially responsible for the present trend toward increased energy availability. Awareness of the irreplaceability of fossil fuel resources warrants continued efforts by designers to avoid its waste.

Energy Sources for Heating

Available heat energy sources frequently include electricity, natural gas, propane, and fuel oil. Compare the probable life-cycle costs and reliability of each before making a choice. For stations in warm to moderate climates, the air-to-air heat pump may be an option for both heating and cooling as long as these loads are reasonably in balance. In an electric motor-driven air-to-air heat pump, a refrigerated coil is used to extract heat from outdoor air. That heat, plus the heat of compression, is then delivered to the space by a condensing coil in the supply airstream. The "coefficient of performance" (COP) of such a system is the ratio of electric energy input to heat energy output. With an outdoor temperature of 7°C (45°F), for example, a COP of about 2.5 (which is not unreasonable) means that the heat energy delivered is 2.5 times greater than would be obtained from an electric resistance heater with the same electrical

input. The greatest economy results when the summer cooling load also approximates the cooling capacity of the refrigeration unit selected, thus allowing year-round operation of the unit.

Solar heating is generally unsuitable because a full back-up heating system (or very large solar heat storage) is necessary to meet demand at the times when solar heat is unavailable. Furthermore, there is no compatible use for the greater heat collection in summer that may require separate expensive disposal.

Major Energy Uses and Conservation

The major HVAC energy use is for heating the station and its ventilation air in cold weather and for cooling the station and its ventilation air in hot weather. Continuously operating fans handling large quantities of air against substantial static resistances are the next-largest energy users. Energy economy results from (1) a well-insulated structure, (2) prudent selection of air quantities and indoor design temperatures, and (3) minimized air-path resistance. Both the maximum and minimum indoor design temperatures should be balanced between operator efficiency and operating cost. Select air conditioning equipment to be fully loaded at peak summer weather conditions so that it can perform more acceptably at partial load conditions. Multiple or two-speed fans allow operation at reduced capacity and cost at light load conditions. Multistage thermostats or step controllers can actuate additional fan capacity as the space temperature approaches the design maximum.

Introducing ventilating air at lower levels (in dry wells only) and relieving or exhausting it at the ceiling augments natural convection flow. Early opening of properly sized inlet and outlet dampered louvers allows excess heat at partial loads to be removed by convection, which delays the need for mechanical ventilation. If the design of the station allows, consider passive cooling and heating by (1) bringing air through ducts buried below grade to take advantage of the earth's temperature, which usually ranges from 10 to 16°C (50 to 60°F); or (2) using skylights and other glass openings arranged to admit direct sunlight only in the winter months.

Automatic backdraft dampers are necessary on multiple supply and exhaust fans to prevent short-circuiting the air. Intake louvers should preferably be located on the windward side of the building so that prevailing wind pressure assists station ventilation. Wall-type propeller exhaust fans should exhaust to the leeward side of the building. Propeller fans move more air per unit of power than other fan types if the air resistance is 125 Pa (0.5 in. WC) or less, and they offer low resistance to natural airflow when not running. Propeller fans are usually less efficient than centrifugal fans against the higher resistances of long duct runs or heat-recovery devices.

Where wet wells must be heated for extended periods, heat-recovery devices should be considered. Their efficiency is the recovered fraction of the heat content difference between the two airstreams. Much less "new" heat needs to be added to the make-up airstream when such devices are used. The following are commercially available types of air-to-air heat-recovery devices. Recovery efficiency, ranging from 70 to 80% for the first three types, varies with the temperature difference between the warm and cool airstreams.

- *Plate-type heat exchangers*. Warm and cool airstreams pass on opposite sides of metal plates and exchange heat by conduction and convection. Cleaning is difficult, condensate drainage is a problem, and duct arrangements are sometimes awkward.
- *Heat wheels*. These rotate through the counterflowing airstreams. The wheel absorbs heat from the warmer airstream and releases it to the cooler airstream. The heat-recovery rate varies with the speed of rotation.
- *Heat pipe coils*. These consist of a number of individual, sealed, finned tubes containing a metered amount of refrigerant. One half of each tube is placed in the warmer airstream where it absorbs heat and vaporizes the captive refrigerant. The refrigerant then condenses in the opposite half of the tube and releases heat to the cooler airstream. The advantages include no moving parts and the ease of corrosion protection by applied coatings. Face and bypass dampers and other methods of heat-recovery control are used.
- *Coil energy-recovery loops*. Coils both in the exhaust and intake ducts are interconnected with piping and a pump to transfer heat from the exhaust air to the outside make-up air. A glycol solution is used as the heat-transfer medium if the coils are subject to freezing. A three-way bypass valve on the intake coil is controlled to keep the air temperature leaving the exhaust coil above 0°C (32°F) to prevent the freezing of condensate on the exhaust coil in cold climates. The exhaust coil, at least, should be corrosion-protected. Overall efficiency is about 50%. This heat-recovery method is useful where supply and exhaust fans are widely separated.

These devices can recover a significant portion of exhaust heat that would otherwise be wasted, but their air resistance, which is in the range of 125 to 250 Pa (0.5 to 1.0 in. WC), penalizes fan energy. Filters are desirable in both airstreams to prevent dirt from clogging the closely spaced fins of the heat transfer coils.

In determining the cost-effectiveness of such devices, consider (1) the installed first cost, (2) the purchased energy cost, (3) the actual energy savings, (4) the additional space required, (5) the added complexity of the system, (6) the increased maintenance costs, (7) the reliability, and (8) the higher resistance against which both the supply and exhaust fans must operate. When used in air conditioning systems, such energy-recovery devices can also pre-cool the hot make-up air by thermal exchange with cooler exhaust air. This dual use makes them more economical in applications requiring heating in cold weather and refrigerated air conditioning in summer.

23-6. Corrosion Protection

Metallic equipment, especially ductwork, is susceptible to deterioration in varying degrees due to moisture, acid (from hydrogen sulfide and moisture), and salt often contained in the air in wet wells. Raw edges of cut metal sheets and surface scratches provide a starting point for oxidation to spread under coatings of metals such as the zinc on galvanized steel. Type 316 stainless steel is generally corrosion-resistant, but its discoloration may be unacceptable aesthetically. Copper and copper-bearing materials are highly susceptible to attack by hydrogen sulfide and should be avoided or protected in wastewater stations. Aluminum should not be used in damp chlorine rooms. Where metals are the logical choice for items such as louvers, dampers, and grilles, they should be Type 316 stainless steel, aluminum, or galvanized steel, or be protected with an epoxy or baked phenolic corrosion-resistant coating applied by the manufacturer.

Polyvinyl chloride (PVC) or fiberglass-reinforced plastic (FRP) are more corrosion-resistant (and more expensive) than Type 316 stainless steel, but their detracting characteristics—such as fire and smoke ratings, structural strength, greater weight and support requirements, and the greater difficulty of on-site modification—must be given due consideration. PVC and FRP ductwork are usually factory-fabricated from sheets of material 4.8 mm ($^3/_{16}$ in.) or more in thickness. Joints in either type of duct can be made by solvent welding. FRP joints also can be formed with multiple layers of glass cloth laid up by hand in a resin binder. Duct accessories used, such as dampers, turning vanes, hangers, and fasteners, should be equally corrosion-resistant.

Fans, tanks, and piping may also be fabricated from PVC and FRP. Heat exchangers (which must be metal for good heat transfer) should be protected by corrosion-resistant coatings, such as thermosetting phenolics. The metal is dipped and then oven-baked for several hours. Two or more coats may be used for severe applications. Epoxy compounds and some phenolics can be applied cold by painting or spraying. The resistance of the coatings to the various causes of corrosion can be obtained from their manufacturers. The heat-transfer capability of corrosion-protected heat exchangers may be decreased by 15% or more, so compare cost, energy, and permanence in making a selection.

The types and severity of corrosion to be expected should be ascertained from previous installations at the same or at a similar site, if possible. Also consult experienced coating vendors. The difficulty and cost of replacing unprotected components should be weighed against the cost for protection. Note that corrosion protection increases first cost from 20 to 400%, but replacement costs can be even higher.

Electrical switches and control components should also be protected from corrosion. Such protection can be obtained in a number of different ways.

- Control hydrogen sulfide at the source to the greatest extent possible. A complete odor-control system mitigates the problem except for electrical equipment within wet wells or under tank covers.
- Isolate control equipment in a separate room and design the ventilation to exclude sulfides or, if that is impossible, treat the air supply for the entire room. If electronic equipment is installed, pressurize the room slightly and eliminate chlorine and other corrosive gases as well.
- Hermetically seal relays and switches in plastic boxes attached to plastic conduit, seal the boxes as completely as possible, and put potassium permanganate pellets in the bottom of the boxes.
- Pressurize the boxes or panels to maintain 25 kPa (0.1 in. WC) with either uncontaminated air or with air filtered through potassium permanganate and activated carbon. Such pressurizing in accordance with NEC 500 also reduces enclosure explosion hazard ratings from NEC Class 1, Division 1 to Class 1, Division 2. Adequate differential pressure can be monitored by an attached pressure switch and alarm. To guard against excessive compressed air consumption if the bolts are loose, the gaskets are old, or the panel is warped, a limited

amount of air can be supplied by a small, inexpensive, diaphragm-type fish tank air pump.

23-7. Sequence of Design Steps

Pumping station design is a cooperative effort among the various design disciplines that must be coordinated as the design progresses. Each design step affecting other disciplines should have confirmed agreement before proceeding to avoid later redesign.

Step 1: Write and distribute a design memorandum defining the applicable codes, the indoor and outdoor design temperatures, the building materials and orientation, the sources and amounts of heat gain (especially from motors and engines), and the design air change rate for each space. State the energy source to be used and describe the proposed ventilating, heating, and cooling systems.

Step 2: Calculate the heating and cooling loads for each space. Measure and record wall, glass, and roof areas from the architectural plans. List exposure directions and calculate space volumes. Obtain or calculate heat transmission coefficients (U values) for wall, roof, and window construction. Use the methods in the *ASHRAE Handbook of Fundamentals* [12] to calculate the heat loss and the heat gain for each space. Sum the heating or cooling loads in watts (British thermal units per hour) for each air handling unit and obtain the grand totals.

Step 3: From the appropriate designer, obtain the locations of the main pumps and piping, cranes or hoists, columns, stairs, beams, and other obstacles to air distribution. Also obtain the heat emissions from simultaneously operating equipment. Calculate the initial ventilation rates starting with 12 air changes per hour for wet wells (more in some states or for severe conditions) and 6 (or more) air changes per hour for dry wells. Make preliminary equipment selections. Estimate the outside air infiltration to each space by either the crack or the air change method, and calculate the expected outdoor air heating and cooling loads. See the *ASHRAE Handbook of Fundamentals* [12] for details of the crack and air change methods.

Step 4: From space volumes, heat gains, and design air change rates, calculate the normal and required purge air supply quantities for each space. If the summer airflow for heat removal exceeds 30 air changes per hour, consider evaporative or refrigerated cooling. Choose supply and return/exhaust register locations, assign the proper air quantity to each, and, from catalog data, select register sizes to give desired air throw (for supply) or velocity (return/exhaust). Note the corresponding pressure drops.

Step 5: Make an initial estimate of HVAC motor number and sizes for the electrical engineer. Fan motor horsepower can be approximated in SI units by

$$W = \frac{QR}{E} \tag{23-2a}$$

where W is brake watts, Q is the airflow in cubic meters per second, R is the static resistance in Pascals, and E is the fan static efficiency as a decimal. In U.S. customary units,

$$bhp = \frac{QR}{6356E} \tag{23-2b}$$

where bhp is brake horsepower, Q is the airflow in cubic feet per minute, R is the static resistance in inches of water column, and E is the fan static efficiency as a decimal.

Lacking more precise information, use the fan resistance and efficiency assumptions given in Table 23-5 for the initial motor size calculations. Then, for safety, select the next larger standard motor size. Size all fans for at least 60 Pa (0.25 in. WC) static resistance. For propeller fans more than 0.6 m (2 ft) in diameter, use a motorized shutter to reduce resistance and to give more positive closure.

Step 6: Calculate the final heating and cooling loads and modify the initial air distribution rates as required to meet the allowable difference between the temperatures of the supply air and the room. Maximum temperature differences of 16°C (60°F) for heating and −4°C (25°F) for cooling are recommended.

Table 23-5. Fan Static Pressures and Efficiency

Static Resistance[a]			
Without heat recovery		With heat recovery	
Pa	in. WC	Pa	in. WC
	Directed supply		
250	1.0	620	2.5
	Directed exhaust		
125	0.5	500	2.0

[a]Unless more accurate information is available, use in Equation 23-2(a) or (b) with an assumed fan static efficiency of 65% as initial assumptions for estimating the size of motor required.

Step 7: Determine feasible types and locations for air-handling equipment and lay out tentative duct runs. Avoid obstructions and consider how the ducts should be supported.

Step 8: Beginning with the outlets farthest from the fan, assign cumulative air quantities to each duct section. Using air friction charts or calculators, size the ducts (1) to fit the available space, (2) to give a streamlined airflow, (3) to look presentable, and (4) to simplify duct fabrication.

Step 9: Obtain gross face areas using velocities of (1) 1.3 m/s (250 ft/min) for wall intake and relief louvers, (2) 2.5 m/s (500 ft/min) for wall exhaust louvers, and (3) 5 m/s (1000 ft/min) for openings to roof exhaust fans. Confirm opening sizes and locations with the architect and structural engineer.

Step 10: Draw the duct systems on the plan. From the resistances for registers, straight lengths of ducts, and fittings, cumulatively sum the actual friction drop of the longest duct run from the farthest point back to the fan. Duct friction can be calculated by the Darcy–Weisbach equation, which for air takes the form

$$p = f_{\mathrm{D}}\ (cf \times L/D)p_{\mathrm{v}} \qquad (23\text{-}3)$$

where p is the total pressure friction loss in Pascals (inches of water column), f_{D} is the friction factor (dimensionless), cf is the conversion factor, L is the duct length in meters (inches), D is the duct diameter in meters (inches), and p_{v} is the velocity pressure in Pascals (inches of water column). The friction factor is a function of the Reynolds number, however, and can be calculated only by iteration. Air resistance for straight ducts is more easily calculated with a slide rule made especially for this purpose, with a programmable calculator, or with a computer. Duct friction tables and charts and a listing of fitting friction losses are available in the *ASHRAE Handbook of Fundamentals* [12].

Step 11: Make final best selections of the heat transfer and air-handling equipment based on the design flow rate and the calculated total air resistance, including the friction drops for louvers, dampers, filters, coils, ducts, and registers. The friction drop for clean, pleated glass fiber filters is about 25 Pa (0.1 in. WC), but the fan motor must be sized to supply the required air at a dirty filter resistance of 120 to 160 Pa (0.5 to 0.65 in. WC). Filter and coil resistances are usually considered to be part of the external resistance—separate from the internal resistance of the air-handling unit given by the unit manufacturer.

Step 12: Show the selected air supply and exhaust equipment on the plans. Provide (1) raised housekeeping pads for floor-mounted units, (2) vibration-isolated supports for suspended and wall-mounted units, (3) flexible connectors for duct connections, and (4) ample access for filter, motor, belt, and bearing maintenance. Allow space for replacing heating and cooling coils and filters.

Provide direct access to the equipment. Floor-mounted units are, therefore, preferable to ceiling-hung equipment. Rooftop equipment is more likely to be maintained if it is accessible by stairs.

Step 13: Select the additional air-handling equipment for purging and summer heat removal. The equipment required may be only strategically placed high wall or roof exhaust fans and low dampered weatherproof wall intakes. Nonducted intakes and exhaust fans should be selected to result in no more than 60 Pa (0.25 in. WC) negative pressure within the space. A force of about 120 N (27 lb) is required to open a standard, outwardly swinging door against such a vacuum.

Step 14: Recheck the actual motor kilowatts (or horsepower) needed for the designed systems and correct the motor list as required. Make sure motors are selected for the electrical characteristics available. Coordinate motor, starter, and control locations with the electrical engineer so that all of the necessary wiring is provided. Motors and controls should be listed by Underwriters Laboratories (UL). High-efficiency motors are cost effective for continuously operating fans and pumps. A specification requirement that air-handling equipment shall be rated in accordance with the AMCA No. 99 Standards gives increased assurance that it will meet the rated capacity.

Step 15: Specify each component of the system, stating each important criterion of construction and performance necessary to determine the acceptability of items subsequently submitted by the successful bidder. Equipment schedules are a simple way of summarizing required equipment designations, capacities, and special features. Write a performance description of (1) the intended sequence of operation, and (2) the operating temperatures. Require a submittal of control components and schematic and wiring diagrams. Require acceptance tests to ensure that the installed system functions as intended and meets specification requirements.

23-8. Ventilating System Design

Pump rooms located where the outdoor temperature range is between freezing and 35°C (95°F) and in areas where noise need not be suppressed can often be ventilated without ductwork. Those rooms located below grade, however, require ductwork. For pump rooms at or above grade, louvered air intakes (with dampers and filters) and roof exhaust fans (with gravity shutters and sequencing thermostats) are sufficient. For lower outdoor temperatures, heating is usually required. For higher temperatures, some form of cooling is normally necessary.

Heating and cooling calculations are always necessary to determine the required extent of heat removal or heat addition. The air quantities to be handled, as derived from the heat load calculations, must be compared to those determined by required air change rates or other criteria, and the larger quantity must be used as the design basis.

Heating Load Calculations

Structure heat load is composed of (1) transmission loss through walls, windows, and roof, and (2) infiltration of cold outside air, which must be heated to the design space temperature.

The transmission loss through any portion of the structure envelope is

$$H = U \times A \times \Delta T \tag{23-4}$$

where, in SI units, H is the heat loss in watts per hour, U is the overall heat transmission coefficient in W/(m$^2 \cdot$ h $\cdot$ °C), A is the surface area in square meters, and ΔT is the inside-to-outside temperature difference in degrees Celsius. In U.S. customary units, H is in British thermal units per hour, U is in Btu/(ft$^2 \cdot$ h $\cdot$ °F), A is in square feet, and ΔT is in degrees Fahrenheit. Transmission coefficients for typical walls, windows, and roofs as well as methods of calculating the coefficient for the combinations of materials are given in Chapter 25 of the *ASHRAE Handbook of Fundamentals* [12].

Cooling Calculations

Heat gain throughout the structure, which makes up the external cooling load, is more difficult to calculate than heat loss because of the effects of radiated heat from the sun. This additional heat source must be considered in addition to the heat gain by conduction due to the difference between outdoor and indoor temperatures. Sunlight does not heat space air directly, but it raises the temperature of sunlit surfaces as a result of absorbed radiation.

Many factors influence the amount and timing of solar heat reaching the space. The intensity of incident solar radiation depends on latitude, time of year, time of day, cloud cover, and atmospheric pollution. The time lag between solar heat input and interior air temperature rise may vary from a few minutes to several hours. The lag is affected by the surface color of the exterior, the heat storage capacity and the insulating value of the construction, and the daily outdoor temperature range, among other factors.

The combined heat from both the higher outdoor temperature and solar radiation moves progressively through walls and roof, finally raising the interior surface temperatures. Loss from those surfaces occurs by radiation to cooler surfaces and by convection to the adjacent air. Air warmed by convection expands, becomes lighter, rises, and is replaced by cooler air, which continues the convective process.

Except for minor reflected losses, sunlight passes directly through clear glass and is absorbed by the interior surfaces it strikes. Their increased temperature transfers heat to other cooler surfaces by reradiation and to the air by convection.

Simply stated, solar heat gain is taken into account by using a higher outdoor temperature than actually would exist at the time considered for each heat gain calculation. The *ASHRAE Handbook of Fundamentals* [12] contains tables of cooling load temperature differential (CLTD) data (for use in calculating conduction heat gain through sunlit walls and roofs) and cooling load factors (CLFs) (for calculating the solar radiation through glass). Both sets of data include the effect of time delay due to thermal storage. The total resistance to heat transfer for each type of construction is found by adding the resistances of its components, as illustrated in Example 23-1.

Peak Heat Gain

The peak heat gain to a space is the largest sum of external and internal heat gains that occur simultaneously. Heat removal capacity, in the form of ventilation or cooling, equal to the peak heat gain must be available to maintain the design space temperature. External heat gain results from an outdoor temperature that is higher than the indoor temperature as well as from solar radiation. Internal heat gain comes primarily from operating motors and engines.

Air Intake and Exhaust Openings

Intakes for ventilation air should be through screened louvers that exclude rain, snow, birds, and insects. Ducted intakes that are connected to air-handling units with filters that exclude insects need only bird screens. Nonducted intakes should include insect screens or filters. All screens and filters must be readily accessible for cleaning or replacement. Bird screens do not need mesh openings smaller than 25 mm ($\frac{1}{2}$ in.). The air resistance of filters used at unducted wall intakes should not exceed 24 Pa (0.1 in. WC) (when clean) to limit negative pressure in the space (created by exhaust fans) to 63 Pa (0.25 in. WC). Low negative pressure makes doors easier to open and close. Both the louvers and screens should be constructed of a corrosion-resistant material or should have a corrosion-protective coating. Provide an intake damper at each louver in cold climates, preferably with positive closure on failure of its pneumatic or electric actuator. Hurricane-prone locations may warrant an additional, manually operated damper (on the outdoor air intake) that can be locked closed when a storm approaches.

Air exhaust openings should be protected from the weather by a wall louver, hood, penthouse, or weather cap. The discharge openings should be located so as to minimize unwanted recirculation into air intakes.

A high-velocity vertical discharge of untreated wet well exhaust air—about 20 m/s (4000 ft/min), which requires 250 Pa (1 in. WC) static pressure—may reduce odor near the station (but with attendant noise). A thorough model study of the likely results (which are not readily predictable) should precede any use of this approach for odor control.

Ducted Ventilation

A ducted air supply is usually necessary to achieve the desired air distribution pattern. Once delivered to the right location, the air can be directed upward or downward by registers (grilles with directional vanes and attached balancing dampers), or it can be more thoroughly mixed with room air by diffusers that give high entrainment of room air. The distance an airstream travels after leaving the outlet (the "throw") depends on its velocity, temperature, and any obstructions to straight-line flow. Catalog data on grilles, registers, and diffusers give the throw and pressure drop for a given flow rate of standard air. The throw is longer for an outlet discharging horizontally within about 0.6 m (2 ft) of a smooth ceiling due to the low induction of room air.

Install a manual volume damper in each main duct branch and at each inlet or outlet grille to allocate or "balance" the total air quantity among the various points of supply and exhaust. Such dampers are furnished as a standard part of registers, but must be separately specified for grilles and diffusers. Complete testing and balancing of ducted systems should be specified to deliver the design air quantity within 10% at each register and diffuser.

Wherever an air duct penetrates a rated fire separation required by code, install an accessible fire damper of equal fire rating. Codes generally require rated fire separations between floors, around vertical shafts and stairs, at the envelope enclosing life safety exitways, and, in some instances, around mechanical equipment rooms and storage rooms. The fire damper is held open by a fusible link that melts and allows the fire damper to close automatically if the air temperature rises above its fusion point of 52 or 74°C (125 or 165°F). The fusible link is selected on the basis of location and normal air temperature. Such dampers are required by NFPA 90A (which is included by reference in many other codes) to prevent fire and smoke from spreading through the air ducts. High-temperature thermostats or smoke detectors (as required by applicable codes) should be installed and interlocked to stop both the supply and the exhaust fans if fire is detected. Some codes require tight-closing, remotely operable smoke dampers (which are actuated by smoke detectors) at certain locations.

Detailed duct design methods are given in Chapter 33 of the *ASHRAE Handbook of Fundamentals* [12].

Ventilation Controls

Excessive and intricate controls for ventilation systems (or any HVAC system) are likely to result in more operational problems than they were intended to overcome. The rule is to keep controls as simple as feasible while still maintaining the temperature within an appropriate range. A temperature variation from above freezing to 43°C (110°F) would not be serious in an unmanned pumping station, but in an occupied office or control room the temperature variation should be held to a range of 6°C (10°F) or less.

The following are suggested ventilation control functions:

- Energize controls when the supply fan starts.
- Control outside-return air mixing dampers to use at least 10% outside air initially and modulate them to increase the proportion of outdoor air as the space temperature rises, subject to a discharge low-limit control set at 7°C (45°F) minimum.

- Interlock the exhaust fan to run with the supply fan, and the exhaust damper to operate in conjunction with the outside air damper.
- Interlock the supply fan to stop if the supply air high-limit thermostat or smoke detector functions, or if the supply air low-limit thermostat senses a temperature below 3°C (37°F).
- For a wet well, start a supplementary exhaust fan and open an air intake damper to purge the space if a gas detector senses 20% of the LEL of a combustible gas.
- For a dry well, start an additional exhaust fan and open an air intake damper when a space high-temperature thermostat senses a temperature above 30°C (85°F).

Example 23-1
Design of a Ventilating System

Problem: Plans for a water pumping station are shown in Figures 23-1 through 23-4. There are currently four 200-hp pumps (three duty, one standby), but six 350-hp pumps (five duty, one standby) will be used in the future. Although the office is currently occupied, the station will also be unattended in the future. The latitude is 41.5° north, the elevation is 660 ft, and the daily temperature range is 20°F. Other temperature data are as follows:

Design conditions Time equaled or exceeded	Winter 99%	Summer 1%
Outside temperature, dry bulb	−20°F	94°F
Outside temperature, wet bulb	—	74°F
Inside temperature (occupied)	70°F	78°F
Inside temperature (unoccupied)	60°F	105°F

Calculate (1) the heat transmission coefficients for superstructure walls, roof, and glass; (2) the heat loss for the pump room; (3) the present heat gain for the pump room; (4) the future heat gain for the pump room; and (5) the requirements for heat and ventilation.

Solution:

(1) Heat transmission coefficients. Refer to the *ASHRAE Handbook of Fundamentals* [12], pp. 23.12–23.20 and Tables 1, 3A, and 3B. Sum the component thermal resistances (R) for each surface of the enclosure and invert ($1/R$) to obtain the heat transmission coefficients (U) values (i.e., $U = 1/R$). An example follows.

(2) Heat loss for pump room. Calculate the heat losses using Equation 23-4

Lower walls		Roof		Glass
Outside surface:	0.17	Outside surface:	0.17	Double solar bronze, nonreflective
4-in. face brick:	0.44	Built-up roof:	0.33	
1-in. styrofoam:	4.0	3-in. lightweight fill	4.69	
1-in. air space:	0.94	1-in. polyurethane	6.25	
6-in. concrete block:	0.91	6-in. flexicore	0.66	
2-in. glazed tile:	0.60			
Inside surface:	1.35	Inside surface:	0.61	Winter $R = 2.05$
	$R = 8.41$		$R = 12.71$	Summer $R = 1.79$
	$U = 1/8.41 = 0.12$		$U = 1/12.71 = 0.079$	$U = 0.49$[a]
				$U = 0.56$[a]

[a]Factors for heat transfer through glass differ for winter and summer conditions (see ASHRAE [12], p. 27.10, Table 13).

($Q = A \times U \times \Delta T$). (Also see the *ASHRAE Handbook of Fundamentals* [12], p. 25.1ff.) For algebraic consistency, heat gains are shown as positive and heat losses as negative. See Figure 23-4 for high and low walls. In tabular format, heat losses are as follows:

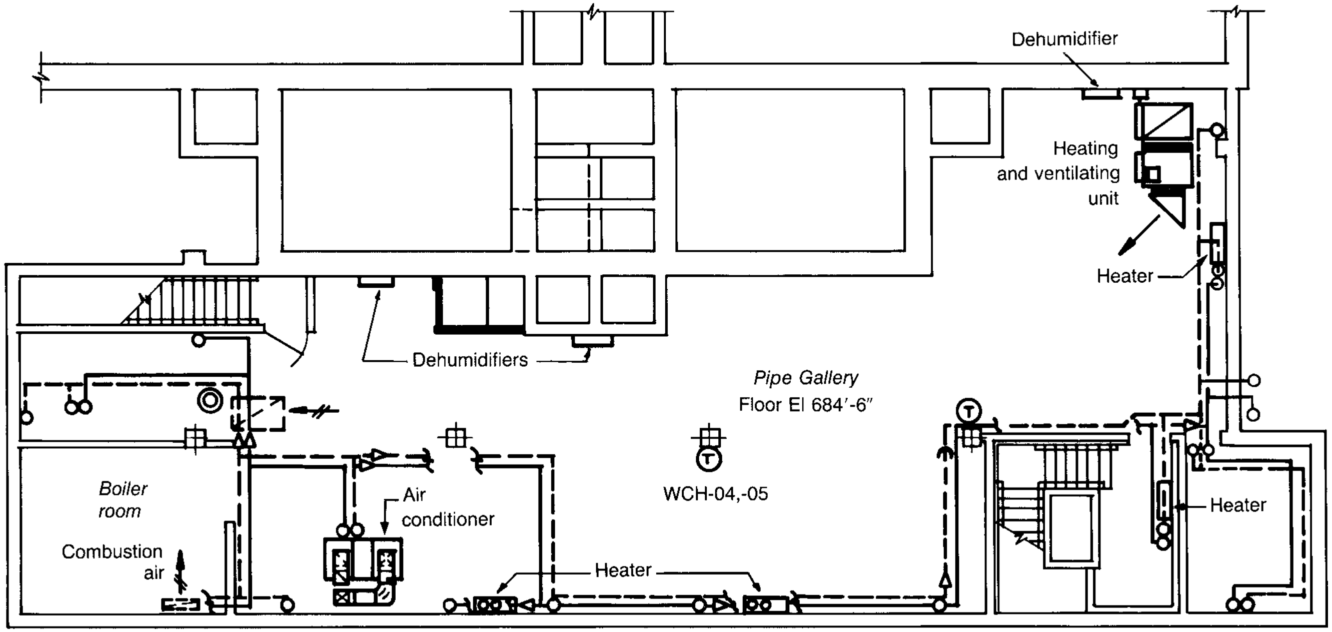


Figure 23-1. Basement plan.

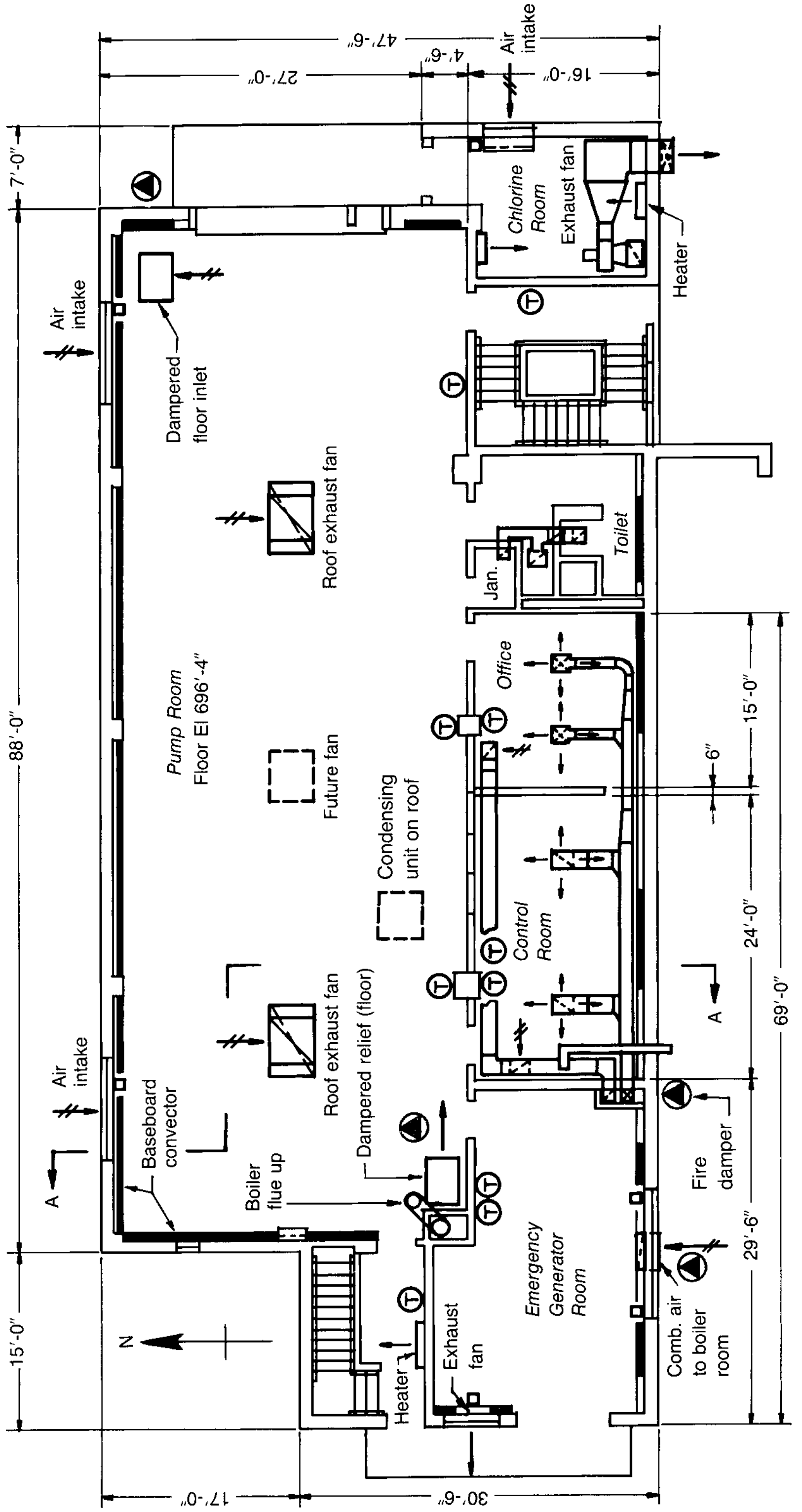


Figure 23-2. Ground-level floor plan.

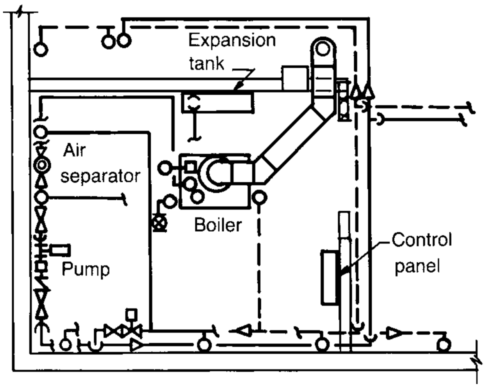


Figure 23-3. Boiler room plan.

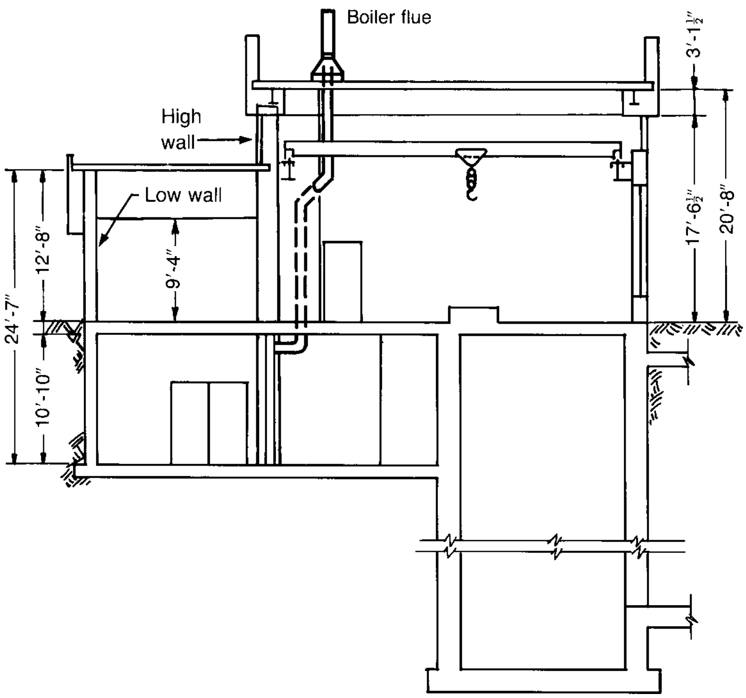


Figure 23-4. Section A-A from Figure 23-2.

Low walls above grade: $(88 + 17 + 27)12.7 \times 0.12(60 + 20) =$	$-16{,}094$
High walls: $2(31.5 + 88) \times 3.3 \times 0.28 \times 80 =$	$-17{,}667$
Glass: $2(85 + 30)3 \times 0.49 \times 80 =$	$-27{,}048$
Doors: $[(12 \times 10) + (3 \times 7)](0.59 - 0.28) \times 80 =$	$-3{,}497$
Roof/ceiling: $30 \times 88 \times 0.079 \times 80 =$	$-16{,}685$
Infiltration (based on 0.75 AC/h at 80°F ΔT):	
$(20.7 \times 88 \times 30 \times 0.75 \times 1.08 \times 80)/60 =$	$-\underline{59{,}020}$
Winter pump room heat loss (Btu/h):	$-140{,}011$

(3) Pump room present summer heat gain. Refer to ASHRAE [12], pp. 26.8, 26.19, 26.29, 26.32, and Tables 5A, 11A, and 24. Note that heat gain due to motors is

$$\text{Heat gain (Btu/h)} = bhp\left(\frac{1-E}{E}\right)2545 \tag{23-5}$$

where *bhp* is the brake horsepower, *E* is the overall motor efficiency as a decimal, and 2545 is the British thermal units per hour · brake horsepower.

For the cooling load temperature difference (CLTD) adjustment for roof and walls in the following example, see ASHRAE [12], pp. 26.8 and 26.10, Tables 5 and 7, Note 2. For glass transmission factors, see Tables 10, 11, and 13, pp. 26.14–26.17, and Table 29, p. 27.29 together with their accompanying explanations and example. For our example, maximum heat transfer occurs at 4 P.M. on July 21. The pump room wall is Group No. C. Directions are identified using capital letters (N = north, etc.).

Structure solar and transmission heat gain:

The heat gain is determined as follows:

Roof with a suspended ceiling (see ASHRAE [12], p. 26.8, Table 5, Roof 12):	
$0.079 \times 30 \times 88[(30+1)+(78-104)+(84-85)] =$	+834
N wall:	
$0.12 \times 15.3 \times 88[(12-0)0.83+(78-104)+(84-85)] =$	−2,753
E wall:	
$0.12 \times 15.3 \times 28[(29-0)0.83+(78-104)+(84-85)] =$	−151
W wall:	
$0.12 \times 15.3 \times 18.6[(16-0)0.83+(78-104)+(84-85)] =$	−469
S wall (upper):	
$0.12 \times 85 \times 1.5[(20-1)0.83+(78-104)+(84-85)] =$	−172
N glass, solar:	
$3 \times 86 \times 38 \times 0.70 \times 0.55 =$	+3,775
E glass, solar:	
$3 \times 30 \times 216 \times 0.24 \times 0.55 =$	+2,566
W glass, solar:	
$3 \times 30 \times 216 \times 0.49 \times 0.55 =$	+5,239
S glass, solar:	
$3 \times 86 \times 109 \times 0.43 \times 0.55 =$	+6,651
Glass conduction:	
$3(86+30)2 \times 0.56(-12) =$	−4,677
Total summer structure external heat gain (Btu/h) =	+10,843

Internal heat gain:

Pump motor heat gain in Btu/h (from Equation 23-5):	
$3 \times 200[(1-0.9)/0.9]\ 2545 =$	+169,667
Estimated auxiliary motor heat:	
$20[(1-0.67)/0.67]\ 2545 =$	+25,070
Heat from lights:	
$30 \times 88 \times 2 \times 3.413 =$	+18,021
Heat added by people:	
$2 \times 350 =$	+700
Total internal peak heat gain in Btu/h:	+213,458

Present total internal heat gain: Adding the structure heat gain to the internal peak heat gain yields the pump room's present maximum heat gain:

$+213{,}458 + 10{,}843 =$ +224,301

(4) Pump room future internal heat gain.

Pump motor heat gain:	
5×350 (10/90) 2545 =	+494,861
Auxiliary motor heat gain:	
300×0.67 (10/90) 2545 =	+56,838
Lights:	
$30 \times 88 \times 2 \times 3.413 =$	+18,021
People:	
$2 \times 350 =$	+700
Total internal heat gain =	+570,420
Total summer structure external heat gain [from part (3) above]:	+10,843
Pump room future maximum heat gain:	+581,263

(5) Ventilation requirements. The present maximum summer ventilation requirement for a 10°F space temperature rise above the outdoor temperature is as follows. The present exhaust in cubic feet per minute is: $224{,}301/(1.08 \times 10) = 20{,}769\ \text{ft}^3/\text{min}$. The total future exhaust in cubic feet per minute is $581{,}263/(1.08 \times 10) = 53{,}821\ \text{ft}^3/\text{min}$.

Currently use two exhaust fans at $17{,}940\ \text{ft}^3/\text{min}$ each and include a capped curb for a third identical fan in the future. Thermostats are provided to turn on the three fans and open corresponding outdoor intake dampers in sequence at 85, 90, and 95°F, respectively.

23-9. Design of Heating Systems

Any structure for which the heat loss exceeds the simultaneous heat gain for a period of time will experience a drop in indoor temperature. To maintain the design indoor temperature, a heating system of some type must add an amount of heat equal to that net heat loss at any given time.

Heating System Types

Heating systems can be categorized in the following ways:

- By heat source—steam, hot water, warm air
- By energy source—gas, oil, electricity
- By distribution method—central system, unitary.

Any combination of the three categories is possible. In stations with few individual spaces and those in warm climates, only unitary equipment (such as cabinet heaters or unit heaters) is needed. For small stations, an oil- or gas-fired warm-air furnace is frequently applied and supplemented by electric unit heaters or baseboard heaters in spaces such as chlorine rooms and toilets from which code prohibits return air. Small stations require minimal or no ducting, although a return duct taking air from the lowest level prevents the pooling of heavy cool air in the bottom space.

Duct distribution systems are used with furnaces or other air-handling units to distribute warm air (including ventilation air) to multiple rooms and separate floors. In the simplest system, the heat source is cycled on and off by a single thermostat located in the most representative space. The temperature is controlled in that space only, and some variation in the temperature of other spaces may be expected.

Heating Load

The extent of heat loss depends primarily on the insulating value of the building envelope and the inside-to-outside temperature difference, and secondarily on wind velocity and outdoor air infiltration. In addition to the structure heat loss, heat also is required to raise the outdoor ventilation air to room temperature. The indoor design temperature is set by the designer. The setting should be high enough to keep the temperature of interior surfaces above the

interior air dew point to prevent condensation. An indoor dry-bulb temperature of 13°C (55°F) usually meets that criterion for a well-insulated building and allows for normal maintenance activities without excessive discomfort. A lower indoor temperature (but well above freezing) may be acceptable in an unstaffed station if (1) the heating system has the capacity to raise the space temperature as needed for extended maintenance work, or (2) portable auxiliary heating equipment is available.

The design outdoor temperatures for many locations are listed in the *ASHRAE Handbook of Fundamentals* [12]. If continuous operation of the pumping station is critical, use the 99% winter design outdoor dry-bulb temperature. Lower temperatures normally occur no more than 1% of the time. Additional weather data can be obtained from the National Climatic Center of the National Oceanic and Atmospheric Administration in the Department of Commerce and from Engineering Weather Data [13].

To calculate the heat loss from a structure, determine the U value, area, and temperature difference for each surface exposed to the outdoors and substitute those related values in Equation 23-4 to obtain the heat loss. The sum of the losses through all such surfaces in each room is the room transmission heat loss, and the total for all rooms is the building transmission loss. To this must be added the heating loads due to the infiltration of outside air, outdoor air introduced to ventilate the space, and the evaporation of moisture occurring within the space. The infiltration heat requirement can be decreased or disregarded if a positive pressure is continuously maintained in each space.

Basic heat load calculations for a small hot-water heating system are given in Example 23-2, and methods for designing warm-air, steam, and hot-water central systems are detailed in Chapters 12, 13, and 15, respectively, of the *ASHRAE Handbook: Systems* [14].

Example 23-2
Design of a Heating System

Problem: Calculate the pumping station heating load and select a hot-water boiler for the water pumping station in Example 23-1.

Solution: The method of calculating the heat loss for the pump room is shown in Example 23-1. Use the same method to calculate the heat loss for other rooms and sum all of the losses to arrive at the total heat loss for the structure.

Because perimeter heat is desirable to offset wall heat losses and to control the temperature in several spaces independently, a system of hot-water heat using a gas-fired boiler is chosen. Natural gas is preferred over fuel oil or electric heat at this site primarily because (1) it is available at a firm rate; (2) it meets the fuel needs for the building heating system; (3) it can power the required standby generator; and hence, (4) it offers low initial cost, simple controls, and low maintenance. A gravity-fired, cast-iron boiler is selected for its simplicity and expected long life (refer to AMCA No. 99 Standards, p. 24.3).

To minimize the danger of freezing in the event of a power or control outage, a 20% solution of ethylene glycol in water is used for the heating medium (see p. 15.19, AMCA No. 99 Standards). Combustion air is supplied by gravity flow at a net face velocity of 200 ft/min to the boiler room through a wall louver, which is sized to pass 12 times more air than the natural gas to be burned. This quantity of air is 20% greater than the stoichiometric volume for natural gas with a heat content of 1000 Btu/ft^3 (high heat value).

The current winter heat gain and loss in British thermal units per hour (for the pump room only) are summarized as follows:

Three 200-hp pumps running (maximum):	
Total winter pump room heat gain (from Example 23-1, part 3)	+224,301 Btu/h
Less structure heat loss (from Example 23-1, part 2)	−140,011
Excess heat to be removed	+84,290
One 200-hp pump running:	
Heat gain from one pump only: 169,667/3	+56,556
Auxiliary motor heat: 25,070 × 0.5	+12,535

Lights and people	+18,721
Total internal heat gain	+87,812
Structure heat loss (Example 23–1, part 2)	−140,011
Minimum ventilation load: 1.08 (−1000) 70 (see below)	−75,600
Total internal heat gain	−87,812
Operating net heat deficiency	−127,799

With all pumps off, the station not operating, no ventilation, no lights, and no people, the building heat loss is 140,011 Btu/h.

The minimum outside air rate for ventilating this city water pumping station was selected as 1000 ft^3/min, or roughly double the calculated infiltration. The ventilation air heating load, H, in British thermal units per hour is

$$H = W \times c \times \Delta T \tag{23-6}$$

where W is the weight of air in pounds per hour ($60\gamma ft^3/min = 1.08\,lb/h$), c is the specific heat in British thermal units per pound-degree (0.2395 from Table A-3), and ΔT is the difference between outside and inside temperatures in degrees Fahrenheit. Note that W can be expressed as $Q \times 60 \times \gamma$, where Q is the flow rate in cubic feet per minute and γ is the specific weight of air in pounds per cubic feet (0.0752 at sea level from Table A-7). Collecting and multiplying the constants gives

$$H = 1.08Q \times \Delta T \tag{23-7}$$

The outside air (ventilation) heat load, H_{OA}, in British thermal units per hour is

$$H_{OA} = 1.08\ (-1000)[60 - (-10)] = -75{,}600$$

Heat equal to the greatest operating net heat deficiency (127,799 Btu/h from the previous calculations when only one pump is running) must be added to the pump room to maintain the design temperature. Heating equipment capacity should be adequate to maintain room temperature at 45°F or higher under any condition.

$$\text{Boiler input} = \frac{(HD + H_{OA})1.2}{0.8} \tag{23-8}$$

where HD is the net heat deficiency in British thermal units per hour, 1.2 is a "piping and pickup" allowance for generated heat not conveyed to the heated space and for extra capacity for quicker warmup from a reduced setback temperature, and 0.8 is the boiler thermal efficiency. Note that flue gas heat loss (including heating of the combustion air) is included in the thermal efficiency factor. Calculations (in Btu/h) for the boiler size are:

Operating pump room net heat deficiency (1 pump)	−127,799
Calculated remainder of building heat loss (not shown)	−77,201
Minimum ventilation heat load	−75,600
Total building heating load	−280,600
Boiler input 280,600(1.2)/0.8	420,900

Use the next larger standard boiler with an input of 450,000 Btu/h.

Required combustion air (12 times the gas to be burned) is

$$\frac{(450{,}000\,\text{Btu/h})(12)}{(1000\,\text{Btu/ft}^3)(60\,\text{min/h})} = 90\,\text{ft}^3/\text{min}$$

The louver face area is

$$(90\,\text{ft}^3/\text{min})/(200 \times 0.25 \text{ free area}) = 1.8\,\text{ft}^2$$

Specify an automatic vent damper in the boiler flue to minimize convection losses from the hot boiler when the burner is not firing (see p. 16.5 of the *ASHRAE Handbook: Equipment* [15]).

Extend a Type B double-wall or insulated full-size steel flue through the roof to a height that is at least 2 ft above a parapet (or other obstruction within 10 ft horizontally). Provide a ventilated sleeve and weatherproof flashing at the roof and an approved weatherproof flue cap at the top.

Additional Required Design Steps

Other design steps are necessary to complete Example 23-2.

Size the Auxiliary Heat-Transfer Equipment

Select finned radiation, a unit heater, or a cabinet heater with the capacity to offset the calculated heat loss for each room. Base the calculation on an average water temperature of 93°C (200°F) and a temperature drop of 11°C (20°F) (see pp. 9.3 and 27.4 of the *ASHRAE Handbook: Equipment* [15] and p. 30.5 of the *ASHRAE Handbook: Systems* [14]).

Water Flow

Calculate the required heating water flow for each heating unit from Equation 23-9. In SI units,

$$Q = \frac{\text{Heat load}}{4184(\Delta T)} \tag{23-9a}$$

where Q is liters per second, the heat load is in kilowatts, ΔT is the temperature difference in degrees Celsius, and 4184 is the specific heat times mass in watt-seconds per kilogram. In U.S. customary units,

$$Q = \frac{\text{Heat load}}{500(\Delta T)} \tag{23-9b}$$

where Q is in gallons per minute, the heat load is in British thermal units per hour, 500 is $8.34\,\text{lb}_\text{m}/\text{gal} \times 60\,\text{min/h}$, and ΔT is the temperature difference in degrees Fahrenheit.

Plans

Sketch the proposed heating circuits including the boiler, the pump, the compression tank, and the supply and return piping for each heating unit. A reversed return piping arrangement (which gives equal circuit lengths through all heating elements) is desirable to simplify the balancing of system water flow (see p. 16.3 of the *ASHRAE Handbook: Systems* [14]). Connecting the compression tank to the system main pipe just upstream of the heating pumps ensures that the pumping head is positive throughout the system, and it prevents air from leaking into the system, thereby causing air binding (flow impedance due to the accumulation of air at high points). Any leaks can be located by water drips.

Sizing the Heating Pipe

Size the heating water piping based on the cumulative flow rate for each section of piping within a velocity range of 1.2–2.1 m/s (4–7 ft/s) and an approximate flow resistance of 0.4 kPa/m (4 ft WC/100 ft) of pipe. Begin by sizing the piping at the most remote unit supplied and at the first connection to a reverse return main and, accumulating the flow rate progressively, work back to the pump through each main.

Pressure Drop

Total the calculated pressure drops through the longest circuit of piping, including the fittings, valves, coils, and other system components (see p. 34.1 of the *ASHRAE Handbook of Fundamentals* [12]).

Heating Pump

Select the heating pump needed to deliver the total cumulative flow rate against the total calculated flow resistance by using a pump manufacturer's catalog data (see p. 30.5 of AMCA No. 99 Standards). The pump should be located downstream of the boiler with the compression tank connection at the pump inlet to avoid adding the pump head to the static pressure on the boiler.

Air Elimination

An air separator with an automatic float-type air relief valve should be connected on the discharge side of the pump. Manual or automatic air vents should also be located at the high points of the piping system and at each piping drop in the direction of flow. It is important that all air be removed from the heating water to prevent air binding and to minimize internal corrosion.

Water Volume

Calculate the total water volume in the system by adding the individual volumes of piping sections, coils, convectors, boiler, air separator, and compression tank (see p. 15.40 of the *ASHRAE Handbook: Applications* [16]).

Compression Tank

A compression tank is needed to accommodate the expanding volume of system water as it is heated. A diaphragm type of tank, in which a flexible elastomeric diaphragm separates a closed air compression space from the heating water, is recommended. The flexing diaphragm compresses the contained air as the water expands and vice versa. Size the compression tank to accept expansion of the system water as it heats from, for example, 10°C (50°F) to the design supply temperature without exceeding the allowable pressure setting of the boiler pressure relief valve, which is 186 kPa (27 lb/in.^{2} ga) for a low-pressure boiler rated for 207 kPa (30 lb/in.^{2} ga) operating pressure.

Maintenance

On a circuit diagram, locate the isolating, balancing, drain, and control valves as well as thermometers and other accessories needed for maintenance and service of the system.

Gas Piping

In accordance with the chapter on pipe sizing in the *ASHRAE Handbook of Fundamentals* [12], size the gas piping for a gas flow equal to the boiler input rating at the pressure required by the boiler burner. This pressure is normally about 1.0 to 1.3 kPa (4 to 6 in. WC) for a gravity-fired natural gas burner. The gas piping installation should be specified to meet NFPA Standard 54.

Safety

Select and specify required operating and safety controls for the hot water heating system (see p. 24.3 of AMCA No. 99 Standards). The following controls are typically included:

- Boiler water temperature operating control to operate the burner.
- Boiler water high-temperature limit control with manual reset to shut off the burner upon excessive temperature.
- Boiler water low-level cutoff control to shut off the burner if the system water falls to an unsafe level.
- Boiler overpressure relief valve rated for maximum fuel input to the burner in accordance with ASME Code VIII.
- Boiler water make-up provision. If make-up water is from a potable source, it must be connected (1) through a backflow preventer or a break tank in accordance with the applicable code, and (2) through a pressure-reducing station that includes a gate valve, strainer, and pressure-reducing valve. Set the pressure-reducing valve to maintain the minimum boiler pressure needed to give at least 1.5 m (5 ft WC) pressure at the highest point in the system piping with the heating pumps off. This minimum pressure ensures positive venting of air from the system.

Chemical Water Treatment

A means of introducing water treatment chemicals into the system is needed to prevent internal corrosion or scale formation. The simplest device is a chemical feed pot with isolating and drain valves. This permits adding antifreeze and corrosion inhibitors when needed as shown by testing heating water samples. The pH of the heating system water should be maintained at about 9.0.

For steam boilers having direct steam usage and resulting condensate loss and make-up, an automatic chemical feed system should be used to supply the required chemicals at the rate recommended from tests conducted by a competent chemical company to prevent boiler corrosion and scaling.

Temperature Controls

Select and specify automatic temperature controls to operate the heating system (see p. 15.16 of the *ASHRAE Handbook: Applications* [16]).

23-10. Design of Building Cooling Systems

The removal of excess heat can be accomplished by moving outdoor air through the building as long as the temperature of the outdoor air is sufficiently below the design indoor temperature. If not, evaporative cooling can reduce the dry-bulb temperature if the initial wet-bulb temperature of the air is well below the desired indoor temperature. Refrigeration, either as an additional stage of the evaporative cooling process or as the sole cooling method, may be necessary if still more cooling is required.

Objectives other than cooling the air may include reducing its moisture content, minimizing its corrosiveness and odor, removing dust and aerosols, and achieving an acceptable air movement and sound level in the conditioned space. The term "air conditioning" encompasses all aspects of air enhancement. The extent of the air treatment applied varies with the requirements of the application. For example, air conditioning for an industrial process may be quite different from air conditioning for human comfort. Where both types of air conditioning are involved, some compromises may be necessary.

Evaporative Cooling

Evaporative air cooling (especially of equipment spaces) is often practical where the design summer wet-bulb temperature is lower than about 24°C (75°F), as is true in most Western states. Wet-bulb temperatures for many locations are listed in the *ASHRAE Handbook of Fundamentals* [12]. The heat content of air at any combination of dry-bulb and wet-bulb temperatures can be read from a psychometric chart for the site altitude. Humid air (with its higher wet-bulb temperature) has a higher heat content than dry air at the same dry-bulb temperature and is less susceptible to evaporative cooling. An indirect evaporative cooling first stage, followed by direct evaporation (called "indirect–direct"), is a more effective method. A high-limit humidistat or some other means of stopping or reducing evaporation (thereby also limiting the cooling effect) may be necessary to prevent excessive humidity.

Due to the smaller temperature differential between supply and room air temperatures, the evaporatively cooled airflow rate may often need to be 20 to 30 air changes per hour as compared with 8 to 12 changes for refrigerated cooling. Such higher rates require larger fans and ducts and better air distribution than the lower rates. The larger airflows also tend to create more noise and drafts. Recirculation of evaporatively cooled air is not practical due to the resulting buildup of humidity.

Example 23-3
Design of an Evaporative Cooling System

Problem: Design an evaporative cooling system for the unoccupied pump room of Example 23-1. Assume that a bridge crane in the pump room leaves too little headroom for ductwork below the roof, but that roof-mounted equipment is aesthetically acceptable.

Solution: As shown in part (4) of Example 23-1, the sensible heat load to be removed after the future increase in station capacity is 581,263 Btu/h. An indirect–direct evaporative cooling system is used to obtain maximum cooling with the lowest air supply quantity.

Supply air. The wet-bulb depression (the difference between outdoor dry-bulb and wet-bulb temperatures) is 94°F − 74°F = 20°F. Manufacturer's data for an indirect evaporative cooler shows that a unit rated at 10,000 ft^3/min can directly cool the waste airstream to within 5 degrees (called the "approach") of its wet-bulb temperature, or to 79°F dry bulb. The evaporatively cooled waste airstream goes through one path of the indirect cooler and flows across the supply airstream in the other path. According to the manufacturer's literature, the supply air is thereby cooled to 85°F dry bulb, and its wet-bulb temperature is reduced to 71.3°F. In passing through the second (direct) stage cooler, the supply air dry-bulb temperature is reduced to 76.3°F, or within 5 degrees of its new wet-bulb temperature. Simultaneously, however, the addition of moisture increases its total heat content by an equivalent amount, which leaves its wet- bulb temperature (the measure of total heat content) unchanged at 71.3°F. Direct evaporative cooling is an adiabatic process (or one in which total heat content remains unchanged).

Supply air equipment. The dry-bulb temperature difference between the evaporatively cooled supply air and pump room design temperature is 105°F − 76.3°F = 28.7°F. The airflow rate required to remove the calculated future pump room heat gain is, from Equation 23-lb,

$$Q = \frac{581{,}263}{28.7 \times 1.08} = 18{,}753\,\mathrm{ft}^3/\mathrm{min}$$

Two indirect–direct cooling units will be needed, one initially and another in the future, each supplying 9500 ft^3/min to the four diffusers located symmetrically in the room length. Each supply fan is selected for a capacity of 9500 ft^3/min at 1.25 in. WC. Using Equation 23-2b, the size of the motor is

$$\mathrm{bhp} = \frac{9500 \times 1.25}{6356 \times 0.65} = 2.9$$

Use a 5-hp, 3-phase, 60-Hz motor. From a manufacturer's literature, select four round supply diffusers to deliver 4750 ft^3/min each that are 42 in. in diameter and have adjustable cones to give a downward projection. Specify installation of the first rooftop unit on a prefabricated curb with weatherproof counterflashing for the roofing. Specify a similar prefabricated curb with an insulated weatherproof cap for an equal fan unit to be installed in the future, when the station pumps are increased.

Water supply. Pipe city water through a backflow preventer to preclude contamination of the city water supply. Connect a $\frac{3}{4}$-in. city waterline through a ball valve, strainer, and three-way solenoid drain valve within the building to the float valve on each cooler. Provide a solenoid drain valve between the cooler pan drain and the common drain line. The solenoid drain valves are activated by an outdoor thermostat and they shut off the water supply and drain the supply pipe and cooler pan when the outdoor temperature drops to 60°F. The make-up water float valve in each unit is set to a level slightly above the overflow standpipe to prevent the buildup of minerals from water evaporation by allowing a small amount of water to be continuously bled off during operation.

Air controls. Install control dampers at the inlet of each supply air evaporative cooler to permit air recirculation from the pump room ceiling in cold weather when at least one of the supply fans runs without evaporation. Enough outdoor air is admitted (under the control of a room thermostat) to remove any net heat gain. Insulate and weatherproof the supply unit casings and ductwork above the roof to minimize heat loss in cold weather.

Cooling controls. The supply fans are manually started. One could be shut off automatically by an outdoor thermostat set at 50°F, below which a single unit is sufficient. A space thermostat modulates the inlet mixing dampers of the operating unit(s). Motorized dampers at the two relief louvers modulate in conjunction with the outdoor dampers on the supply units. The relief louvers are sized on the basis of a 250 ft/min face velocity or 38 ft^2 each. When the outside air damper is fully open and the space temperature rises above 85°F, the evaporative system water solenoid valves are activated and the waste airstream fans are started. A discharge low-limit control in the supply air from each unit takes control of the mixing dampers if the supply air temperature drops below 45°F. A smoke detector in each supply airstream, as required by NFPA Standard 90A, shuts off the fans and energizes a remote alarm if combustion products are sensed. For an unattended station, provide pump room high- and low-temperature warning signals to an attended station.

23-11. Design of Refrigerated Cooling Systems

Both the temperature and the humidity of air affect the comfort of occupants because they both affect the rate at which the body loses heat. High humidity causes discomfort even at normal air temperatures, so both humidity and temperature are controlled in an air conditioning system. Because of its power demand and relatively high operating cost, refrigerated air conditioning is mainly used where (1) human comfort or reduced humidity are prime considerations, or (2) the maximum allowable ambient temperature for installed equipment cannot otherwise be maintained.

Thus, room comfort is affected by (1) "sensible" heat gain (from solar radiation, thermal transmission, internal motors, and lights or other dry heat sources), and (2) "latent" heat gain (from water evaporation and moist ventilation air). Both should be evaluated and controlled.

Different types of refrigerated air-conditioned equipment can be contrasted as follows:

- Packaged units versus custom-designed systems
- Direct expansion coils versus chilled-water coils
- Air-cooled refrigerant condensers versus evaporative or water-cooled types.

Heat Gain Calculations

Determine the maximum sum of simultaneous heat gains to the space or "peak heat gain" for the year. This maximum load is used to establish the required capacity of the refrigerating equipment. It is usually necessary to calculate the heat gains for air-conditioned spaces at more than a single time. Heat gain varies with the time of year, time of day, orientation and construction of the building, and the internal heat gains present (such as from occupants, motors, engines, and lights). The procedures for these calculations are outlined on p. 26.3 of the *ASHRAE Handbook of Fundamentals* [12].

Latent Heat Gain

Latent heat gain in pump rooms usually is of concern only if the pumped water temperature drops below the indoor air dew point temperature, which results in moisture condensation on piping, valves, and walls. In wastewater pumping stations, such occurrences usually are brief and infrequent. In city water stations, condensation can be a severe problem during the spring, which may make a separate dehumidification system advisable. Coating piping and valves with up to $\frac{1}{8}$ in. of a sorbent material minimizes the effects of condensation.

Humid outside air introduced for ventilation is a major source of latent heat. Standard air conditioning equipment readily handles that heat if the proportion of outside air is limited to about 10% of the total under peak load conditions. The extent of both sensible and latent ventilation loads can be determined by means of a psychometric chart or psychometric tables. For the detailed methods and the extensive data needed, see Chapter 26 of the *ASHRAE Handbook of Fundamentals* [12].

Supply Air

Supply air quantities for refrigerated air conditioning systems are based on the calculated sensible cooling load and an acceptable temperature differential—limited to a maximum of 17°C (30°F)—between supply air and room air. Air changes ranging from 8 to 12/h are typically required. The total air-handling capacity of a constant-volume air conditioner is the sum of the required design air quantities for all of the spaces supplied. For variable air-volume systems, however, peak loads for all of the various spaces supplied are unlikely to be concurrent due to differences in solar exposure or occupancy. Thus the required total air quantity is determined by the largest sum of simultaneous sensible heat gains in the spaces supplied.

Refrigeration Equipment

The design cooling load for the refrigeration equipment is the highest sum of simultaneous sensible and latent cooling loads, including ventilation air, as calculated for all of the air-conditioned spaces served. The capacity of the refrigeration compressor should be somewhat less than the maximum cooling load because any rise in space temperature under brief peak loads will be minimal. Compressor performance at partial load (which is most of the running time) is better if the compressor is not oversized.

Select the refrigeration compressors in conjunction with the cooling coils to obtain the proportion of sensible to latent cooling capacity best matching the load. Equipment suppliers can assist with the choice.

Air Conditioning Units

Air conditioning equipment is rated by its cooling capacity at industry standard conditions, and 1 ton of refrigeration is equal to 3.52 kW (12,000 Btu/h) cooling capacity, which is equivalent to melting 1 ton of ice.

The simplest and least expensive of the available types is usually a prepackaged air-cooled conditioner. Either it must be installed outdoors or it must circulate an outdoor airstream to which the heat removed by the refrigerant plus its heat of compression is rejected. Package air conditioners of this type are designated as (1) rooftop, (2) at grade pad-mounted, (3) through-the-wall, and (4) above-ceiling units. If lack of space precludes these types of air conditioners, a "split" system (separate air handling and remote condensing units) is necessary.

Cooling Coil Drain

Most refrigerated air conditioning coils sometimes condense moisture from the circulated air. The condensate must be conducted to a satisfactory disposal point. The condensate drain line should be at least 25 mm (1 in.) in diameter to minimize clogging. A trap should be installed in the drain line near the coil drain pan and sized to prevent air leakage at the design static pressure. The direct connection of the drain to a plumbing waste pipe is prohibited by code, so terminate the drainpipe to leave an air gap of at least 65 mm ($2\frac{1}{2}$ in.) above a floor drain.

Noise Considerations

Air conditioning equipment in or near occupied rooms or outside of a pumping station in a residential area must be acceptably quiet. Many cities have noise ordinances defining sound levels permitted at property lines. Most compressors and condenser fans are direct-driven, so choose a speed low enough to meet the allowable sound rating. However, supply and exhaust fan speeds must be adequate to develop the required static pressure. The fan with the highest efficiency is usually the quietest. Because sound level diminishes with distance, place fans as far from the occupied space (or from neighbors) as practical. Where a noise ordinance limit cannot be met by equipment selection or location, a masonry sound barrier or other sound-attenuating means may be needed (see Chapter 22).

The air velocity in ducts and outlets also contributes to the noise level. Duct liner, consisting of specially treated insulation attached to the inside duct walls, helps to meet room sound criteria. The sound level in an office, for example, should not exceed about 45 dBA (decibels on the A-scale of a sound meter).

Example 23-4
Design of a Refrigerated Cooling System

Problem: Design an air conditioning system for the control room and the present office for the pumping station shown in Figures 23-1 through 23-4. See Example 23-1 for the design conditions.

Solution: Calculate the heat gain for each room at the peak time, which is at 8 P.M. on July 21, because it takes solar heat an additional 4 h to penetrate the south wall and roof.

Load	Control room, Btu/h	Office, Btu/h
Roof	692	496
South wall	674	421
Interior wall	735	830
Interior glass	3,079	1,703
Infiltration	2,401	1,087
People (2)	1,300	650
Lights	2,457	1,536
Electrical equipment	8,400	—
Total heat gain	29,838	6,723

System selection. A "split system" (the refrigeration condensing unit remote from the air-handling unit) is selected because it fits the available space and meets aesthetic requirements. It is composed of an air-handling unit in the basement, an air-cooled refrigeration condensing

unit on the roof, return and supply air ductwork, and controls. The air-handling unit includes mixing dampers for outside and return air, filters, a direct-expansion cooling coil, a hot-water heating coil, and the supply fan, all enclosed in a metal casing. The supply fan capacity is based on meeting the total sensible cooling load rounded to 36,500 Btu/h at a temperature difference (between the supply and room air) of approximately 20°F. From Equation 23-lb, the required airflow is

$$Q = \frac{36{,}500}{20 \times 1.08} = 1690\,\mathrm{ft}^3/\mathrm{min}$$

which is divided into 1380 and 310 $\mathrm{ft}^3/\mathrm{min}$ for the two rooms, respectively. The thermostat, which controls operation of the compressor and the heating coil valve in sequence, is located in the control room. The cooling load there is continuous, although it varies with the number of pumps operating. A self-contained thermostatically controlled supply diffuser in the office varies the air quantity supplied to match the actual load. In winter, a heating thermostat in the office modulates heating water flow to the baseboard convector to maintain a room temperature of 70°F. The damper in the outside air duct is adjusted so that the minimum make-up air quantity matches the toilet room exhaust rate.

Basement dehumidification. Assume the pumped water temperature in the basement mains and adjacent reservoir varies from about 40°F in winter to about 75°F in summer. In the spring, water temperature is below the dew point temperature of the outdoor ventilation air and would tend to condense moisture from the ambient air on the piping. To prevent rusting and wet floors caused by such condensation, install three small refrigerated dehumidifiers in the basement to reduce the air dew point to below the cold surface temperatures. In cold weather, warm dry air from the upper level automatically is circulated through the basement by a transfer fan. For additional information on dehumidification, see p. 41.8 of AMCA No. 99 Standards.

23-12. References

1. *Criteria for a Recommended Standard, Working in Confined Spaces,* DHEW (NIOSH) Publication No. 80-106, U.S. Department of Health, Education and Welfare, U.S. Government Printing Office, Washington, DC (December 1979).
2. *Papillon Creek Wastewater Treatment Plant, Omaha, Nebraska,* HETA 83-440-1537, NIOSH, U.S. Department of Health and Human Services, Atlanta, GA (1984).
3. *Occupational Safety and Health Standards, General Industry Standards and Interpretations,* Vol. 1, OSHA 2077, U.S. Department of Labor, Superintendent of Documents, Washington, DC (1977).
4. Ten-State Standards, *Recommended Standards for Sewage Works,* Great Lakes–Upper Mississippi Board of Sanitary Engineers, Health Education Service, Inc., Albany, NY (updated periodically).
5. WEF, *Operation of Wastewater Treatment Plants,* Vol. I—MOP-11, Water Environment Federation, Washington, DC (1996).
6. WEF and ASCE, *Odor Control in Wastewater Treatment Plants*—MOP-22. Water Environment Federation, Washington, DC (1995).
7. ASCE, *Sulfide in Wastewater Collection and Treatment Systems,* Manuals and Reports on Engineering Practice No. 69, American Society of Civil Engineers, New York (1989).
8. Wolstenholme, P., et al., "Comprehensive odor and VOC performance tests on biofilters." *Proceedings*, Odor and VOC Emission Control Specialty Conference in Jacksonville, FL, Water Environment Federation (April 1994).
9. *Design Standards for Controlled Environments and Selection of Gas-Phased Filtration Systems*, Technical Brochure 300, Purafil, Inc., Doraville, GA (undated).
10. *Design of Municipal Wastewater Treatment Plants,* MOP 8, Joint Committee of the Water Environment Federation and the American Society of Civil Engineers, New York (1992).
11. *Chlorine Manual,* 4th ed. The Chlorine Institute, Washington, DC (1976).
12. *ASHRAE Handbook of Fundamentals,* American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, GA (1985, updated quadrennially).
13. *Engineering Weather Data,* Departments of the Air Force, the Army, and the Navy, U.S. Government Printing Office, Washington, DC (1978).
14. *ASHRAE Handbook: Systems,* American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, GA (1982, updated quadrennially).
15. *ASHRAE Handbook: Equipment,* American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, GA (1984, updated quadrennially).
16. *ASHRAE Handbook: Applications,* American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, GA (1983, updated quadrennially).

23-13. Supplementary Reading

1. *Dodge Construction System Costs,* McGraw-Hill, Princeton, NJ (updated annually).
2. *Means Mechanical Cost Data,* Robert Snow Means Co., Inc., Kingston, MA (updated annually).
3. Strock, C., and R. L. Koral, *Handbook of Air Conditioning, Heating and Ventilating,* Industrial Press, New York (1965).
4. *Pocket Manual on Heating,* Dunham Bush, Inc., West Hartford, CT (1978).

Chapter 24

Designing for Easy Operation and Low Maintenance

BAYARD E. BOSSERMAN II
GEORGE JORGENSEN
GARY ISAAC
ROBERT L. SANKS

CONTRIBUTORS
Richard (Rick) Arbour
Nicholas J. Arhontes
Roger J. Cronin
Ronald P. Kettle
Earle C. Smith
Dale Stiller
John A. Redner

This chapter deals with those design considerations that reduce life-cycle costs by facilitating operation and maintenance (O&M) and by improving labor efficiency. The term "life-cycle" is used to define the cost of owning an asset from planning through decommissioning. The ongoing, recurring costs are operational costs for utilities; electrical energy, water, gas, labor, and material costs for performing preventive, predictive, and corrective maintenance. Ongoing costs also include the periodic rehabilitation and replacement costs for the individual assets. The cost of O&M varies from about 25 to 50% of the total life-cycle cost (Table 12-2) for small pumps to about 10% for large ones [1, 2, 3]. Although designing for O&M will result in a higher level of service to the community at a lower life-cycle cost, regulatory compliance is another important consideration (see Subsection "Cost of Ownership" in Section 12-10).

To provide background material on the many issues involved in designing for easy O&M, the opinions of many operators with an aggregate of two centuries of experience with wastewater pumping stations are given in Section 24-9. Furthermore, distillation of knowledge gained from interviews with 15 public sanitation agencies that operate more than 2700 pumping stations and site visits to more than 50 of these stations is given in Section 24-10. Chapters 12, 25, and 26 also contain valuable advice.

Although the chapter is primarily directed toward wastewater pumping, much of it also applies to water pumping and to sludge pumping.

24-1. Site Selection

When selecting a site for a pumping station, consider access, not only for operating personnel but also for equipment. Equipment can vary from a pickup truck for small stations to a crane for large stations. Weather conditions may influence access requirements. For example, road gradients may be limited in areas of heavy snowfall.

Route the force main to minimize the need for air and vacuum valves and the possibility of column

separation, even if it means relocating the pumping station. Avoid siting pumping stations in areas prone to flooding or where there is a potential for washout of force mains.

Consider O&M staff safety regarding:

- Traffic and traffic control
- Material handing for removal and replacement of equipment
- Access to wet well with equipment for cleaning the wet well (vacuum and high-pressure jet machines)
- Removal and disposal of screenings from manual or automatic bar screens
- Confined space access requirements
- Electrical installations in hazardous locations

The design should also include:

- Adequate room to install isolation valves and a means to bypass the station in the event of failure.
- A driveway that is either drive-through or has an interior turnaround with a minimum of 15 m (50 ft) curb-to-curb turning radius. Tee turnarounds are also acceptable.
- Adequate access for boom trucks, especially at submersible stations.
- Adequate room for siting portable generators and emergency pumping systems.
- Space to plow and store snow on-site, if needed.
- Good drainage away from the facility, but no excessive slopes.
- A slight crown or a slight slope to all exterior concrete slabs to prevent ponding of water on sidewalks or wet well roof slabs over submersible stations, for example.
- Bollards and/or other protection around gas lines or other critical structures emerging from pavement outside the station. A minimum of exterior lighting, particularly where the site is fenced. For example, light entrance doors only.
- Permanent easement access for O&M equipment and inspection of force mains.
- Site security such as fences and gates.
- Considerations of noise, odor, vibration, or other nuisances.
- Considerations of indoor space for chemical storage tanks for odor/corrosion control.
- Consideration of access to the force main.

24-2. Landscaping

Landscaping the pumping station to fit or to blend into the surroundings is desirable, and so is designing the landscaping for low maintenance by using ornamental woody perennials and ground cover that require little attention. Install automatic sprinkler systems activated by moisture probes and a timer. There are now much higher expectations from communities regarding the appearance of pumping stations that require the design to incorporate aesthetic features, again with consideration of the ultimate neighborhood appearance.

24-3. Hydraulics

The hydraulics of pumping systems are covered in several other chapters, but it is worth repeating that if the analyses are improper, inadequate, or do not include *all* of the operating conditions (including start-up, shut-down, and power failure), excessive maintenance may result. Examples of operational problems leading to excessive maintenance include:

- Solids deposition and grease accumulation in wet wells and force mains.
- Poor pump selection that causes pumps to operate too far from the BEP.
- Vortexing resulting in pump cavitation.
- Release of H_2S resulting in corrosion and odor.
- Entrained air resulting in loss of capacity, head, efficiency, pump damage, and possibly pump prime.
- Accumulation of solids in risers ahead of check valves.
- Cavitation damage to valves.
- Turbulent flow affecting flowmeter accuracy.
- Excessive pump vibration.
- Motor overload.

Select wastewater pumps for their maximum efficiency (BEP) during normal, dry weather flow, but keep peak flow conditions within the AOR. Most wastewater collection systems have significant inflow and infiltration (I/I) in the upstream gravity sewer system. I/I vary depending on the condition of the sewers, rainfall, groundwater level, spring runoff, and other factors, and they can result in flows that exceed the station firm capacity and in static heads excessively reduced due to high wet well levels. In stations that discharge into force mains serving more than one pumping station, pumps must overcome higher dynamic heads if all stations are operating at capacity due to wet weather events. In both examples, the station H-Q curve changes and creates operational problems that may lead to equipment damage or failure.

Consider two-stage pumping systems when heads are higher than about 60 m (200 ft). Both stages must have matching performance curves. Controls must be configured never to operate one stage without the other, and operating speeds must be matched to prevent driving one stage into a portion of the H-Q curve where damage is likely.

Design so that prime is reliably maintained at all times in pumps operating on a suction lift. Balance reliability with the cost over the entire life-cycle. Priming devices are likely to be required for dry pit pumps in trench-type pumping stations because of the large volumes of air in suction pipes when pumps lose prime during cleaning. Priming devices can be active (e.g., a vacuum pump) or passive (opening a valve to let pressurized air escape) and automatic (preferred) or manually operated.

When pumping stations are arranged in series (one pumping into another pump station wet well), make arrangements for the effect of a short-term (depending on the anticipated response time) shut-down of either station upon the other. For example, require a SCADA interlock to shut down the lower station if the upper shuts down. Storage in the sewer can be considered in this unlikely event, but make sure all circumstances are covered.

Head-Capacity Curves

A careful study of the full range of conditions under which pumping will occur is essential so that the proper pump, impeller, and motor sizes are selected. Such a study is particularly needed where a battery of pumps discharges into a manifold. As additional pumps come on line, the total flow and head increase, but as the head increases, the flow from each individual pump decreases. In a smaller pump (or one with a lower shutoff head than the others), the discharge may drop to zero, which can cause excessive radial loads. If the analysis is incomplete, the pumping conditions may be outside the limits of the pump curve for a particular pump (see Sections 10-6 and 10-8). Variable-speed (V/S) pump system hydraulics, in particular, require a very thorough analysis, especially in wastewater stations and where large variations in static head can occur. The effectiveness of V/S drives depends on static head and the shape of the pump curve.

Pressure Gauges

Taps (in bronze pipe saddles secured to the pipe with double bronze or stainless steel straps) for screw connections of pressure gauges are useful on both the intake and discharge pipelines. Permanently mounted pressure gauges can be installed, but portable gauges, which can be used when needed, are usually better because they can be easily recalibrated and can be used for many locations (see Chapter 20). Fixed gauges are subject to equipment vibration and, unless recalibrated periodically, do not remain reliable. The maintenance schedule should require pressure readings at regular intervals (e.g., monthly) along with simultaneous flowmeter readings. These records can be used to detect abnormalities such as clogging of the force main or impeller wear. Suction and discharge pressure gauges are useful for determining individual and multiple pump operating heads and capacities, and for diagnostic information when operators are troubleshooting performance problems. Avoid using pipe nipple materials subject to corrosion (e.g., galvanized pipe). Include a method for the operator to unclog taps that are subject to plugging when not in use.

Flow Measurement

To check the efficiency of the pumping system, a flowmeter in conjunction with the pressure gauges facilitates checking the power consumed against the water output. It is also a good idea to make it easy to calibrate the flowmeter in place; for water systems, a means of inserting a pitot tube is a good way to do so. Time–volume measurements using the wet well, a storage tank, or a clear well are reliable and can often be used. The use of tracers (dye dilution) is accurate, almost universally applicable, and easy to use if taps (one to introduce the tracer and another at a point far downstream for withdrawing samples) are installed (see Section 3-9).

The lengths of straight inlet and outlet pipe recommended by the manufacturer must be observed for all types of meters; otherwise, the meter readings will probably be erroneous. The errors increase exponentially as the inlet and outlet pipes are shortened.

Flow measurement devices in pumping stations can be permanently mounted pressure pipe devices, portable "clamp-on" units, or even open channel flumes. Meters are now available that can be installed using a "hot tap" in an existing discharge pipe. Evaluate costs, accuracy, installation constraints and operation and maintenance required—cleaning, calibration, and component replacement. Make it easy and safe to access the meter for cleaning, calibration, or replacement (see Figure 20-12, for example).

Flow data, especially real-time data, are another important tool useful for monitoring flow conditions and pump performance. Flowmeter data, easily captured, should be telemetered by SCADA systems where SCADA is available. Flow data should be monitored and trends established for historical flow baselines on hourly, daily, weekly, and monthly bases. Flow data that varies from historical baselines should then be investigated to determine the cause of the change.

Surge Control

Water hammer need not instantly rupture a pipe to be damaging. Surges that continue for many years gradually destroy a pipe's integrity through material fatigue, so surge control is required in most pumping stations to prevent damage to the piping system. High pressure transients can result in the gradual structural failure of pump volutes, valve housings, and piping that, in turn, results in catastrophic flooding. Piping failure can also occur in the force main with similar results.

As explained in Chapter 7, if the flow rate exceeds about 30 L/s (500 gal/min) and the total dynamic head exceeds 12 or 15 m (40 or 50 ft), a surge control analysis is needed. Computer-based analysis for high pressure transients is now commonly performed even on smaller systems. If air relief valves or combination air release/vacuum relief valves are necessary, install them in pairs with a multiport isolation valve (see Figure 7-2) that prevents simultaneous closure of both valves for obtaining a higher level of reliability. See Figure 5-17 for reliable valves and Section 7-1 for strategy.

Several methods of surge control can be employed:

- Pump-control valves (which are popular) for start-up with controlled closure on shut-down. Power failure could be acceptable under some installation-specific instances.
- Motor starters that ramp up and down when pumps are started or stopped prevent the continual pounding of transients. However, the use of such motor starters does not prevent surge when power fails.
- Flywheels to control the rates of acceleration or decay of liquid system velocities.
- Pressure relief valves, installed in the pump station and piped to the wet well, can also be effective under specific installation situations.

These and other surge control methods are discussed at length in Chapter 7. Neither of the first two methods, however, provides surge control for power failure. If surge can occur in the intake pipe, consider surge control both in the discharge and in the upstream conduit, especially if it is greater in diameter than 1.25 m (4 ft) and greater in length than 3 km (2 mi).

Air–Vacuum Release Valves

Try to avoid the need for air–vacuum release valves. If the need is imperative, study Section 7-1 for strategy and see Figures 5-16, 5-17, and 7-2. Always install the valves in pairs on a multiport valve that allows isolating either valve but not both simultaneously. Install the multiport valve at the springline to ensure a cushion of air upon column separation. See Table B-9 for the velocity required to scour the air pocket above the springline. As grease floats, the springline location avoids most of the grease. If the valves are to be flushed, install quick-connects.

Place the valves in a structure designed to prevent any possibility of flooding whatever. (A submerged valve is utterly useless.) There must be adequate access to the site for the equipment for O&M and easy access to the valves for replacement or flushing.

A frequent point of failure is the pipe connection between the valve and force main due to corrosion and/or fatigue. Right-of-way acquisition includes the type of equipment that will be used for O&M of the valves.

Consider the odor and noise when valves discharge air as part of the normal operating cycle. Valves frequently fail in the open position, resulting in a sanitary sewer overflow, so consider the health and environmental impacts of the overflow and the ability of O&M crews to reach the site quickly and clean up the mess.

Wet Wells and Other Pump Intakes

- Wet wells and other pump intakes are an integral part of pump performance. Poorly designed sumps can cause vibration, cavitation, excessive maintenance, and even the early destruction of pumps. Pay heed to Chapter 12.
- If there are unusual conditions or if the pumps or the total station flow rates are very large, model studies should be used to optimize the geometry and the need for baffles and suppression of swirling and vortices.
- Make sure that the wet well has enough volume and/or distance coaxially from the inlet conduit to dispel any turbulence from entering water. If

practical, provide enough storage in the wet well and influent sewer so that short-term (e.g., 1 h) power outages can, at least for dry-weather flows, occur without damage.

- Also make sure that there are no general currents, regardless of which pumps are running, to induce swirling in pump suction inlets.
- Heat generated by inrush electric current during start-ups can damage motors. The sump should be of adequate size to limit the starts per hour to the manufacturer's recommendation.

An easy graphical way to determine starts per hour and length of resting times for pump motors is illustrated in Example 12-2.

Clean Water Intakes

Designers of wet wells must avoid following obsolete literature such as Hydraulic Institute Standards of 1983 [4] or other standards published before ANSI/HI 9.8-1998. The older designs were limited and, furthermore, especially unsuitable for water containing either settleable or floating debris. Note that:

- Trench-type wet wells are excellent for clean water wet wells. See Figure 9.8.6 in ANSI/HI 9.8. The curved ramp in Figure 12-2 is unnecessary, its omission saves footprint area, and the confinement ensures there will be no crosscurrents. There are floor and wall vortices, however.
- Cones with six vanes under the pumps eliminate swirling and floor vortices.
- Fillets eliminate wall vortices.

Wastewater and Storm Water Intakes

It is no longer acceptable to design wet wells in which half of the basin is continued in service while the other half is dewatered and cleaned manually. Cleaning must be user-friendly. One type of suitable wet well is the trench-type.

- Trench-type wet wells with a curved ramp at the entrance (Examples 12-1 and 12-2) to induce a high velocity along the floor when dewatered are ideal. They have been studied extensively in a number of laboratories and many trench-type wet wells—even ones exceeding 4.4 m^3/s (100 Mgal/d)—have been constructed with complete satisfaction. See Sections 12-3 to 12-7 for recommendations for design.
- Include hose outlets for 38-mm (1.5-in.) wash water hoses for cleaning the walls of the wet well. For small wet wells, a 25-mm (1-in.) hose is probably adequate. Plan for a flow rate of 1.5–4.5 L/s (25–70 gal/min) at a pressure of about 600 kPa (90 $lb/in.^2$) at the nozzle—the higher flow rates for large wet wells and the smaller flow rates for small ones. If water at that pressure is not available, install a small tank with an air break and pressurize the water with a regenerative turbine (Figure 11-33) or a multistage centrifugal pump (Figure 11-34).
- Much higher pressures can be used with less water. One device is the high-pressure jet machine used for cleaning gravity sewers, but there must be access for the use of the jet machine.
- Make access for washing easy; a sidewalk beside the wet well is best. A catwalk inside the wet well is almost as good. For either method, the wet well must be ventilated.
- Vortices form on the floor under the pump intakes and at the sides of the trench beside the suction bell. They are eliminated by fillets at the wall and a flow splitter along the floor. Unless trenches are at least 0.9 m (3 ft) wide, they are too narrow for workers to install those features, and they can be omitted. None of the early trench-type wet wells (constructed before 1970) had them, and their omission has caused no problems. Nevertheless, the requirements of ANSI/HI 9.8 Section 9.8.5.6 are not satisfied unless fillets and flow splitter are installed, and if the trench is 1.2 m (4 ft) wide or more, these facilities should be added.
- Flow splitters should be carried to the top of the ramp where the water velocity is low. Splitters terminating at the base of the ramp cause the high velocity of the water to burst into spray, lose much of its energy, and never reform into the swift, coherent flow that quickly moves deposited solids to the last pump.
- The noses of flow splitters (where they taper from full-size to zero) can be short (one-half the length of the upper ramp curve) for V/S pumps. For C/S pumps where the upstream inlet is a steep approach pipe, the nose must be gently tapered to avoid disrupting the supercritical flow; a length from the top of the ramp to the mid-height of the ramp is appropriate.

Small, Round Wet Wells with Submersible Pumps

- Round wet wells with flat bottoms are usually impossible to clean easily. For one exception, see Figure 17-22.
- Make the bottom cone-shaped with a minimum slope of about 60 degrees with the sides clearing

the pump volutes by about 100 mm (4 in.), as shown in Figure 12-31. Sludge is then pumped out every time a pump is turned on.

- Program the pumps to draw the water as low as possible (to the top of the volutes) once or twice per week to eject scum automatically.
- Never allow wastewater to fall into the pool below. Use an approach pipe in the plane of the pump center with its invert somewhat above the pump volute. Prevent turbulence in the wet well by confining the approach pipe's hydraulic jump within the approach pipe. Do so by setting the normal LWL above the approach pipe invert by about 60 to 65% of the pipe diameter, as shown in Example 12-2.

Cleaning

Whatever the sump design, periodic cleaning is necessary. Grease accumulates on the water surface and on the walls between LWL and HWL, and (depending on severity) can form a thick blanket on the water surface. Debris (rags, plastic items, stringy material) also accumulates on the floor and on the water surface, and it can foul bubbler pipes, floats, and the slide rails of submersible pumps. The design must address the need for periodic O&M in the wet well area. For self-cleaning (trench-type or round sumps with conical bottoms), pump-down removes solids and scum from surface and floor.

Grease needs to be washed off the walls occasionally. Walls coated with epoxy or lined with PVC do not accumulate as much grease as concrete does, and the grease is more easily washed off. Wet wells that are not designed for self-cleaning present a costly problem. They may require the use of a combination jet and vacuum machine. Again, there must be access for this large equipment.

Safety

All wet wells are considered confined spaces. Consider applicable safety standards such as National Electrical Code and National Fire Protection Association (NFPA) Standard 820. Minimize equipment in wet wells that requires entry by O&M staff for maintenance.

Corrosion

The wet well structure needs to be protected from corrosion that results from H_2S production. Any fittings and piping (especially submersible stations) should be constructed of materials that are corrosion-resistant to moisture, H_2S, and sulfuric acid. Also address the potential for pump plugging from rags, stringy material, and plastic. Although pumps may be advertised as "non-clog," that is often untrue, especially in smaller variable-speed wastewater pumps operating at speeds that encourage recirculation.

Storage

Calculate pump cycle time (and therefore motor starts) for low, average, and peak flow conditions, taking into account storage in the upstream sewer where appropriate. See Example 12-2 for a versatile analysis that gives resting times as well as cycle time. Elevation of upstream low points should be determined and this elevation marked in the wet well so the operator can quickly visually inspect wet well level during periods when an overflow can occur.

Bypass

Bypass capability is becoming a common feature. Consider installing a bypass suction pipe from the wet well, or at least consider how a bypass could be implemented. The fixed pipe extends into the well and the top is fitted with a quick-disconnect fitting to allow rapid connection of a bypass pump. Also consider access to the bypass suction connection and the means for priming the bypass equipment. Self-priming pumps are limited to a suction lift of approximately 80% of barometric pressure. A bypass feature also requires a companion connection to the force main.

Odor

Evaluate the potential for odor generation and the need for odor control. Odor control is essential if the O&M staff must enter the wet well for hosing grease off the walls.

24-4. Mechanical Considerations

One of the most costly elements over the life of the facility is maintenance labor. Hence, developing a concept or method for the disassembly of piping, pumps, check valves, gate valves, traps, flowmeters, and so on for maintenance or removal and replacement is important, even if three-dimensional draw-

ings or spatial models are necessary. Include such plans in the O&M manual. The following guidelines make the maintenance or replacement of pipe, fittings, and pumps easier:

- Either locate the above facilities at a convenient height or install platforms and stairs for convenient access.
- Orient them so that removing valve covers or the device itself does not drench the workers.
- Include a drain and a vacuum cock for emptying the pipe.
- Install strong, safe lifting eyes (sized for maximum potential load) in the ceiling for a chain hoist in small stations, monorails and electric hoists for small to medium stations, and traveling bridge cranes for large stations. Even in small stations, consider a trolley rail for moving the heaviest load to a hatch accessible to the upper-level hoist. Hoists should travel over all heavy equipment and terminate at a doorway. Alternatively, especially for column pumps, install roof hatches for access by a mobile crane parked outside. The hatches and floor space must be adequate for removing the largest equipment. It is bad practice to build the upper part of the station around equipment already in place.
- Provide plenty of clearance [0.8 m (30 in.) minimum, 1.1 m (42 in.) recommended] between the piping and pumps or other equipment for a crew of workers and their wrenches and other tools. Make sure that nuts and bolts are accessible. Increase the clearance for larger pumps. There must be adequate space under the crane or hoist to lay removed parts on the floor.
- Design the piping with enough flexible joints such as sleeve and/or grooved-end couplings to permit the easy removal of pumps, valves, etc. Blocking, tie-downs, or bridles are required for sleeve couplings.
- Install valves to isolate pumps so that any pump can be removed without draining the header and force main.
- Install a spool of the same length as the flowmeter with shut-off valves on the header to permit the removal of the flowmeter. An alternative arrangement is depicted in Figure 20-12.
- Specify rubber vibration isolation sleeves on compressed air piping (from reciprocating air compressors) installed on common pipe support channels with other piping.
- Ensure adequate lighting for safe working conditions. Add electrical outlets for additional lighting for repairs.
- Install GFCI protection on all service receptacles.

Locate the following items in places that are easily accessible for all routine maintenance:

- Traps that must be periodically cleaned
- Check valve covers that must be removed occasionally for cleaning the valve
- Pump vent piping
- Grease fittings (especially intermediate bearings on long shafts)
- Intermediate bearings on long shafts (safely accessible)
- Oil reservoirs
- Dry-break, quick-connect couplings on hydraulic hoses
- Dehydrating canisters
- Mechanical or electrical devices that need periodic adjustment
- If necessary, provide permanent steps and/or platforms for access.

Miscellaneous considerations include the following:

- Keep walking areas clear of overhead obstructions to a height of at least 2.2 m (7 ft). If obstructions cannot be eliminated, cover them with padding and hang warning devices (light chains, for example) on both sides.
- Install convenient hose bibbs. A proper backflow preventer must be installed on the potable water connection to the station.
- Install a lavatory or at least a wash basin.
- Access must comply with OSHA requirements to ladders and manhole steps.
- Provide adequate storage space for spare lubricants, parts, and special tools.
- In larger stations, include at least a wash sink, a toilet, and a 12-L (3-gal) water heater mounted overhead.
- Install a reduced pressure zone backflow preventer downstream from toilet facilities to allow connection of a sink to the potable water system.
- Consider adding a slop sink on lower floor of deep stations.
- Provide an eyewash/emergency shower with drain for operator safety and to comply with safety regulations
- Isolate sanitary piping to the wet well with a check valve to prevent drywell flooding.

Piping

- Pump discharge pipe connections to a manifold should be a lateral entrance into header pipe in

the horizontal plane through a tee or wye fitting. The discharge shut-off valve should be on the lateral adjacent to tee or wye. Never connect a vertical pipe to the bottom of a discharge header.
- Install discharge check valves in horizontal pipes only and below the operating level of the wet well for cavitation and slam prevention and maintenance access.
- A long-radius elbow adjacent to a pump discharge helps to minimize flow-induced vibration.
- When installed, the flowmeter should be located appropriately for ease of maintenance and to obtain required recommended upstream/downstream pipe lengths.
- Use eccentric reducers for pipe sizing changes.
- Provide adequate support for pump suction and discharge piping. Refer to Chapter 4 and ANSI/HI 9.6.6 for guidance.
- On variable-speed systems, analyze the piping to determine natural frequency so as to avoid resonance at every speed. Lock out any speed that produces resonance.
- Design the piping with enough flexible joints such as sleeve couplings and/or grooved-end couplings to permit easy removal of pumps, valves, etc. Blocking, tie-downs, or bridles are required for sleeve couplings.

Access

The Electric Power Institute surveyed nine power plants to identify problems that affect productivity and the safety of maintenance personnel [5]. The suggestions made are also useful for pumping station designers. They include the following:

- Ensure easy access to adjustment points, test points, and filling and draining locations on all equipment.
- Arrange equipment so that access to the malfunctioning unit does not require the disassembly of adjacent units.
- Allow sufficient clearance for effective personnel interaction when a team effort is required.
- Ensure that equipment clearances in hazardous areas allow access by personnel encumbered by protective garments and associated gear.
- Maintain a clearance of at least 1.1 m (42 in.) at the front or back of electrical panels.
- Code or key interchangeable units that are functionally dissimilar to prevent their insertion in the wrong unit.
- Provide places to put and support components being removed or installed.
- Use hinged doors rather than cover plates (which are time-consuming to remove) where frequent access is needed.
- Place controls and associated instrumentation within easy visual and manual reach of normal work positions.

A universal complaint in the survey was the difficulty of accessing equipment that required attention. About 30% of the maintenance time could have been saved had ideal or unrestricted access been possible.

Remember that some types of equipment (e.g., submersible well pumps and submersible wet pit wastewater pumps) are maintenance-intensive and must be removed and reconditioned at regular intervals, and this task may cost about one-third as much as a new pump. Parts for some kinds of equipment (e.g., variable-frequency drives and some foreign-made pumps) are very expensive or, worse yet, may be available only after long delays. Investigate maintenance requirements before using such equipment and conduct lifetime present-worth analyses of the maintenance labor and parts as well as of the costs for energy. Avoid considering only the first cost.

Pumps

- Specify spacer shafts between pumps and motors to allow for the quick and easy removal of backheads and rotors of end suction pumps to make replacement of seals, shaft sleeves, and wearing rings easy without moving the motor or pump.
- Consider the use of a SpiralTrac™ Environmental Controller [6] in the stuffing box to reduce flush and extend the life of both mechanical seals and packing in dirty water (e.g., raw water and wastewater) service.
- For dry pit pumps subject to possible flooding, specify a grit expeller to prevent scoring shaft sleeves and bearings. Consider immersible motors to drive the pumps.
- Learn how to extend the life of bearings and reduce the cost of maintenance (see Section 24-5).

24-5. Smooth-Running and Reliable Pumps

To have a smooth-running pump, several factors must be considered. Some of them are within the control of a pump manufacturer, whereas the design engineer controls the following:

1. Pump operation within the preferred operating range of the pump. ANSI/HI 9.6.3-1997.

2. No cavitation or air entrainment. ANSI/HI 1.3-2000.
3. Piping with straight and uniform flow into the pump suction. ANSI/HI 1.3-2000
4. Proper balance quality grade specified and provided. ANSI/HI 9.6.4-2000, ISO1940/1 [7].
5. Piping connected and anchored without pipe strain. API 686.
6. Couplings and components aligned with the pump. API 686 [7, 8, 9].
7. Pump foundation of adequate mass and rigidity. ANSI/HI 1.4-2000, API 686 [7, 9, 10].
8. Proper anchor bolts and baseplate grouting. ANSI/HI 1.4-2000, API 686 [7].
9. Effective and thorough field inspection and acceptance testing or verification of all the above [7, 8].

Many organizations, both public and private, have significant numbers of centrifugal pumps in service and have determined that the above items are critical to good, long-term pump performance, and some information on the subject has been published [7, 8, 9, 10]. For example, the National Aeronautics and Space Administration (NASA) maintains a large mechanical, structural, electrical, and civil engineering infrastructure in support of its space launching systems. Shell Oil Company has a large inventory of pumps in its refineries. Lockheed Martin Michoud Space Systems has an extensive array of various type of rotating equipment in its facilities. Lockheed Martin, Shell, and NASA have developed stringent specifications for pump vibration, shaft alignment, pump installation, field testing, and verification. Many large public water and sanitation agencies have done the same thing in recent years. NASA developed their standards as a result of the following observations on their projects and installations in 1997 [7, 8]:

- At the Johnson Space Center, 92% of rotating equipment was improperly installed. Issues were primarily related to balance and alignment. Similar problems were encountered at the Goddard Space Flight Center and the Kennedy Space Center.
- The underlying cause was attributed to specifications that did not address current best reliability practices and commissioning practices that had not been updated to reflect changes in system and component capabilities available.
- A high percentage of new rotating equipment in two other referenced government agencies had defects.

The following subsections summarize the effects of some of these factors.

Baseplates and Foundations

The very basis for controlling or limiting vibration and shaft alignment lies in the pump baseplate and associated concrete foundation. A stiff baseplate and properly and fully grouted base are essential. Flimsy baseplates and voids in the baseplate grouting diminish the dampening effect and permit resonating. Specifications and designs should typically require that the baseplates be fabricated steel per API 610 or ANSI/HI 1.3. Foundation preparation, baseplate installation and grouting, driver mounting, base leveling, and pump shaft alignment should conform to ANSI/HI 1.4 or API 686.

Shaft Alignment

The experience of many organizations reveals the importance of alignment specifications. A good straightedge alignment produces errors of about 0.5 mm (20 mils) angular and perpendicular instead of the 0.05 mm (2 mils) called for in the Hydraulic Institute standards and as recommended by nearly all mechanical seal manufacturers. The NASA data show a 10:1 bearing life reduction as the difference between the two alignment standards. Two ways to achieve the 0.05 mm (2 mils) alignment are: (1) the use of dial indicators by qualified millwrights, and (2) the use of laser equipment. Laser equipment is now inexpensive and so easy to use that ordinary mechanics can easily become qualified to align equipment.

Rotating Equipment Balance Quality Numbers and Bearing Longevity

A key factor pertaining to pumps is balance and vibration. NASA's research shows the effects of bearing life in years compared to the ISO 1940/1 balance grades. The data show bearing life for the various balance quality numbers as follows [7]:

- Grade G1.0 — 12 years
- Grade G2.5 — 8 years
- Grade G6.3 — 7 years
- Average (no specified balance quality number) — 4 years
- Grade G16 — 3 years

Typically, specifications for centrifugal pumps reference standards such as Figure 9.6.4.15B in ANSI/HI 9.6.4, API 610, and ISO 1940/1 indicate acceptable unbalance. ANSI/HI 9.6.4 conforms to an ISO balance quality grade of G6.3, whereas both API 610 and NASA [7] require a balance grade of G2.5. Large pumps (100 hp and greater) in the water and wastewater utility industry are also sometimes specified to conform to this G2.5 limit.

Effect of Vibration on Bearing Life

Bearing life increases many times over with modest reductions of vibration. For example, the life of ball bearings is increased by 700% with a 50% reduction in vibration. See Table 12-4 for other data.

Torsional Vibration

Torsional vibration is difficult to detect until a shaft breaks. It does, however, influence the life of bearings. For large pumps (> 75 kW or 100 hp), consider requiring the pump manufacturer to have the candidate pump analyzed by an independent, experienced consultant using the programs developed by Corbo and Melanoski [11, 12] and to modify the pump as needed. See also Section 1.05.B.3 in Appendix C.

Small Pumps: A Possible Exception

There is at least one instance [9] of an organization that has tentatively determined that the traditional, field-installed epoxy grouting of pump baseplates for small pumps [apparently 55 kW (75 hp)] and smaller) may not be economically justifiable. That is, the cost of the field-installed epoxy grouting may exceed the savings in reduced O&M costs. Instead, alternatives such as cementitious grouting, pre-grouted baseplates (performed at the factory), and polymer concrete bases should be considered for such pumps.

The Results: Reduction in Annual O&M Costs

The reduction in annual O&M costs is dramatic, as shown in Section 12-11. The dire consequences of factors that prevent smooth operation of pumps are outlined in Tables 12-3 and 12-4. Obviously, maintenance is greatly reduced and equipment service life increased significantly when pumps run smoothly, and the result is significantly reduced life-cycle costs (LCC)—the cost of ownership.

Achievement

Reliability and smoothness can be achieved by observing the following recommendations. They begin with design and specifications and extend through final commissioning, and they make it possible to extend intervals between repairs beyond 40,000 operating hours—five years of continuous operation.

1. Develop the calculations for the performance requirements completely and on the basis of the best- and worst-case assumptions for system dynamic head losses (see Section 10-8 and Example 12-3).
2. Specify *all* potential operating conditions or an envelope of operating points that encloses all operating conditions. Make certain the installation, including pumped fluid and operating environment, is described fully and that pumps always operate within the AOR and for usual operation, within the AOR.
3. Specify materials that are suitable for the application. Do not be swayed by manufacturers' recommendations for standard materials but, rather, address each aspect of the application from the position that the small additional cost for corrosion-, erosion-, and cavitation-resistant materials is well worthwhile for extending the service life of the equipment and avoiding much of the cost of repairs.
4. Specify the proper balance grade number. The Hydraulic Institute allows a balance grade number of G6.3 per ISO Standard 1940/1. Both the API and other specifications [7] require a more stringent balance quality number of G2.5. Large pumps [75 kW (100 hp) and greater] in the water and wastewater utility industry are also sometimes specified to conform to the G2.5 limit.
5. Design the intake system to Hydraulic Institute requirements (see Section 12-1 et seq.). Ensure there is no air entrainment and no cavitation. Design piping for straight, uniform flow into the pump suction.
6. Design the foundation to be substantial and rigid enough to absorb vibration—at least five times the weight of the pump, per ANSI/HI 1.4-2000.
7. Design the installation for easy access to all components, especially those portions of the equipment that require monitoring and routine preventive maintenance.

8. Make certain the installation details meet or exceed standards such as the Hydraulic Institute limitations (HI 1.3 and 1.4) or API Recommended Practice 686.
9. Require the manufacturer to guarantee the equipment for the required peak capacity and head, but make sure that the most frequent operating condition is located in the pump's POR.
10. Specify NPSHA at all operating conditions and require that the manufacturer selects pumps with generous NPSH margins (see Section 10-4).
11. Require the contractor to purchase and to test the entire pumping package, including drive equipment and controls from a single manufacturer, who must certify that the corporation is assuming unit responsibility obligations for all equipment specified with the pumps.
12. Require a detailed submittal process documenting that all components meet the specified requirements.
13. As part of the submittal process, require the manufacturer to demonstrate, using finite element stress analyses and up-to-date dynamic analysis methods, that the equipment is specifically designed for the specified operating conditions and it will be free from lateral and torsional resonance in all components, including support frames in both the as-new and as-worn conditions (see Section 10-6 and Appendix C, Sections 1.04 and 1.05).
14. Require the contractor to employ competent millwrights to install baseplates leveled to 0.2 mm/m (2 mils/ft) by means of precision instruments and to pour epoxy grout under at least 150 mm (6 in.) of head. (ANSI/HI 1.3 and API Recommended Practice 686 offer advice on this subject.) For very large pumps, consider requiring the millwright crew to practice on a throw-away baseplate. Baseplates for small (< 60 kW or 75 hp) can be pregrouted [9].
15. Require the contractor to use millwrights to check for soft foot [10] (allowable error is 0.05 mm or 2 mils) and to mount motors and align pumps and piping by using either the reverse dial indicator method or laser alignment procedure. Never allow the straight-edge alignment method or the use of laborers or untrained personnel. Ensure connecting piping is without strain.
16. Require the millwrights to align shafts within 0.05 mm (2 mils). Increasing misalignment to 0.13 mm (5 mils) reduces bearing life by 50% or more.
17. Require effective and thorough field inspection and acceptance testing or verification of all the above.

The Results: Reduction in Annual O&M Costs

Done properly and with studied attention to detail, the installation will be successful and will provide the owner with a lowest overall cost installation that will be reliable and maximize the total service life.

24-6. Electrical Considerations

According to the NEC, electrical control panels must have a minimum clearance in front of at least

- 0.91 m (3 ft) for 0 to 150 V
- 1.07 m (3 ft, 6 in.) for 151 to 600 V
- 1.22 m (4 ft) for 601 to 2500 V
- 1.52 m (5 ft) for 2501 to 9000 V.

The same clearance is required at the back if the panel is a free-standing unit. Free-standing panels are recommended over wall-mounted panels.

A lock-out stop should be installed at each motor (or at any rotating equipment) that cannot be seen from the control panel to prevent accidental energizing while the equipment is being inspected or serviced. A time-running meter for each pump can furnish useful information for scheduled maintenance. Include heaters or desiccant containers (to keep electrical equipment dry) and put control panels in a location free from dust, fumes, and H_2S.

For large pumping stations, consider computerization for monitoring and controlling the system.

Make certain that the electrical engineer provides a liberal number of convenience outlets for tools and maintenance equipment, such as lights and fans (120-V, single-phase), and for welders (480-V, three-phase). All outlets should be GFI protected.

Use hermetically sealed switches and relays in wastewater pumping stations. Use NEMA 7 equipment for hose-down spaces.

Post a single-line diagram of the power system—especially if there is auxiliary power. Electrical and instrumentation and control (I&C) systems require specialized considerations for the designer. Design, installation and safety are largely determined by the NEC, NFPA, and OSHA codes and must be followed. The design must incorporate only the most current codes (they change periodically). Pumping systems are becoming more sophisticated because of microprocessor-based devices such as PLCs, VFDs, and soft-start controls. Many of the protective devices such as circuit breakers and motor starters also incorporate more sophisticated technology.

Consider how electrical and I&C systems can affect pumping station reliability and ease of O&M. Special equipment is required for O&M of these systems. Similarly, a high level of knowledge, skills, and abilities is required for an effective O&M program.

Equipment and system-wide standards should be developed to minimize the need for equipment, O&M knowledge, and spare parts needed.

Specify procurement under a single contractor specializing in electrical and I&C systems to ensure compatibility among components and systems. Such a specification also produces more consistency in documentation such as single-line and detail schematic diagrams. Standards (NEMA) should be specified for wire size and type, connections, marking, and splicing (splicing should not be allowed).

Variable-Speed (V/S) Systems

A V/S drive system must be carefully evaluated during the design to determine whether the higher acquisition and O&M costs are justified in terms of intended function and hydraulic system performance. A great many V/S systems have been installed under the perception that they will reduce energy cost, reduce wet well size, and provide more balanced hydraulic loading to the wastewater treatment plant. In many of the installed systems, one or more of these expectations have not been met, in large part due to insufficient system hydraulic analysis and the application of V/S systems that were not understood by the designer. Consequently, these stations suffer from increased life-cycle costs.

Other considerations include the type of drive used. VFDs, for example, (depending on the type) may require large isolation transformers that reduce overall wire-to-water efficiency, increase heating (and may require air conditioning), and always require more space. Motors may also require special construction of the windings and insulation, depending on the inverter output. Noise and possible harmonic interference with other I&C equipment may also be problems. Early obsolescence may reduce useful life due to unavailability of spare parts. Procurement must include the necessary equipment needed to operate and maintain VFDs, such as programming devices, diagnostic meters, and analyzers necessary for troubleshooting. Either the O&M staff must be trained or a reliable outside source for maintenance must be readily available.

SCADA

Install SCADA systems in all but the very smallest of stations. SCADA provides invaluable real-time pump station performance information. The information allows the operator to make informed decisions on the deployment of resources (pump, generators, people) during widespread power outages, fire, high peak flow conditions, or other emergency-related events. Information such as pump start/stops and motor running times can be interfaced with other information management systems (such as computerized maintenance management systems) to generate preventive maintenance work orders or investigate anomalies that vary from historical trends. Most systems (electrical, pumps, wet and dry well level, and others) can easily be interfaced with SCADA. If an existing SCADA system is not available, carefully evaluate the method of communication and telemetry to ensure reliable data communications.

Motors

Motors are dramatically affected by operational conditions caused by poor hydraulic analysis. Motors should be selected as non-overloading throughout the full speed head/capacity curve. They should be specified with a 1.15 service factor to accommodate limited periods of overload, up to the service factor rating. The service factor should not be used for routine overload conditions in the normal operating range. The minimum insulation rating specified should be Class F. Insulation failure is the most common mode of failure in ac induction motors. Specify motors as Code G or better, and coordinate them with the engine generator selection. Specify standards for vibration (for example, NEMA, MG-1-2003 or IEEE 841-2001).

Specify high-efficiency motors because the energy savings over the life of the motor will exceed the difference in cost by several times.

Energy Considerations

The power factor correction should be evaluated if the power utility levies a power factor charge. Also evaluate the operational configuration for peak electrical energy demand charges.

Where permanent engine generators are installed, the utility company may offer peak shaving contracts that should be considered. Fuel tanks for diesel engines must be sampled annually to measure

contaminants, so install a petcock for sampling. Also install a fuel gauge.

By all means, add wet well high- and low-level alarm floats, independent of the PLC or other supervisory controls so that excessively high and low levels can be sensed if the level control system fails.

Consider installing spare conduit for power and control during construction. Install drip legs in underground conduit to prevent the accumulation of water.

Avoid installing electrical equipment in the wet well, if possible. If absolutely necessary, it must be either explosion-proof or intrinsically safe in conformance with NEC Section 500. Explosion-proof fixtures and other devices are not moisture-proof. Replacing a light bulb in a fixture may require replacement of the entire fixture because of corrosion and loss of explosion-proof integrity. Hence, place fixtures in easily accessible locations (using flexible conduit, for example), and provide for fixture replacement. Fixtures, conduit, and junction boxes should be coated with PVC (or other corrosion-resistant material) to minimize attack from moisture, H_2S, and sulfuric acid.

Where conduits are connected to the wet well from the dry well (or an electrical control enclosure) they must be sealed off to prevent migration of toxic, corrosive, or explosive gases from the wet well to the electrical equipment and/or the dry well.

24-7. Architectural Considerations

Pumping stations are frequently sited in undeveloped areas and become objectionable when housing development encroaches. The design should be predicated on future growth and development. The entire layout, including housing, drainage, storage for equipment, and safety, is very important to the maintenance operation. Consider the following items.

Housing

- Access for the installation and removal of equipment (for example, use double doors or roll-up doors to remove large pieces of equipment such as pumps, motors, or engines).
- Heating and air conditioning, which may be necessary depending on weather conditions.
- Convenience for the cleaning and painting of the facility.

Drainage

- The drainage system and sump for wash-down water.
- Drainage from the sump of water pumping stations piped to a storm drain or sewer, or (if that arrangement is impossible and a French drain is used) consider the groundwater table for all seasons of the year. If a submersible sump pump is used in a wastewater pumping station and the discharge is piped to an adjacent upstream manhole, it may be possible to dewater a small flooded station.
- A drip line for water-lubricated packing and a drain system to the sump.
- Sloping the floor at about 1% to floor drains to eliminate puddles.

Storage

- Cabinets to store records, spare parts, and tools.
- Record drawings (plans) and O&M manuals should be stored at each station.

Safety and Security

- Wastewater pumping stations must be equipped with an adequate forced-air ventilating system with air powered in and air powered out.
- Adequate lighting to supply at least 200 to 300 luxes (20 to 30 ft · cd) at the floor or, alternatively, about 100 luxes (10 ft · cd) with auxiliary portable lights and GFI receptacles.
- Security and the risks of vandalism.
- Intrusion protection with a remote alarm.

Miscellaneous

- There should be a means of checking the water level in water wells because this helps in determining the proper pump head, drawdown, or changes in the water level in the aquifer.
- It may be desirable to have a color-code system for the piping, particularly if a water station pumps into more than one pressure zone.

24-8. Standby Facilities

Consider using standby pumping facilities for important pumping stations. Most states require a second source of power for wastewater pumping. Possible sources include:

- Another utility source. (Feeding from two separate substations is almost always adequate, but it does not guard against area-wide blackouts.)
- A nearby major industrial plant with a large electrical generating capability.
- A fixed, dedicated engine-generator set.

- A dual engine/motor drive with an appropriate, automatic clutch that disengages the engine when the motor is energized and engages the pump shaft, thus allowing the motor to spin freely when the engine is running. This configuration is least desirable from the standpoint of increased costs for O&M and reliability.
- A portable engine–generator set that is readily accessible and well-maintained.

Standby diesel or natural gas engines should be exercised for at least 20 to 30 min after reaching operating temperature once each month at nearly full load (see also Section 14-22). Low-load operation results in carbon deposits in the exhaust of diesel engines, and inadequately exercised engines require periodic dismantling for cleaning. An engine sized for pumping peak flows or operating under the worst conditions means that the engine may operate at one-third or less of its capacity when exercised. One solution is to design the electrical system so that a portable load bank can be used to exercise engines under full-load conditions. Preferably, the load bank will provide an inductive load, similar to that of the motor load and not a resistive load. Another solution is to feed the electricity generated back into the utility's power lines, but note that synchronization with utility power is required.

Reliability can be improved by electrically heating the engine, by trickle-charging the batteries, and by vigilant maintenance—all of which should be included in the O&M manual. Engines should start and be ready for loading in 8 seconds.

24-9. Specifications

Include specifications that require the supplier of pumps and other equipment to furnish pump curves, maintenance data, parts lists, and the manufacturer's recommendations for service and maintenance. The names and locations of parts suppliers should be a part of the records.

Many old, poorly designed pumps are still being sold (some by well-known manufacturers). Specifications obtained from the manufacturers (or any organization representing them) coupled with low bids may lead to the use of these shoddy pumps. For normal service, specify a limit for shaft deflection of no more than 0.15 or 0.18 mm (0.006 or 0.007 in.) at the wearing ring and 0.05 mm (0.002 in.) at a mechanical seal. The allowable deflection at stuffing boxes is greater but, because operators may want to replace a stuffing box with a mechanical seal, the more restrictive deflection is preferable. Interviews with manufacturers at pump exhibits and with pump maintenance crews or supervisors of several utilities provide valuable instruction in maintenance economy, and a present-worth analysis of the expected future costs of maintenance, labor, parts, and power is revealing.

24-10. Operators' Preferences

The opinions of operators are invaluable. This section contains a potpourri of selected opinions on a variety of subjects. Although these comments come from operators of wastewater pumping systems, most of them apply to fresh water systems as well. Comments by non-operators are given in italics, and the commentators are named.

Los Angeles County Sanitation Districts

The following comments are paraphrased from conversations with Kettle [13].

Wet Wells

The Lancaster Reclamation Plant wet well in Figure 11-12 is very poor because it is so large and because the floor is so flat. Large quantities of grit and scum accumulate. Rebuilding it with floor slopes of 45 degrees and pumps or pump inlets set into a small pocket so that scum could be readily sucked out was considered, but the cost was prohibitive.

Valves

Trouble was experienced with high-quality eccentric plug valves in digester lines. They were replaced with solid wedge gate valves exercised weekly, and there have been no problems with them. In fairness, however, the plug valves had not been exercised. *Sanks: Others have used eccentric plug valves with complete success. Lescovich [14] warns that excessive tightening of the flange bolts on eccentric plug valves squeezes the rubber gasket out of the flanges and binds the plug. Bolts should be only tight enough to prevent leakage.*

Bubblers

Air tubing was plagued by plugging with grease and with leaks that could not be fixed because the tubing

was buried in concrete. These ills, plus the maintenance of compressors and associated mechanical equipment, made the air bubbler system irritating. Pressure cells were substituted at all pumping stations by attaching a 150-mm (6-in.) flange to the dry pit wall below LWL, installing a 150-mm (6-in.) gate valve, and drilling through the wall into the wet well. A short spool is attached to contain the pressure cell and a fresh water line is fitted to flush the cavity continuously with 0.06 L/s (1 gal/min) of fresh water. An electronic control box completes the installation. *Sanks: Leaks can be fixed when air tubing is installed in conduit to make all joints accessible. Suitably derated air compressors are trouble-free. Arbour: Solid-state transducers are now in common use for wet well level control and provide a low-maintenance, high-reliability solution. A common practice is to continue to use low- and high-water level floats (independent of the supervisory control system) to ensure a fail-safe back-up.*

Packing Glands versus Mechanical Seals

According to Redner [15], new pumps are always required to have mechanical seals, and the seals that come with the pump are immediately replaced with the owner's standard, tungsten carbide-faced seals. To fit mechanical seals to their older pumps that were equipped with packing glands required machining the pumps for installing larger shafts and bearings. The maximum allowable deflection at the end of the shaft is (and was) 0.025 mm (3 mils). More deflection would quickly destroy a mechanical seal.

Packing glands leak, of course, so housekeeping is a problem. They must be adjusted weekly (a continual source of maintenance), and shaft sleeves eventually become scored and must be replaced at considerable expense. Mechanical seals are preferred because they require no maintenance, last for many years (10 or more), and eventually pay for themselves, although their first cost is many times greater than the modest cost for packing glands. *Sanks: The costly mechanical seal can be reconditioned to be like new for about one-third of the cost for a new one. A mechanical seal can be reconditioned several times.*

County Sanitation Districts of Orange County, California

Fresh water for seal water systems is avoided in most pumping stations. By using graphite-Teflon® packing, the wastewater itself, without filtration, becomes the seal water. This scheme has worked well for a decade, and, according to Arhontes [16], the wastewater does not cause excessive wear. This system is economical and more reliable because there is no dependence on a city water supply that might be interrupted by an earthquake. When pumps are dismantled to replace worn impellers, new shaft sleeves are also installed.

In contrast to the antipathy to air bubbler systems in the Los Angeles County Sanitation Districts (see above), air bubbler systems are preferred, and the Orange County engineers have designed a standard package for the entire system. The air tubing is installed in stainless-steel conduit to allow the bubbler system to be easily removed for fixing leaks.

Seattle Metro

After two decades of operation, the operational friendliness of 29 Seattle Metro (now King County Wastewater Treatment Division) pumping stations with V/S drives and two older, "traditional" designs with C/S pumps was studied carefully by 10 senior operators with an aggregate total of 200 years of operating experience. Most of the 29 pumping stations have trench-type sumps of the type illustrated in Figures 17-14 through 17-20. In addition, there are two "traditional" designs with large, flat floors. The following conclusions were reached by the operators in a day-long seminar held in 1989 and attended by Garr Jones (one of the original principal designers of these stations) and Gary Isaac (superintendent of operations).

Access

1. We recommend catwalks of fiberglass or aluminum gratings with rails (for safety) to all parts of the wet well and for all out-of-reach locations where routine maintenance is required. *Bosserman: Make catwalks or platform gratings strong enough to support those pieces of equipment that may be removed or installed.*

Alarms

2. Warning systems (whether internal or transmitted elsewhere for monitoring) should be limited to those critical for station operation. Too many alarms can cause operator complacency and failure to respond when response is really needed. We have eliminated station-occupied and ventilation-system-failure alarms in an effort to reduce troublesome calls.

3. Having early warning of high water followed by flood warning usually gives operators enough time to respond.
4. It should be impossible to clear alarms at the treatment plant when the acknowledge button is pressed. Alarms should be cleared only at the remote station, but when two stations are interconnected, it should be possible to clear alarms at either one.
5. There should be alarm lights for all significant water levels in the wet well.
6. Explosive gas detectors are desirable if they are maintained properly. *Jones: NFPA 820 mandates the use of explosive-gas detectors.*

Auxiliary Power

7. All stations should have back-up power, either engine–generators or dual power feeds.
8. Dual power feeds are preferred over engine–generator sets and seem to be more reliable. High winds can sometimes disrupt both feeds, so at least one should be underground. *Sanks: The reliability of dual feeds is suspect because they offer no protection in the event of an area-wide power failure. Arbour: The reliability of dual feeds should be examined very closely as it is related to the power utility and is often a regional issue.*
9. We prefer natural gas for engine–generator sets. *Jones: The use of gas is site-specific. In the South, where it is common practice to shut down gas-producing and distributing installations for days at a time when a hurricane threatens, diesel is the fuel of choice. In the Pacific Northwest, gas may be the best choice because buried utilities are secure against windstorms and most floods. However, leaks are very hazardous. As the engine is presumably in an unattended station, a hydrocarbon gas detector is a better choice for shutting off fuel than a fuel pressure detector. The latter gives rise to spurious shut-downs. Cronin: Gas may be unavailable if power failure is due to earthquake. Also, gas engines are larger than diesel engines.*
10. Portable engine–generator sets are of no value in snow or for remote stations. *Arbour: The application of portable engine–generator sets is limited to the capacity needed. Gen-sets up to about 150 kW can be successfully used in hilly terrain or northern climates with snow if the towing vehicle is suitable. Over 150 kW, the weight and size of the set, cables, and fuel probably require specialized towing equipment and may become difficult to tow in hilly areas or over snow and ice.*
11. Stationary engine–generator sets may not start automatically. *Jones: For reliability, exercise them with regularity and long enough at or near full-load operating temperatures to cook off moisture in the crankcase and elsewhere. See also the advice in Section 14-22. Arbour: If the station is monitored by a SCADA system, automatic start, stop, and power transfer can be accomplished with a high degree of system safety and confidence.*
12. We think engine–generators should shut down automatically when commercial power is restored. *Isaac: I disagree. Operators should be dispatched to check all systems. Jones: There is great potential for damage if control circuits are not reset and full power is not restored to divided busses as a consequence of any power failure, no matter how brief.*
13. Adequate noise control is needed for engines. *Sanks: See Chapter 22 for measures to reduce noise.*
14. Storage tanks for diesel fuel should be large enough for at least one day of operation. *Jones: Typically, power failures rarely last longer than one hour, although storms, earthquakes, and other disasters have interrupted power in every section of the country for several days, thus prompting engineers to design for excessive storage. Diesel fuel deteriorates, so the exercise program should be able to recycle the fuel inventory every 6 to 12 months. Arbour: An alternate is to have dedicated fuel delivery capability from a supplier or an internal tank truck if available.*
15. There should be load banks or some other means to exercise engines under no less than about 90% of full load. *Arbour: In larger metropolitan areas, load banks can be rented for this purpose from engine–generator suppliers. The bank should provide an inductive load.*
16. The transfer switch and the station's main breaker should be sized to take the station's entire load.
17. PLCs (programmable logic controllers) should be programmed to start pumps one at a time.

Auxiliary Water Systems

18. Solenoid valves to main pumps should all be on essential power.
19. In addition to a backflow preventer, we [in Seattle] must also have a water storage tank with an air gap, and we use a regenerative turbine pump to pressurize the system. *Sanks: Multistage centrifugal pumps are also excellent for either seal water or wash water (see Figure 11-34).*

20. Make sure that water pressure is available in the station while the tank is filling.

Controls

21. Keep controls as simple as possible.
22. We prefer the control room to be isolated from the pumps. *Sanks: All controls should be above the flood plain. Jones: Modern electronic controls should be located in air-conditioned rooms with air treated to remove hydrogen sulfide and particulates.*
23. PLCs work well but need a manual back-up for those times when the system goes down. *Cronin: All stations should have duplex PLCs. Sanks: Some operators would like a pump-down selector switch with a timer to return the switch to automatic operation after a preset time. Jones: Timers are not recommended. Cleaning operations should be supervised and the station returned to normal operation with operating personnel present. Arbour: If a bubbler system is used, a pneumatic wet well level can be used as a back-up to digital level indicators. It allows the operator to manually control via the HOA (hand-off-automatic) switch for each pump. Sanks: Small pumping stations with submersible (self-priming) pumps of less than 50 hp can be automatically cleaned safely with no operators present.*
24. Duty switches need to be set so operators can select any combination of lead/follow pumps. *Sanks: Light switches and other switches in dry wells should be waterproof so that the entire dry well can be hosed down.*
25. It should be possible to control the pump either at the control panel or at the pump unit. *Jones: This seems to me to be a bad idea because it causes an unnecessary complication in controls. Worse, it can lead to damage by making it possible for an operator to lock out equipment or leave it in the manual mode when leaving the station. Furthermore, with hand-held, low-power radios so readily available, one operator can be at the machine and another can be manipulating the controls. Cronin: If the client insists on a control at the pump when in manual mode, install an operation light to remind operator to reset the controls to AUTO. Sanks: Do so regardless of the location of the manual control. Arbour: The NEC requires, under some circumstances, having a local motor control lock-out that can serve as a local control. Having local control enables staff working on the pump level to operate the system after maintenance is performed. Often, because of the interior noise levels in a pump station with equipment operating, communication between the pump floor and control level is difficult and sometimes impossible. Written procedures must be in place and heeded to ensure that the pump station is operational (all switches in position) when departing the station.*
26. It is nice to have an indicator of the water level in the pump room for stations with deep dry wells. *Jones: The operator should also have the wet well level instruments available when operating the pump in manual mode.*
27. Bubbler systems are still our standard, but sonar has improved in reliability recently. Back-up floats for high-water alarms are also needed. Otherwise, we do not recommend float switches except for constant speed pumps. *Sanks: See "Air Bubblers," Section 20-3, for proper design. Jones: Bubblers are immune to power failure whereas electronic instruments are not. Float switches are mechanical devices and subject to wear. Bubblers and pressure cells (with virtually no moving parts) are inherently more reliable.*

Cranes, Hoists, Lifting Eyes, and Access Hatches

28. All equipment weighing more than 45 kg (100 lb) should be accessible by crane.
29. Bridge cranes, properly located to allow direct vertical access to major station equipment, are absolutely necessary.
30. The crane hoist should be electrically operated. Removing a heavy object from far below grade with a manually operated hoist is a waste of a journeyman-level worker's valuable time.
31. Include the "inching" mode of operation—a feature (available from most manufacturers) that allows slow and precise positioning of large items such as motors.
32. Extend crane rail girders outside the building if possible. *Jones: A better design allows trucks to move into the building and under the crane for loading heavy equipment directly onto the truck bed.*
33. Lifting eyes should be an integral part of station design and need to be strategically located over critical equipment. *Arbour: Lifting eyes, bridge cranes, and hoists must all be load-rated and periodically load-tested.*
34. Hatches or floor covers with plenty of clearance for pump removal should be located over each pump. They should be light in weight (preferably aluminum) and either spring-loaded or easily removable.

35. Openings around hatches need proper barricades built into the hatch design to ensure a safe working area.
36. When lifting eyes are needed, locate them properly and design them with an adequate reserve of strength. (See Figure 25-6.)

Custom-Engineered Pumps

37. We are firmly committed to high-quality equipment and custom-engineered pumps with larger shafts and bearings.
38. The best metallurgy for impellers and pump casings (particularly for raw wastewater) is cost-effective in prolonging useful life. For example, our pumps have generally lasted 20 years or more. Pumps in four stations finally wore out and our engineers, unfamiliar with the thought that went into the original designs, replaced them with standard pumps. Troubles developed at the outset, and the standard pumps lasted only four years or less.
39. We now replace custom-engineered pumps with units of the same quality. Where grit is a problem, pump casings should be hard cast iron.

Design Implications

40. Use common sense in laying out piping and valves so stations can be operated safely and with easy access to all equipment.
41. Good pump system hydraulics are important.
42. Design for the lowest head in pumping. Do not allow a free fall of wastewater into a pool below.
43. Involve the operators in the design of the station.
44. Work with the neighbors so that they can claim ownership in the final station with regard to colors, landscaping, and noise and odor control.
45. Success requires good design, a good contractor, and good inspection.

Doorways

46. Doors should allow heavy equipment to be loaded easily onto maintenance vehicles for transport to the shop.
47. Measure the utility's trucks to be certain doors are wide enough. At two of our stations, the entrance door had to be modified to allow adequate clearance for maintenance vehicles.
48. Roll-up doors are preferred over double doors.

Drains

49. All areas should have adequate drains and all floors should slope about 1% to the drain to avoid standing water.
50. In the immediate vicinity of the drain, the slope should be increased to 2%.
51. Drains should be large and must not have check valves. *Arbour: Only if the dry well cannot be flooded during high wet well conditions*
52. Pipe drains are preferred to an open, rectangular drain along one wall.

Floor Finishes and Coatings

53. Use non-skid coatings where water is prevalent.
54. Use finishes that are easy to keep clean. Urethane finishes are satisfactory and control concrete dust.
55. Do not paint floors; paint peels in a matter of months. *Arbour: Peeling is highly dependent on surface preparation and conditions such as moisture. Surface coatings can be successfully applied to below-ground dry well floors and walls. It improves lighting and makes cleaning easier. Pumps, piping, and valves also should be coated for corrosion protection.*
56. The newer sealers have not been holding up well.

Force Main Draining and Cleaning

57. Keep force mains small to maintain adequate velocity. They must be large enough to accommodate only the sewage volumes to be removed.
58. Sump pumps and drains should be plenty large enough to take care of accidental spills. (See Section 12-3.)
59. Low turbulence discharge structures reduce hydrogen sulfide damage.
60. Pig launching facilities should be provided for all force mains. Pigs should not be larger than 600 mm (24 in.). Use dual force mains if necessary to avoid larger pipe. Use dual headers for dual force mains.
61. Design the force mains for ease in accessibility in the station, along the alignment, and at the discharge structure.

Gear Boxes

62. We dislike gear boxes because of the heat, noise, and oil leakage. *Sanks: They also tend to induce vibrations.*

Gratings

63. Removable gratings over the wet well (e.g., Kirkland Pumping Station, Figure 17-14) are a nuisance. To facilitate hosing the side walls, design a catwalk along the full length of the wet well.
64. If gratings cannot be avoided, use non-skid fiberglass that can be easily lifted. Even aluminum is too heavy.

Horizontal versus Vertical Pumping Units

65. Pumps and motors in a horizontal configuration are excellent because alignment is better than in vertical units, but they require a larger footprint area and are more susceptible to flooding.
66. Avoid the use of intermediate shafts if possible. Vertical units require shafts, and vibration problems tend to develop.
67. *Isaac: We have some units with flywheels. They have not been a problem, but nevertheless they should be avoided unless necessary to solve a hydraulic problem. Jones: To the best of my knowledge, they have worked well.*

Instrument Air Systems

68. Air tanks (receivers) should have hand bleed-offs. Automatic bleeds are not necessary.
69. Dual compressors should be valved so that one can be removed while keeping the second in service.
70. Both compressors should not be on the same switch gear.
71. Purge valves should be spring-loaded to return to their normally closed position.
72. Air dryers are recommended for all instrument air.
73. Design for high flow rates at low pressure: 240 to 350 kPa (35 to 50 lb/in.^2) is adequate.
74. Lead/follow compressor selection is better than automatic alternating compressor operation. *Jones: We have not advocated automatically alternating lead sequencers. We think it is just one more thing to go wrong. Instead, we furnish the operating staff with hour meters on the MCC cubicle doors and a lead/follow selector switch. This approach leaves it up to the operator to decide when to select the other unit to function as the lead unit in a lead/follow automatic starting sequence. Arbour: Manual lead/lag selection also allows the operator to observe successful operation of the new lead pump. It also allows the operator to control the timing of alternation.*
75. Some new level control systems eliminate the need for instrument air; sonar and pressure cells are two examples.

Landscaping, Irrigation, and Shrubs

76. Many of our stations are located in areas of heavy public use and high visibility, so good landscaping is necessary. Landscaping should blend with surrounding area and be kept simple with ease of maintenance in mind. Low, slow-growing, disease-resistant plants should be specified. Avoid ivy and thorny bushes.
77. Lawns should be kept to a minimum. Consider grasses that need a minimum of weeding and mowing.
78. Automatic sprinkler systems with timers should be installed. *Sanks: Moisture probes are also useful for irrigating when necessary.*

Lighting

79. Good lighting is very important. Interior lights should come on instantly.
80. Where gases can exist, lighting should be explosion-proof.
81. Pumping stations should be well-lit outside for protection from vandalism.
82. Include emergency battery-powered back-up lights for each pumping station floor.
83. Some lights take too long to light (e.g., high-pressure sodium) and take a very long time to turn on right after they have been turned off, so choose other types. *Sanks: Sodium vapor lights are alright for outside lights that turn on automatically.*
84. Locate fixtures so that access by ladder for changing bulbs is easy.

Noise Control

85. Noise control is an important consideration for employee health and public relations.
86. Engine–generator sets may require noise control.
87. Variable-frequency drives are also noisy and may require special treatment. *Sanks: AF converters are sensitive to dust, hydrogen sulfide, and temperatures higher than 32°C (90°F), so consider an air-conditioned enclosure.*
88. Consider the use of noise absorption walls and ceiling.
89. Odor control systems will probably require noise abatement.
90. Rubber-mounted equipment helps to absorb vibration and reduce noise.

Odor Control Facilities

91. Facilities should be built to suit the environment.
92. The discharge of air from the wet well should be diffused. Consider mixing wet-well air with fresh air from a booster fan.
93. Consider noise problems from odor control or ventilation fans.
94. Packed wet chemical scrubbers work well.
95. Carbon towers work well, but they still allow the escape of a slight odor that is not always acceptable to the neighbors. On heavy-odor days, however, carbon is reasonably satisfactory. The carbon needs to be changed about twice a year, and it is dirty to handle.
96. Hypochlorite-generating towers require lots of servicing time and the hypochlorite generators are too inconsistent. They need to be recharged often, and on heavy-odor days the odor control is erratic. Because of the salt, these units corrode rapidly.
97. Potassium permanganate/alumina pellets require too much maintenance. They are expensive and do not last. *Jones: North Mercer Island pumping station was originally designed with "filters" for these pellets, but the system did not work well (an experience we and others have had) and was replaced with a hypochlorite wet scrubbing system.*
98. Masking agents are ineffective.
99. Proper maintenance equipment must be included.
100. Provide a proper storage area for replacement chemicals.

Overflow Structures

101. Structures should be kept as inconspicuous as possible but easily accessible.
102. Use flap valves rather than hydraulic or electrically actuated valves.
103. A measuring weir with recorder would be useful, together with an alarm to signal when overflow is occurring.
104. A catch basin to collect solids would be desirable to reduce clean-up problems.

Potable Water Supply

105. No significant problems have been identified with potable water systems.
106. All outlets should be above flood level with no possibility of cross-connections.
107. Non-potable outlets should be clearly signed.

Raw Wastewater Pumps

108. Provide easy access to pumps and a clear working space around them.
109. Pumps are much more trouble-free when speed and head are low.
110. Impellers should be nickel-iron or stainless steel for better wear. *Jones: Some types of stainless steel have excellent wearing properties but come at an exorbitant cost premium. We prefer cast iron with 2 to 3% nickel.*
111. Provide guards for all drive shafts.
112. We do not like mechanical seals and are replacing the few we do have with packing glands. *Jones: Mechanical seals generally do not function well where the shaft may deflect because of high radial thrust. Arbour: Vibration is usually the most likely culprit causing premature mechanical seal failure. However, it will also reduce the life of packing. In either case, the pump must be mechanically sound if the expected life of packing or mechanical seals is to be reached*
113. Plumb seal water pressure lines and drain lines to keep them out of the way of the operators. Provide seal water drains at the bottom of pump casings.
114. Valves should be located to make operation easy. Make sure manual actuators have enough mechanical advantage to make valves easy to open and close.
115. Bleed-offs are required. They should be large and plumbed into a drain.
116. Provide covers for volutes during repairs.
117. Provide spare parts.

Security

118. Provide vandal-proof doors, vandal-proof lights, and unbreakable windows.
119. Fencing may be required in some areas.
120. Switch outside lighting on and off by a timer.
121. For stations in remote locations, it is wise to put special security locks on station doors.
122. Avoid long driveways to facilities.
123. *Isaac: We have "locked covers" for manholes that make it difficult for amateurs to stuff debris into the sewer. Sanks: For discouraging vandalism and midnight dumpers, some manholes in Europe have a special, heavy (140-kg or 300-lb) cast-iron hatch (below the manhole cover) that can be lifted only with a hoist.*

Stairwells

124. Stairwells located on the outer perimeter of the station leading to the various doors are preferred.
125. Fiberglass treads are preferred.
126. Aluminum handrails are preferred over galvanized iron.
127. Provide adequate headroom over stairs.
128. Circular stairs are better than ladders or ship-type companionways, but straight stairs with intermediate landings are best. *Sanks: Design stairs so that it is easy to carry an injured worker or a box of tools.*
129. We prefer non-skid, spiral stairwells for smaller stations but like wide, straight steps if there is enough room.

Sump Pumps

130. Sump pumps should be industrial-grade units of relatively large size capable both of passing wastewater solids and of dewatering an entire flooded dry well.
131. There should be at least two pumps. *Arbour: Dual check valves should be provided in the event of valve failure during high wet well conditions. If the valves are installed in the vertical position, the valves must be designed for vertical installation. Sanks: Design so that there is enough horizontal pipe for the two check valves. Jones: Sump pumping stations should never discharge to the wet well but, instead, discharge to the nearest upstream manhole. The whole idea is to eliminate a possible siphon if the pump check valve fails. That has happened on several occasions. The result was a flooded dry well.*
132. Check valves should have an external arm with a spring or counterweight to prevent slam and to indicate whether the pump is pumping.
133. Ease of access is critical. The pumps should be conveniently located in the dry well for ease of cleaning and pump removal. *Arbour: Pits should be provided with a screen to prevent large debris from entering the sump pit during cleaning.*
134. Gratings over sumps should be light in weight and readily removable. Pumps should be easy to flush out. Grease collection on the surface of sumps is a problem that takes either manual removal or a lot of hosing down.
135. Built-in floor drains are preferable to open channels along one wall. The floor drains should be of relatively large size.
136. Lead/follow switches should be provided.
137. Add an alarm so that operators can tell when the follow pump is activated.
138. Either air bubblers or float controls are satisfactory.
139. We have numerous submersible sump pumps as well as submerged, extended-shaft pumps with encapsulated motors located above the pumps. Although both types are acceptable, we prefer the latter. Our experience with submersibles has been less satisfactory, possibly because the pumps have not been rated for the intended service. *Isaac: Many submersible pumps are not designed for passing wastewater solids—many are too light for this duty, and seals tend to fail and cause motor shorts. Jones: Once the shaft seal fails, the motor windings are sure to go next and that is the first indication to the operating staff that there is a problem. After much trouble with submersibles, we switched to wet pit pumps with the motors above grade where they are less vulnerable and the operator can service them. According to NFPA 820, submersible pumps in sumps receiving wastewater-contaminated drainage must have explosion-proof enclosures. Sanks: Submersible pumps can be satisfactory if they are carefully selected for the service conditions, are of rigid design, and have moisture probes to warn of seal failures.*

Support Systems

140. We are firmly convinced that support features of a pumping facility are of paramount importance. Some support systems that can make a major difference in ease of maintenance (but are sometimes given too little attention) include: (1) sump pumps; (2) cranes, hoists, lifting eyes, and access hatches; (3) alarms; (4) doorways; and (5) landscaping. *Sanks: These topics are discussed herein under separate subheadings.*

Switches

141. MANUAL/OFF/AUTO switches are too easily left on MANUAL with the possibility of damaging the equipment.
142. Either use PUSH-TO-TEST/OFF/AUTO or add a timer to the MANUAL position that automatically turns the switch to AUTO after a preset period. *Arbour: Lock-out/tag-out procedures must be strictly heeded for staff safety.*

Telephones

143. For safety and convenience, every station should have a telephone—particularly in deep stations. *Jones: Portable phones (often carried in the maintenance vehicles anyway) are good substitutes. Arbour: Cell phones and radios rely on line-of-sight communication. A repeater may have to be installed above ground for radios or cell phones to work.*

Toilets

144. Every station should have a toilet and washing facilities for the convenience and safety of operators and maintenance personnel.

Valves

145. Locate for easy isolation of pumps and force mains.
146. It should be possible to close all valves manually if necessary, and overhead valves should be activated with a chain. Large valves should have a setup for operation with an air wrench.
147. Do not use knife valves on discharge lines because they tend to leak. *Isaac: Some manufacturers make quality knife valves that are adequate for this service and specifications can be written to ensure this quality.*
148. Use gate valves for pig launchers.
149. Force mains discharging into channels should have flap gates instead of valves. (See Figures 12-36 and 12-38.)
150. Valves over walkways or located in walking areas should have a clearance of 2.1 m (7 ft).
151. Avoid cheap gate valves on discharge lines. *Isaac: Avoid cheap valves everywhere. Arbour: Gate valves require periodic exercising (as do other valves) and will likely fail to operate as material accumulates in the seats and on the operating screw. Consider the use of plug valves for isolation.*
152. We have no problems with hydraulic systems to operate valves. *Jones: Hydraulic systems unquestionably require more maintenance and are messier than electric actuators, but hydraulic actuators are better for protection against hydraulic transient conditions and flooding.*

Variable-Speed (V/S) versus Constant-Speed (C/S) Pumps

153. Variable-speed pumps are preferred and seem to do a better job of keeping the wet well clean.
154. Our types of V/S drives include liquid rheostats, eddy-current couplings, and adjustable-frequency drives (AFDs). All have been satisfactory, but each has shortcomings. *Arbour: Liquid rheostats (using wound-rotor motors) are low-efficiency, high-maintenance devices, and so are eddy-current couplings. In both cases, they are likely to become obsolete and spare parts will be difficult to obtain Sanks: See Item 156. Maintenance of the brushless type of eddy-current coupling is no greater than for an electric motor. Eddy-current couplings with brushes do require periodic replacement of brushes and occasional dressing of slip rings.*
155. Liquid rheostats are relatively slow in response to wet well level changes. They have more maintenance problems. They can become fouled by electrolyte leaks. Changing electrolyte should be done with care and only by trained technicians.
156. Eddy-current couplings are the most maintenance-free system in use. They run well with very few problems and have proven to be very reliable. But we have had difficulty recently in obtaining parts from certain manufacturers who no longer build them in the smaller sizes, 56 to 112 kW (75 to 150 hp).
157. AFDs are quick in response, but they are subject to nuisance shutdown from small changes in incoming line voltage. All of ours are sensitive to voltage dips. Overheating is a problem. In general, AFDs give more trouble than other types of variable-speed drives. *Sanks: AFDs have recently become more versatile and robust. Automatic restart with power line synchronization is obtainable and should always be specified. Some drives now produce a nearly perfect sine wave without the spikes that cause heating. The overheating problem at low speed can be solved by adding a fan driven by a small motor directly connected to 60-Hz, ac power.*
158. The control system must be tailored for the application and flow rate.

Ventilation

159. Good ventilation is very important for both equipment and operators.
160. No less than 10 air changes per hour (AC/h) for dry wells and 20 AC/h for wet wells are required to prevent corrosion and odors. *Jones: We recommend 10 AC/h for pump rooms and a minimum of 20 AC/h for wet wells and screen rooms (40 AC/h in southern states where the wastewater is warmer and the need for hydrogen sulfide protection is greater).*

161. The general design principle of using both inlet and exhaust fans to introduce air at the ceiling and discharge air into ducts at the floor has been effective in providing a pleasant work environment and protection of the equipment. For example, there has never been condensation on any equipment—even on tanks. *Sanks: Proper ducting is just as important as the number of air changes per hour.*
162. If the dry well is vented to the wet well, set up the duct work so that a high wet-well level cannot flood the dry well. A few problems in early designs were corrected in subsequent designs. The dry well in the Bellevue Pumping Station was flooded shortly after being placed in service. Power failure caused a high wet well level and allowed wastewater to flow through a ventilation duct between the wet and dry wells. *Jones: Any ducting whatsoever between wet and dry wells is now prohibited by NFPA 820.*
163. Sail switch alarms in the ventilation ductwork systems were included in many of the early stations and were subsequently connected to the main treatment plant control centers over leased telemetry lines. These caused numerous nuisance calls to check stations that could have been serviced during the next routine inspection.
164. The high (10 to 20 AC/h) air exchange rates specified in early stations has created some difficulties when trying to retrofit facilities with odor control equipment. Replacing fans or adjusting sheave sizes has largely alleviated this problem.
165. If exhaust gases must be scrubbed for odor control, high-volume ventilation becomes very expensive. *Jones: This consideration puts the engineer at a decided disadvantage. Ventilation at lesser rates (than above) creates, in effect, confined spaces that require three-person entry teams and safety equipment. It also exposes the engineer to the potential for liability claims if an accident occurs. Given the value placed on human life, the added expense of an odor control system for the greater ventilation rates seems a justifiably needed expense.*

Wet Wells

166. The long, narrow, self-cleaning (trench-type) pump intake basin works well in practice. All of our future pumping stations will be of this type.
167. Seattle Metro's wet wells are pumped and hosed down at least weekly and even more frequently in troublesome locations.
168. The older, "traditional" wet wells for constant-speed (C/S) pumps were dangerously odiferous and huge piles of grit accumulated rapidly.
169. We prefer suction bells that terminate near the floor over other types of intakes.
170. Provide high-pressure "hose-down" facilities. *Sanks: Supply at least 1.6 L/s (25 gal/min) at a pressure of no less than 550 kPa (80 lb/in.^{2}) at the nozzle.*
171. Smaller wet wells, where storage can be provided in the sewer lines (and not in the wet well) have fewer problems in general.
172. Minimize control instruments in wet wells.
173. Eliminate heavy gratings that must be lifted to gain access to the wet well for hose-down and/or maintenance. (See "Gratings.")
174. In other cities, wet wells are often dark, dismal, and poorly ventilated—a scourge for operations. Yet operators must enter them frequently to control odors and remove grease and accumulated solids. Some of the features that make frequent wet well entry less objectionable are:

    - Easy-opening, counterbalanced access hatches.
    - Ventilation systems that provide enough air exchanges (15–20 AC/h) to prevent condensation and remove potentially harmful gases.
    - Wash-down stations with local explosion-proof START/STOP switches to provide large volumes of water at high pressure.
    - Flood and explosion-proof lighting that provides good visibility for any operation chores.
    - Wet wells with no bar screens. Our experience has been that with V/S pumping, flow and debris come into the station constantly and uniformly, so they move through the system with minimal clogging. Our experience with inlet bar screens has not been very satisfactory. When operators rake the screens, a considerable amount of debris reaches the pumps in bunches and tends to clog them. Most stations are not equipped with facilities to handle removed screenings except to hand-carry them in buckets—a sure way to make operators unhappy. *Bosserman: Provide screenings removal equipment at stations where bar screens are deemed necessary. Jones: If screenings removal equipment is provided, then screening treatment equipment (washing, dewatering, and pressing) and facilities for storage and removal are required to prevent odors and minimize labor-intensive operation.*

24-11. Survey of Two Thousand Wastewater Pumping Stations

During 1996, one of the editors [17] investigated the performance of a great many raw wastewater pumping stations in 14 large and (1 small) sanitation agencies in the United States [18, 19, 20]. The initial part of the project (February and March) consisted of telephone interviews with O&M supervisors and chief engineers at 15 large public sanitation agencies. These 15 agencies operated a total of more than 2700 raw wastewater pumping stations. Based on the interviews, four agencies (City of Houston, Texas; City of Baton Rouge, Louisiana; City of Nashville, Tennessee; and King County Department of Metropolitan Services, Seattle, Washington) were visited in April. These agencies operate a total of more than 1000 pumping stations. Interviews were conducted with O&M personnel to determine what problems they frequently encountered and to establish their preferences for improving O&M characteristics of the pumping stations. Finally, more than 50 pumping stations were visited. Operations were observed, and the actual station O&M personnel were interviewed. The following is a short summary of some of the key issues that were identified.

1. How Not to Design a Wet Well

The design engineers for several wastewater pumping stations built between 1985 and 1995 in one agency insisted that baffle walls between pumps needed to be designed into the wet wells to provide quiescent conditions for the pumps and to provide greatest pump efficiency. The resulting baffle wall system did just that. It provided *very* quiescent conditions. As a result, the grit and solids, settling to the bottom of the wet wells, caused severe odor problems and sometimes clogged the intake pipes to the pumps. It takes an eight-person crew to remove the grit from the wet wells because the wet wells are considered to be confined spaces. Willing to accept a little reduction in pump efficiency in exchange for eliminating the cost of removing the grit that accumulates in wet wells with baffle walls, the agency no longer installs them and has begun to demolish the existing ones. *Sanks: The designers did not heed the warning in the Hydraulic Institute Standards: All Hydraulic Institute Standard wet wells (up to and including the 1994 edition) are for clear water only. If they are nevertheless to be used for solids-bearing waters, provisions for cleaning accumulated grit and scum must be made. Deposition occurs wherever currents decrease much below about 0.6 m/s (2 ft/s). Demolishing the baffle would make vigorous mixing (as a means of solids suspension and removal) more effective. See Sections 12-6 and 12-7 for viable alternative designs.*

Bosserman: For a conventional wet well design, consider the removal of scum from the water surface and grit and other settleable solids from the bottom of the sump. Unfortunately, only since 1998 (with some exceptions) have designers and published design standards or guidelines begun giving this factor the serious consideration that it deserves. Access must be convenient for bringing in cleaning equipment such as hoses to wash grease off the sump walls and vacuum hoses or other means to remove the accumulated sediments. The wet well walls and ceiling should be lined with smooth PVC to prevent the attack on concrete by sulfuric acid (derived from hydrogen sulfide) above the water line and to make it easy to wash grease off the walls. Gratings, sidewalks, or landings that do not interfere with hoses should be provided for the convenience of the cleaning crews.

Sanks, 2004: If ANSI/HI 9.8-1998 is followed, designers are required to plan for the cleaning of wastewater wet wells. See Chapter 12 for designs that are particularly appropriate for cleaning.

2. More Comments on Wet Well Design

The vast majority of the dry well pumping stations in these agencies have sealed wet wells without ventilation. They contain a "breather" vent pipe, typically 150 mm (6 in.) in diameter, coming out of the wet well roof. When O&M personnel have to enter the wet wells (which is not very often), they use portable blowers and ducting to supply the necessary ventilation. In the latest designs, the wet wells are lined with PVC. One agency (King County Department of Metropolitan Services) that does desire routine and convenient personnel access to the wet well used the following design in a recent pumping station (Interurban), started up in March 1996:

- A catwalk of FRP grating over the entire length of the wet well about 0.6 m (2 ft) above the high-water level, so that personnel can walk over the entire wet well. The clearance between the catwalk and the walls is about 0.3 m (1 ft) to permit unobstructed wash-down of the walls.
- Variable-speed pumping, so that the wet well water level does not fluctuate very much.
- Explosion-proof (NEMA 7) lighting fixtures, accessible from the grating for replacement.

- Walls and ceiling are lined with PVC.
- No screens or grinders.
- The ventilation system is so effective that there is hardly any "wastewater" odor, in spite of the fact that raw wastewater is just 2 ft below the grating. There are several ventilation ducts with diffusers on each, to distribute the air into the wet well.

3. Example of a Design Engineer's Unwarranted Optimism

A consulting engineering firm decided to use close-coupled vertical wastewater pumps because the dry well was of a design that could not possibly be flooded. There was a solid, unpierced concrete wall between the wet well and the dry well from the bottom slab of the structure all the way up to the roof slab of the dry well containing the pumps. The following sequence of events occurred one day:

- There was a storm, during which a power failure occurred.
- In accordance with Murphy's Law ("If anything can go wrong, it will"), the standby engine generator failed to start.
- There was a rest room with a toilet on the roof slab over the dry well. The toilet was connected via a pipe to the wet well. There was no check valve in the pipe. The wastewater backed up in the wet well, flowed up the pipe, over the toilet, across the floor, and down the stairway into the dry well that "couldn't" be flooded. *Sanks: With immersible motors, there would be no pause in pumping either during flooding or cleanup.*

4. Maintaining Submersible Pumps in Dry Pits versus Those in Wet Wells

All the O&M personnel at all the agencies visited felt that it is much easier to maintain a dry pit pump, unless the submersible pump was manufactured by one specific manufacturer. This one manufacturer (a large corporation) was almost unanimously acclaimed as having the most reliable pump on the market, with O&M requirements that were comparable to those for conventional dry pit pumps. *Three* other manufacturers were frequently cited as providing good pumps, although not quite as good as Manufacturer No. 1. *Sanks: In the intervening years, other manufacturers have developed reliable, excellent pumps.*

- Their main reason for stating that it is much easier to maintain the pump in a dry pit pump is that O&M staff can inspect it and tell when something is amiss and repair it before the problem becomes too serious. Such inspection is impossible with submersible pumps installed in a wet well because they cannot be observed while running and must be hauled out of the wet well for repairs. Consequently, there is a greater tendency to let a wet well submersible pump run until it simply fails. Then the cost of repair is usually very high.
- Submersible pumps in wet wells are quite variable in frequency of maintenance required. Some agencies have had to pull submersible pumps out of wet wells as frequently as every six months. Others have gone as long as six or seven years before having to pull the pumps for maintenance. The typical interval seems to be about two years. Seal and bearing failures are the most common problems with submersible pumps. Damage to power cables and motor windings occurs less frequently.
- It can take a crew of 6 to 10 workers to remove a submersible pump from a wet well. Depending on the size of the pump, the crew typically consists of the following workers:

  a. 1 or 2 crane operators
  b. 2 or 3 electricians
  c. 3 to 5 other persons to actually remove the pump.

- One agency reported that it takes a six-person crew up to half a day to remove a submersible pump from a wet well. If maintenance then has to be performed, additional time and labor are needed to transport the submersible pump to the maintenance shop, repair it, transport it back to the pumping station, and reinstall the pump. A conventional dry well pump can be repaired with a fraction of this time and effort. However, this agency also felt that good-quality submersible pumps (of which there are not very many manufacturers, in their opinion) are more reliable than the conventional dry well pumps in the sense that they do not fail as often as do dry well pumps.
- Most agencies have discovered that O&M costs for submersible pumps larger than 75 to 150 kW (100 to 200 hp) become extremely high. Generally speaking, they discovered that:

  a. Submersible pumps and motors larger than 75 kW (100 hp) break down more often than smaller ones.
  b. Removal of large submersible pumps from wet wells takes a lot more effort than they had originally thought. In general, it requires more

personnel to remove submersible pumps from wet wells than from conventional dry pits. With large pumps (greater than 75 kW or 100 hp), workers in at least one large public utility usually enter the wet well to disconnect the electrical power cable from the pump, unseat or disconnect the pump from the connecting piping, and help guide the pump out of the wet well. When the pump is reinstalled after being repaired, workers must again enter the wet well to reconnect the power cable and reconnect the pump to its discharge piping. As the wet well is classified as a confined space, this effort must be made by workers wearing rubber suits and air masks and accompanied by observers and supervisors above. *Sanks: Contrast that scenario with the experience of a large utility that manages 425 pumping stations of which 170 have submersible pumps, including many large ones. No worker ever goes down into a submersible pump wet well except to work on damaged guide rails or brackets. Pumps of 150 kW (200 hp) or less can be pulled up or replaced in less than half an hour by a crew of two—an electrician and a boom operator. Aided by a flashlight, the electrician swings a hook to engage the pump bail. (The lifting chain is not used because it requires a long boom.) A keeper prevents disengagement. The pump is lifted straight up by the well-positioned boom and deposited beside the floor hatch with its cable still connected. In a demonstration, it took 13 minutes to position the truck and boom, hook onto a 75-hp pump bail, lift the pump, and set it on the ground floor. Minor service (such as changing oil or checking shafts for deflection) is completed on the pump while it rests horizontally in a cradle beside the wet well hatch. The cable is disconnected at the motor starter and accompanies the pump and motor to the shop for major repairs such as the replacement of seals and bearings. The power cable is not disconnected from the motor. The difference between the two kinds of operations lies in the training of the workers, attitude (do the best you can with what you have), and designs that make these operations easy (see Section 25-6). Rounding the edges of the hatch so as to avoid damage to the power cable is helpful. Aggressive management and operator training are the keys to good operation at the lowest cost.*

c. The frequency of repair of submersible pumps installed in wet wells is greater than that for either conventional or submersible pumps in dry pits. One agency thinks (although exact records are not available) that such submersible pumps must be repaired 20 to 40% more often than dry pit pumps. *Bosserman: These figures are a composite or average of all different manufacturers and models installed in the system. There is a limited number of high-quality submersible pumps installed in wet wells that offer much superior O&M performance—see the first paragraph in Subsection 4. Submersible pumps in dry wells have O&M costs equal to or less than those of conventional dry well pumps. See Subsection 25 for more discussion.*

- A majority of the agencies send all their submersible pumps to private shops for repair. The most common repairs are replacing bearings, replacing seals, and rewinding motors. Typically, it cost \$4000 to \$8000 to repair a submersible pump, and sometimes much more. For example, it could cost as much as \$27,000 to replace the rotor and stator on a 160-kW (215-hp) submersible pump. Use an ENRCCI of 5600 for these costs (see Chapter 29).
- It can cost \$600 to purchase a new upper bearing assembly for a submersible pump. A lower seal assembly for a submersible pump can cost \$2400. By comparison, the seals (conventional packing) and bearings for a dry pit pump can be purchased for only \$200.
- In the opinion of the O&M staffs at most of the agencies, the degree of effort required to maintain various types of wastewater pumping stations (from easiest to most difficult) is as follows:

  a. Submersible pumps in a dry well.
  b. Conventional dry well pumps in a dry well.
  c. Submersible pumps in a wet well.
  d. The sales representatives for submersible pump manufacturers tend to be very optimistic regarding the effort that it will take to maintain a submersible pump in a wet well. A careful and realistic life-cycle cost analysis should be performed before deciding to use submersible pumps, especially if the contemplated pumping station is fairly large—greater than 440- or 660-L/s (10- or 15-Mgal/d) capacity.

5. Depth of Wet Wells for Submersible Pumps

It was observed and reported that in deep wet wells, the heavy electrical power cables to the pump motors

can sometimes stretch and crack. The turbulence in deep wet wells can also cause the long cables to whip and bend, thereby damaging the cover, the power wires, or especially the connection at the motor cap. When a crack develops in the cable cover, hydrogen sulfide can enter the cable and corrode the power wires. It is expensive to purchase a replacement power cable and it sometimes takes four to six weeks to obtain a new cable from the manufacturers, most of which are overseas. *Sanks: Stretching can be controlled by fastening the power cable to a stainless-steel wire rope designed to carry the tension. Some manufacturers regard cable replacement as emergency repair and keep cable on hand so that replacement may require as few as three work days.*

Another factor in wet well depth, especially important for removing large pumps, is that the long, thick power cables cannot be bent to a short radius without damaging the cable. Consequently, it requires much care and effort to lift both the pump and the cable together without overstressing the latter. It is for this reason that some agencies disconnect the cable from the pump before lifting the pump out of the wet well. Agencies with experience in removing large submersible pumps prefer to limit the maximum allowable wet well depth to about 12 m (40 ft). *Sanks: Excessive flexure near the motor connection can be controlled by a cable saddle or by the use of semi-flexible stainless-steel cable sheathing that controls the radius of bending. It may be preferable for the cable to exit downward from the motor caps of large pumps to prevent excessive bending when the pump (with cable attached) is lifted. Cable sheathing or a saddle is still required because the cable must be tight during normal pumping operation to keep it from swinging with the currents and thus wearing out the cable connection at the motor cap.*

6. Discharge Elbows for Submersible Pumps

A major problem sometimes encountered (fortunately, not very often) with submersible pumps is the discharge elbow in the wet well tearing loose from its anchor bolts. When that happens, it requires a major effort to repair it. The wet wells are classified as confined spaces per OSHA, so it takes an eight-person crew to do the work in compliance with all the confined-space observation and supervision requirements. It is believed that the cause of this problem is as follows:

- A pump ingests a large rock or some other solid.
- The pump then vibrates as it tries to expel the solid.
- This vibration creates forces and movements in the pump and its discharge elbow that are much greater than those the elbow anchor bolts and associated concrete anchoring system were designed to withstand.
- Each time such an event occurs, it weakens the concrete and anchoring system a little more, and eventually the bolts tear out of the concrete or the concrete breaks, and the agency has a major repair problem. *Sanks: Beware of light anchor bolts. A bolt smaller than 22 mm (7⁄8 in.) is a toy. Heavier bolts cost very little more than smaller ones. Bolts should (1) be buried deep enough to develop their ultimate tensile strength, and (2) be surrounded by a strong cage of reinforcement to prevent cracking of the concrete.*

7. Variable-Speed Pumping

Many agencies use variable-speed pumping extensively. Those with significant experience feel it is a very effective and efficient means of operating raw wastewater pumping stations. Most new variable-speed units are adjustable-frequency drives (AFDs). One AFD manufacturer in particular was cited as providing especially reliable equipment and excellent service and training support. The drives feature insulated gate bipolar transistor (IGBT) technology, which results in an extremely quiet design compared with that of other older units. There is only a slight 60-cycle background hum, compared with the ear-piercing noise sometimes produced by older designs. In addition, the IGBT design runs cooler and does not emit nearly as much heat, thereby reducing the air conditioning requirement for the control room.

8. Standby Engine—Generators

Although standby power is usually thought to be essential for raw wastewater pumping stations, there are sometimes exceptions to this rule. One agency (City of Houston) has none in any of their 330 pumping stations. Another agency (City of Baton Rouge) has none in more than 400 pumping stations. Both have concluded that their power supply is very reliable, and there is no need for standby generators. Some of their older stations have connections for use with portable generators, but the newer stations do not. These agencies have determined that the hydraulics of their systems allows them to withstand power outages of three to five hours duration before

any wastewater spills occur, and the power failures have never lasted that long. Both have portable generators (up to 400 kW) that they can use when needed, but they are typically only used 10 to 12 times per year. Not having to provide permanently installed standby generators or dual power feeds to these several hundred pumping stations has saved considerable construction cost. *Sanks: Each of the above agencies may have both a fortunate and a unique situation. Each facility should be evaluated for standby power need on the basis of size and the consequences and probability of power failure. See Section 13-14 for developing a more balanced viewpoint and for examples of power outages that have caused disasters. Arbour: There are numerous options for providing reliable pump station operation in the event of a power failure. These include on-site generators, portable generators with fixed connections and manual transfer switch, dual feed, upstream storage, pump-around provisions using portable pumps (engine-driven or electric if a portable generator is available), and the use of tank trucks or combination machines (usually in small-capacity stations). The designer must consider the level of service, environmental impact, and regulatory compliance in determining which of these options is suitable for minimizing the risk and consequences of failure.*

9. Odor Control

Most O&M staff stated that they prefer packed carbon towers because they are easy to maintain. Both the activated-carbon type and the impregnated-carbon type have been used. *Jones: The impregnated-carbon type has a well-established poor safety record (fires, hazardous chemicals) and is not recommended.* The carbon systems last a long time. At some pumping stations, the carbon has not had to be changed in over three years. Wet scrubbers (such as caustic soda and sodium hypochlorite) are high-maintenance systems. Sodium hypochlorite corrodes everything it touches when it spills and is a difficult system to maintain.

10. Flowmetering

Four agencies (City of Houston, Texas; Ann Arundel County, Maryland; City of Orlando, Florida; and Frederick County, Department of Public Works, Maryland, with a combined total of more than 900 raw wastewater pumping stations) have no flowmeters at any of their pumping stations. They just read the elapsed time meters (ETMs) for the pumps and, knowing the approximate capacity of each pump, they can then approximate the flow rate. They see no need for greater accuracy. Flowmeters are installed at the treatment plants where the flow data are needed. Typically, flow measurement devices of some sort are needed only at pumping stations to indicate flow build-up over time to allow the O&M staff and engineering staff to determine when they need to add additional pumps or to expand a pumping station. In the opinion of these agencies, time meters furnish adequate data.

11. Pump Speeds

Most agencies prefer no more than 1200 rev/min speed for any pump (dry pit or submersible) or perhaps even no more than 900 rev/min maximum speed. Some of the pumps run at speeds up to 1800 rev/min. These higher-speed units wear out much more quickly than pumps operated at 600 to 1200 rev/min. Packing has to be replaced more often, and the bearings and shaft sleeves wear out faster and must be replaced more often. In general, pumps with operating speeds above 1200 rev/min have significantly higher O&M costs than slower-speed units. *Sanks: Wear varies approximately as speed to the third power.*

12. Cranes and Monorails

Almost all agencies have permanently installed bridge cranes or monorails in their conventional dry well/wet well pumping stations. In the opinion of the O&M staffs at several of the agencies, design engineers need to do a better job of considering the crane layout and how the cranes will be used. The O&M personnel also stated that they need more access space around equipment to be able to use the crane systems effectively. In some of the pumping stations, the crane layout was so poor that rework was necessary to make them function properly. For example:

- Some cranes were designed so that that the crane blocks had to be operated at an angle to remove pieces of equipment—a violation of OSHA crane safety regulations.
- In another pumping station, a bridge crane was supposedly designed so the trolley could be centered over the motors and pumps for removing them. Unfortunately, the designers placed a 90-degree elbow and discharge riser directly on the pump

discharge nozzle. When the trolley was moved toward the motor, it struck the riser pipe and could not be centered over the motor or pump.

- In several pumping stations at four agencies, the overhead bridge cranes could not be easily used because of interferences from structural beams. Better coordination is required between the mechanical engineers who lay out the pumps, piping, and crane systems and the structural engineers who design the structural beams and supporting columns. *Sanks: Project leaders, take note!*

13. Accessibility of Equipment

A nearly universal complaint of the O&M staffs was that more space or clearance is needed around equipment for maintenance access. In some of the pumping stations visited, the access was so cramped that maintenance personnel could not even remove the bolts to the pump flanges without a major team effort. In others, access to the stuffing boxes was restricted so that changing the packing was difficult. Designers need to give much better thought to important issues such as:

- How will pump shafts be removed so that the shaft sleeves can be replaced?
- How will impellers be removed and replaced?
- How will flanges be disassembled and reassembled?
- How will the top covers on check valves be removed?

Designers need to keep in mind that all these maintenance activities require a team effort of two to four people, and there must be sufficient space around the equipment so they can work together effectively. *Bosserman: The increased cost of the pumping station structure to include adequate space for team O&M effort is insignificant when compared with the total pumping station construction cost. That is certainly true when the total life-cycle costs are compared if an allowance is made for higher O&M costs due to lack of space for effective and efficient maintenance.*

14. Screens and Grinders

Generally speaking, most agencies have no screens or grinders in any of their pumping stations. Typically, O&M staffs see no need for them and consider them to be high-maintenance items. At some older stations, the existing screens have been taken out of service because they served no real purpose and required maintenance. Of the 15 agencies investigated, only one (City of Nashville, Tennessee) required screens or grinders at all of their stations. That agency reported they consistently get large quantities of construction debris (pieces of brick and masonry, lumber, drill bits, bolts, nails, etc.) in their wastewater and they need to have screens or grinders ahead of the pumps to avoid excessive wear of the pumps. This problem does not, however, appear to be a typical or common occurrence in municipal sewage systems.

15. Check Valves

Most agencies normally use hydraulically buffered swing check valves or at least require springs or weights on the lever arms of the check valves. In pumping stations built within the past 5 to 10 years, the trend toward hydraulically buffered controls on the swing check valves was especially evident. *Sanks: See also Section 5-4 and Figures B-2 and B-3. With a heavy spring or counterweight, the liquid in the valve is a hydraulic buffer as effective as a dash pot (although the spring increases the headloss).*

16. Mechanical Seals

Another definite trend was the use of mechanical seals in the pumps for new pumping stations. Agencies that have investigated mechanical seals most extensively prefer the split-seal type with tungsten carbide (and not ceramic) faces. Split seals are easy to install and disassemble. Leakage is much less with a mechanical seal than with packing, and the mechanical seal lasts much longer than packing. One agency reported that in their life-cycle cost analysis, they concluded that a mechanical seal pays for itself in three years because of the reduced maintenance requirements. The mechanical seal lasts 10 years—an excellent bargain, in their opinion.

17. Packing

Another trend observed, especially in those agencies that have not switched to mechanical seals (Orange County Sanitation District and City of Baton Rouge), was the use of packing materials that do not require an external clear water flush. At least two agencies have

acquired extensive experience with such materials. One agency (Orange County Sanitation District) has been using one of them [21] for more than 15 years, and they report excellent success. The packing typically lasts two to three times as long as conventional packing. The sealant material is so fine that it screens out wastewater solids that leak up the shaft, so that the solids cannot come in contact with and abrade the shaft sleeve. These agencies no longer even have the flushing water systems installed in their new wastewater pumping stations.

18. Level Sensors in the Wet Wells

Bubbler systems are by far the most popular wet well level sensing devices at all of the agencies. The bubbler systems have the fewest problems and are the easiest to maintain. Various agencies have also used floats, pressure transducers, and ultrasonic level sensors.

- Floats were generally disliked. They became caked with grease and scum, and were ruined by the turbulence in the wet wells. Even placing the floats in stilling wells or pipes in the wet wells does not solve the turbulence problem. Floats were regarded as high-maintenance items and were considered to be much less reliable than bubbler systems. Most agencies used floats only for emergency high-level alarm service but no longer for controlling pumps.
- Only one agency (Los Angeles County Sanitation Districts) reported that they preferred pressure transducers. This agency has developed a design that consists of a pipe spool mounted against the dry side of the wet well wall near the dry well floor. A gate valve is attached to the spool and the transducer is attached to the gate valve. A 12-mm ($\frac{1}{2}$ in.) water flushing line is connected to the spool, and the water stream keeps the pipe spool clear of grease and solids. The design is reported to be extremely effective and almost maintenance-free. The other agencies have not reported success with their designs, but none of them used the wall spool mount with the water flushing system as did the one successful agency. See Section 24-9.
- One agency (King County Department of Metropolitan Services, Washington) has tried several manufacturers of ultrasonic level devices to monitor the water level in their wastewater pumping station wet wells. They were able to find only one manufacturer whose product worked reliably in such service.

19. Coatings for Impellers, Pump Volutes, and Shaft Sleeves

One agency (City of Baton Rouge, Louisiana) reported that they coat the pump impellers and volutes with IPI Fluid Ceramic or Cerami-Tech C. R.® [22] in two equal layers to a total thickness of 0.50 mm (20 mils). These coatings are very hard, offer excellent abrasion resistance, and increase the life of the impellers and volutes by making them more resistant to abrasion by grit.

They also coat the shaft sleeves by first machining about 1.25 mm (50 mils) off the sleeve and then applying Cerami-Tech E. G.® [22] to build the sleeve slightly oversize. The coating is so hard that, after it cures, it can be machined smooth to its original size only with a diamond-tipped cutting tool.

They apply these coatings in their maintenance shop. Their pump specifications do not require the coatings to be applied as part of the original equipment manufacturer's scope.

20. Electrical Conduit Installation

All the O&M personnel reported that they prefer conduit to be exposed and not embedded in the walls or floors. Over the life of a pumping station, changes are inevitably made to the electrical and control systems, and it is easier to make these wiring changes if the conduit is exposed.

21. Air Release Valves on Wastewater Force Mains

One agency has over 600 air release valves installed in their force mains. These valves are extremely high-maintenance items. They have two teams (of two persons each, plus a fifth person serving as a supervisor) whose only duty is to remove air release valves from the force mains and bring them back to the maintenance shop for cleaning and repair. Additional staff then does the cleaning and repair work. The most common problem is that the valve outlets or connections become clogged with grease and scum, thus making them useless.

- Each valve is sandblasted to remove build-up of grease and crud. The internal parts are then cleaned or replaced.
- The staff can remove, clean, and reinstall 10 valves per day. Hence, each valve is cleaned every 60 working days (about once every three months). Even this level of effort does not keep all the valves

in operating condition 100% of the time. *Bosserman: Any agency contemplating the use of traditional air release valves in wastewater force mains should consider the high O&M effort that will be required to keep them operating. If this effort is not provided, the valves quickly become useless and were better not installed at all, because they give a false sense of security. Sanks: There is now a type of valve [23] that overcomes the excessive maintenance required of traditional valves. See Section 5-7.*

22. Materials of Construction

All the agencies are much more aware of corrosion than they used to be, and now insist on corrosion-resistant materials. The following materials are typically used by all the agencies for various pieces of equipment or components in pumping stations constructed within the past 10 to 15 years:

- Stairway treads, landings, grating, access platform decks, etc.: fiberglass reinforced plastic (FRP) or aluminum.
- Outdoor electrical cabinets: stainless steel.
- Handrails, access hatches, HVAC ductwork, and doors: aluminum.
- Odor control system ductwork: FRP.
- Chain-link fencing: PVC-coated for corrosion protection. Galvanized fencing near wastewater pumping stations eventually corrodes because of the minute concentrations of hydrogen sulfide. The PVC is typically green, although red is available (and definitely eye-catching!).

23. Lighting

The O&M staffs typically now prefer halide lighting. Fluorescent lighting is too dim for dry wells. Some of the O&M staffs also felt that the designers underestimate the level of lighting required for them to see easily what they are doing during maintenance. Newer pumping stations typically have the walls and ceilings in the dry wells painted off-white or beige. The light color gives a noticeable improvement in the overall brightness of the room and helps to provide better lighting for maintenance.

24. HVAC Design for Control Rooms Containing AFD Equipment

Control rooms containing adjustable-frequency drive (AFD) control panels in four pumping stations in two cities had heating, ventilating, and air-conditioning (HVAC) systems so poorly designed that extensive rework was required after construction. Apparently, the HVAC designers did not consider the heat loads emitted by the AFD equipment. In one station, the rework consisted of adding two outdoor air-conditioning units and additional ductwork to connect them to the control room. In another station, the cooling system had to be more than doubled in capacity. Temperatures as high as 61°C (142°F) occurred prior to the rework. In all four pumping stations, the AFD equipment was not of the IGBT design discussed in Subsection 7.

25. The State of the Art in Wastewater Pumping Station Design

A majority of the agencies investigated prefer submersible pumps installed in a dry pit. Their most recent pumping stations use this concept for the following reasons:

- Submersible pumps eliminate the lineshafts associated with conventional dry well pumps. These lineshafts can require a lot of maintenance, especially with pillow-block bearings. When the pillow-block bearings fail and are replaced, extraordinary effort is required to reinstall and realign the lineshafts.
- Submersible pumps eliminate the O&M costs associated with shaft and equipment vibration, which is never zero (regardless of what anyone says).
- Submersible pumps eliminate the initial construction cost of catwalks for access to the intermediate bearings, lineshaft supports, etc.
- Submersible pumps allow much better access from the overhead bridge cranes because there are no interfering lineshafts, catwalks, or bearing support structures in the dry well.
- Submersible pumps are noticeably quieter than conventional dry well pumps.
- *Sanks: In addition to being vibration-free, submersible pumps are more compact and cleaner. They eliminate the need for a seal water supply. On the other hand, frame-mounted WP-1 motors and dry pit pumps are less expensive in both first cost and maintenance. Even if a dry well is flooded (a rare occurrence), the motors can be dried out in a shop, the bearings replaced, and the motors put back into service in (usually) a week. Furthermore, TEFC immersible motors from 11 to 1300 kW (15 to 1750 hp) are available; they can run for two weeks submerged under 9 m (30 ft) of water. Furthermore, energy costs are slightly greater for submersible*

pumps. Balance the risk of flooding and its consequences with the cost of submersible pumps or immersible motors.

24-12. Auxiliary Support Systems in Raw Wastewater Pumping Stations

A survey of operator's opinions at raw wastewater pumping stations in six large southern California pumping stations was conducted by one of our editors [17] in 2002 and 2003. The results of that survey pertaining to auxiliary support systems are given below.

1. Standby Equipment

In some designs, there is only one piece of equipment, such as a pump, for the auxiliary or support systems. In the opinion of the O&M staff, that is a mistake. *Every* pump should have an installed standby unit. It can take days to remove and install a replacement pump, and during that time the entire pumping station may be out of service because this support system is not available.

2. Conveyor Belt System and Storage Bins for Solids from Screens

Access to conveyor parts is critical for maintenance. Even under the conditions of good accessibility to parts and components, it can take several days to fix a conveyor. Under conditions of restricted accessibility, the maintenance time will be even greater. Be sure to provide sufficient O&M space around it. Consider the following requirements in laying out the conveyor and storage bin system in a project:

- The belt is usually pulled from the front of the conveyor. Consider installing a door in the wall opposite the front of the conveyor so that the task can be easily done.
- If you specify that the conveyor is to be installed in an enclosure (perhaps for odor control because the conveyor is used with wastewater or other odorous solids), require continuous access doors on both sides that the enclosure for routine access and maintenance. Removable panels in the enclosure are poor because it can take a lot of effort just to remove and reinstall the panels.
- The conveyor shoes (the supports that the conveyor belt rides on and which in turn are mounted on the drive) are high-maintenance items. Be sure it is easy for the O&M staff to access them.
- Some conveyors require that the drive chain (between the motor and the gear reducer) run in an oil bath. Be sure that it is easy to change the oil, and require a drain outlet and valve.
- Provide access to the pillow-block bearings (on which the conveyor support shafts or rollers ride) on both sides of the conveyor for changing bearings.
- Provide access stairs and catwalks around conveyors mounted more than about 1.5 m (5 ft) above the floor.
- If the conveyor dumps solids into a bin, be sure that the owner has access to a waste truck that can lift the bin and transport it to a landfill. *DO NOT* assume that every city or agency uses a standard bin size. Some cities or agencies have trucks that can only lift or carry a certain size of bin, so ascertain the size. Make sure that the bin is large enough to contain at least three days of solids production while the facility staff is off duty—for example, one-half day on Friday and Monday plus Saturday and Sunday. If the bin is used with wastewater or other odorous solids, design the screenings room to be large enough to contain the bin entirely within the room. (A bin mounted outdoors will likely cause a severe odor problem.)
- Design trench drains in the floor beneath the storage bin to carry away wash-down water and leakage from the bin. Include wash-down hoses and hose bibs.

3. Floor Drains

Design floor drains for every room in a pumping station. *Sanks: Always slope floors at about 1% gradient to the drains.*

4. Air-Conditioning Equipment for the Variable-Frequency Drive (VFD) Equipment Room

Air conditioning is important for the VFD system, as VFDs are temperature-sensitive and will fail if the temperature gets too high. If the VFDs fail, the entire pumping station may be out of commission. The air-conditioning units for the VFDs need to have 100% redundancy (i.e., a standby unit), in the

opinion of the O&M staff. The temperature of the VFDs, or at least the temperature in the VFD room, should be monitored and also connected to the alarm system. Air-conditioning units are of varying reliability, depending upon the manufacturer. The compressors are generally of sound design and construction, but the coils and other components are sometimes of poor quality. These components can fail about every three years (or even more often), so make it easy to replace them.

5. Access to Pump Bearing Lubrication Systems, Seal Water Bladder Tanks, and Pump Suction Piping and Valves

In some designs, the bearing lubrication systems, the seal water tanks, and the pump suction piping and valves are located in a large pit with a grating roof over it. The grating allows O&M staff to walk around the pumps. Frequently, however, the only means of getting down into the pit is by lifting off a section of grating and climbing down a ladder. As it is almost impossible to carry tools and parts up and down ladders, install stairs with a hinged, spring-loaded grating for access.

6. Lubrication Oil System for Pump Bearings

Some manufacturers of lube oil systems use reduced-port ball valves that cause enough pressure loss to inhibit cooling the oil for pump bearings. Specify full-port valves and ensure an adequate flow of oil.

7. Alarms for Sump Pumps

A running sump pump usually means that something is flooding or overflowing, so install alarms for them.

24-13. References

1. Hart, R. J., "Pumps and their systems—A changing industry," *Proceedings of the 19th International Pump Users Symposium*, Houston, TX (February 2002), pp. 141–144.
2. Hodgson, J., "Predicting maintenance costs accurately," *Pumps & Systems Magazine* (April 2004).
3. Hodgson, J., Private communication, September 2004.
4. *Hydraulic Institute Standards for Centrifugal Rotary & Reciprocating Pumps,* 14th ed. Hydraulic Institute, Cleveland, OH (1983).
5. Parris, H. L., and J. L. Seminara, "Factoring human needs into power plant maintainability," *Power,* 127, 82–85 (January 1983).
6. EnviroSeal Engineering Products, Ltd., 1310 Rocky Lake Drive, Waverley, NS, Canada B2R 1R6. Phone 902-861-3464, Fax 902-861-3522, http://www.enviroseal.ca/.
7. NASA, *Reliability Centered Building and Equipment Acceptance Guide*, February 2001.
8. Berry, L. D., "Vibration versus bearing life," *Reliability*, Vol. 2, No. 4 (November 1995).
9. Jones, R. L., C. A. Liekteig, and J. J. Zoppe, "Grouting of small pump baseplates? Is it worth it?," *Proceedings of the 21st International Pump Users Symposium*, Baltimore, MD (2004).
10. Robertson, J. C., "Stamp out soft foot," *MRO Today*, Vol. 6, No. 1 (February/March 2001).
11. Corbo, M. A., and S. B. Melanoski, "Practical design against torsional vibration," *Proceedings of the 25th Turbomachinery Symposium*, Sponsored by the Turbomachinery Laboratory, The Texas A&M University System (September 1996), pp. 186–222.
12. Corbo, M. A., and S. B. Melanoski, "Rotordynamics made simple," *Proceedings of the 15th International Pump Users Symposium*, Turbomachinery Laboratory, The Texas A & M University System (1998), pp. 167–204.
13. Kettle, R., Superintendent of Desert Operations, Los Angeles County Sanitation District. Private communication, August 10, 1989.
14. Lescovich, J., Chief Engineer, Golden Anderson Industries. Private communication, 1989.
15. Redner, J., Sewerage System Superintendent, Los Angeles County Sanitation Districts. Private communication, October 3, 1995.
16. Arhontes, N., Staff Engineer, Maintenance, County Sanitation Districts of Orange County. Private communication, October 3, 1995.
17. Bosserman, B. E., Principal Engineer, Boyle Engineering Corp., Newport Beach, CA (May 1996).
18. Bosserman, B. E., "DG-04 Value engineering study. Report of value analysis," Boyle Engineering Corporation, 1501 Quail St., Newport Beach, CA (July 1996).
19. Bosserman, B. E., A. Will, and S. Collins, "Current concepts in wastewater pumping station design," presented at the 28th Joint Annual Conference of the Chesapeake Water Environment Association and the Water and Waste Operators Association of Maryland, Delaware, and District of Columbia, Ocean City, MD (July 9–11, 1997).
20. Bosserman, B. E., and S. Collins, "Pumped-up pumping stations," *Civil Engineering,* 67(7), 65–66 (July 1997).
21. TP 5400 made by Tom Pac Inc., 7575 Trans-Canada Highway, St. Laurent, Quebec H4T 1Z6.
22. Cerami-Tech C. R.® and Cerami-Tech E. G.® Thortex Division of E. Wood Ltd., Standard Way, Northallerton, N. Yorks, UK DL6 2XA.
23. Vent-O-Mat®. International Valve Marketing, Inc., P.O. Box 56, Plainfield, IL 60544. E-mail limeym@aol.com.

Chapter 25

Summary of Design Considerations

GARR M. JONES
ROBERT A. "RANDY" NIXON
RANDALL R. PARKS
ROBERT L. SANKS

CONTRIBUTORS
Charles T. Blanchard
Pat H. Bouthillier
Roland S. Burlingame
Fredric C. Burton
A. L. Charbonneau
Harry E. Covey
Patrick J. Creegan
Roger J. Cronin
Johannes de Waal
James C. Dowell
Edward J. Esfandi
Richard O. Garbus
Harold D. Gilman
Mayo Gottliebson
L. V. Gutierrez, Jr
Stanley S. Hong
W. Eric Hopkins
George Jorgensen
Frank Klein
R. Russell Langteau
John Leak
Ralph E. Marquiss
Warren H. Mesloh
Stephen G. Miller
Russell L. Parr
Allen W. Peterson
John Redner
David M. Reeser
Marvin Dan Schmidt
Earle C. Smith
Brian G. Stone
LeRoy R. Taylor
R. Daniel VanLuchene
Leland J. Walker
David Walrath
Horton Wasserman
Frederick R. L. Wise
Frank A. Woodbury
Eugene K. Yaremko

The many considerations involved in the design of pumping stations make this summary chapter desirable for project engineers, for designers, for city engineers, for utility managers, and for both beginners and (as a partial checklist with helpful hints) for experienced designers. Topics discussed in other chapters are treated briefly, whereas topics such as structural engineering not mentioned elsewhere are covered in more detail. As in other chapters, published codes and specifications are referenced by number only.

The purpose of the preliminary engineering phase of a pumping station design project is to make the major decisions that establish such things as:

- Whether a pumping station can be avoided in favor of an alternative solution
- The best site (with all factors considered)
- The type of pumping station and pumps
- The kind of drivers (constant- or variable-speed, electric or engine)
- The need for (and type of) standby power
- The type of construction: excavation with sloping sides, sheeting, cofferdam, or caisson
- Extent of automation, control, and data recording and/or transmission
- Force main route and profile
- Auxiliary systems such as cranes and chlorination.

Good "front-end" engineering is vital. No astuteness in subsequent work can overcome poor front-end decisions, so this stage of design should be supervised by the wisest and most experienced engineers in the firm. One pump expert has stated that 95% of all

pumping stations contain significant design blunders and that these mistakes occur in every conceivable category, including hydraulics, the mechanical system, the electrical system, and common sense. Blunders are expensive in construction, operation, or maintenance. At best, they occasion minor annoyance and embarrassment. Make yours one of the elite 5% with sound planning, forthright conferences with clients and their operators, periodic design reviews and design checklists, and thoughtfulness and the application of common sense to reduce errors and omissions to an acceptable level.

Some decisions can only be reached after cost comparison studies are made (see Example 29-1), but nearly all decisions should be made with the coordinated help of supporting professionals, whether in-house or carefully selected outside consultants, in the following disciplines and subdisciplines:

- Civil engineering
  - Surveying
  - Soils (or geotechnical) engineering
  - Hydraulics (including transient analysis)
  - Structural engineering
- Mechanical engineering
  - Heating, ventilating, and/or air conditioning
  - Noise
  - Vibration
  - Odor control
  - Pumps and piping
  - Engines
- Electrical engineering
- Instrumentation and control
- Architecture

The project leader must be able to communicate with the professionals in these disciplines, and to do so effectively requires familiarity with the language, symbols, and (to some degree) the problems involved with each discipline.

After reaching the major decisions that affect the basic design criteria, the requirements of each system must be clearly conveyed to each designer. The duty of the project leader is to coordinate information transfer between support disciplines to ensure an efficient design and an economical pumping station devoid of blunders caused by the interference of one discipline with another. The importance of keeping a complete and legible set of records in, for example, a three-ring binder cannot be overemphasized. The records should include memoranda of important conferences and telephone calls, design memoranda to individuals in support disciplines, design calculations, and sketches. All records should be indexed, and they should be self-explanatory and easily understood because (1) accidents and opportunity frequently change the complement of an office, (2) design commenced by one engineer may have to be completed by another, and (3) a lawsuit may hinge on adequate records analyzed and interpreted by someone other than either author.

Augment this chapter with Sections 6-8, 15-1, 16-1, the first third of Chapter 17, the introduction to Chapter 22, and (as all of them apply to preliminary design) Chapters 24, 26, and 27. Codes and standards are identified only by their abbreviations.

25-1. Need for Pumping Stations

Pumping stations are expensive to construct, maintain, and operate. They should be avoided whenever practical. Consider the following factors when deciding whether to install a pumping station:

- Topography, excavation, elevations, and capacity of existing water distribution and treatment systems or wastewater collection and treatment systems
- Capital, operation, and maintenance costs along with the possibility that additional skilled personnel may be needed
- Problems such as odor or noise and other adverse aesthetic effects.

In some circumstances, odor control may represent the major cost of operation. In 1981 regulatory agencies forced the owner of a 1.1 m^3/s (25 Mgal/d) wastewater pumping station in California to install an odor-control scrubbing system that cost \$200 to \$300/d for operation. Because there is no scientific way to evaluate odors, the owner and the engineer may be at the mercy of hostile residents.

Possible alternatives to installing a pumping station include the following:

- Increasing the head at an existing water pumping station by changing impellers, pumps, or drivers
- A reservoir at a critical elevation that could be filled when water demands (and transmission losses) are low
- A deep, gravity-flow interceptor sewer, perhaps installed in a tunnel
- Individual, on-site, or community underground wastewater disposal systems at a lower elevation, especially when the area served is small.

In some situations it may be desirable to construct a "bare-bones" design for a short service life pending the eventual construction of a long-term (and more costly) conveyance facility.

An economic analysis of alternatives should include:

- Capital cost
- Annual operation and maintenance costs
- Spare parts inventory
- Service life and replacement cost recovery
- Power costs, including heating and ventilating
- Annual lubrication and parts such as seals, bearings, packing, and gaskets
- Capital and annual costs for odor-control facilities and the additional treatment and odor-control costs related to the septicity of wastewater detained several hours in force mains.

Equation 29-6 is useful for evaluating the economics of alternatives, and an engineering comparison of alternative designs is presented in Example 29-1.

25-2. Site Selection

Designers may have little choice in selecting the sites for wastewater pumping stations, which must often be located in low-lying areas characterized by poor soils and high groundwater. The deep excavations for wet wells create difficult problems requiring expertise and care. The designer of a water pumping station has a better choice of sites. The location of sites for raw water intake pumping stations usually depends on the requirements of the intake structure rather than the pumping station. Sites for lake or reservoir intakes are governed by such factors as shoreline topography, unstable soils or rock, depth of water and need for multiple port inlets, aquatic growth, fish protection, surface trash, and ice. For rivers, the factors also include the scour or deposition of bed or bank material, the distance from unstable channel reaches, the adverse effect of nearby bridge piers or other hydraulic structures, the stability of river banks, the location of river bends, the character of the thalweg, and the hazards from floods. Attempts to improve poor river intakes by adding groins, training walls, or other works is not only expensive but often ineffective. Even the U.S. Army Corps of Engineers has experienced unsuccessful attempts to make a poor site satisfactory.

Factors to Consider for Site Selection

The following factors should be considered when selecting a site.

- *Land availability and cost*. Allow for construction activities, future expansion, and parking for maintenance vehicles and even mobile cranes. Except in wet, unstable soils or very deep pits, it is less expensive to excavate side slopes than to use sheet piling.
- *Topography*. Land should be flat enough to minimize construction problems but have enough slope for surface drainage.
- *Soils*. Complete the subsurface survey prior to land acquisition.
- *Protection from flooding*. Some states require designs based on the "hundred-year" flood. Flood elevations may be obtained from (1) the U.S. Geological Survey, (2) the Soil Conservation Service, (3) the U.S. Army Corps of Engineers, (4) the Department of Housing and Urban Development Flood Insurance Studies, (5) state agencies, or (6) by observing the high-water marks pointed out by the owner or local residents. Note that changes in upstream or downstream developments may alter future flood levels.
- *Availability of utilities:*
  - *Water and fire protection*. Field check the available pressure and obtain the local fire department requirements for minimum flow and pressure.
  - *Power*. Check the available voltage and capacity. If electricity from a second, separate substation is used for standby power, the owner may have to pay for installation. Find the costs of extending water and other utilities to the site.
  - *Gas*. Heating and standby power may be operated with natural gas or with biogas from digesters.
- *Access*. Roads adequate for required construction and future maintenance are necessary.
- *Security*. Consider the potential for theft and vandalism. Avoid remote sites that are visible and readily accessible from a road.
- *Aesthetics*. Highly visible sites or sites near other structures may require special architectural treatment and/or provisions to minimize odors and/or noise.
- *Multiple use*. Obtain owner input on site use and combining other facilities with the pumping station.
- *Local land use and zoning ordinances*. Are there any special restrictions imposed by planning agencies?
- *Transmission pipelines*. Minimize profile elevation changes and the length of pipelines. Consider the problems and costs of rights of way, eminent domain, and construction in busy neighborhoods.

Subsurface Investigations

It is vital that a qualified geotechnical engineer make subsurface (soil) surveys and prepare an engineering

report. The geotechnical engineer must inform the structural engineer about expected soil conditions and describe their probable effect on the various options available for the structural design engineer. It is the structural engineer's responsibility to evaluate the geotechnical engineer's report and select the most suitable option. It is often advantageous to involve the soils engineer during the construction phase to ensure that construction procedures conform to design assumptions regarding the magnitude of earth pressures and the sequence of construction. A sufficient number of borings should be made to determine the stability of the excavation. The following information is required:

- The stability of slopes.
- The possibility of heave in the bottom of the excavation.
- The high and low groundwater levels.
- The best dewatering procedure: (1) pumping from open sumps, (2) well points, (3) deep wells, (4) ejectors, (5) freezing, or (6) sheeting driven to an impervious stratum and caulked to prevent leakage.
- The need for permanent underfloor drainage.
- The best method of excavation: (1) open pit (include the allowable slope), (2) sheeting (steel sheet piling or soldier beams and lagging), (3) the need for bracing (either cross-lot bracing or prestressed tie-backs), (4) freezing the embankment (to eliminate the need for sheeting and bracing), (5) coffer dams, or (6) caissons.
- The best method for resisting uplift due to groundwater: (1) weight, (2) tension (uplift) piles, (3) prestressed drilled anchors, or (4) a coffer dam driven to impervious substratum with a permanent sump pump as backup for any slight leakage.
- The lateral earth pressure (1) on rigid walls with passive earth pressure or (2) on flexible walls for both moist soil and saturated soil.
- Soil properties at all changes in strata: (1) natural moisture content, (2) soil unit weight, (3) Atterberg limits for cohesive soils, (4) unconfined compression tests in cohesive soils, (5) standard penetration tests in sand, (6) cores in rock, (7) coefficient of friction between the bottom of the slab and soil, and (8) shear strength.
- The probable effect of excavation, dewatering, and/or pile driving on nearby structures.
- Any unusual conditions such as variable rock formations and elevations, springs, quick conditions, the sinkhole potential in deposits over limestone, the potential for "boil" in the bottom of the excavation, and perched water table.
- Feasible foundations: (1) base slab on grade, (2) base slab on piles with treated or untreated wood piles, steel H piles, concrete-filled steel pipe piles, concrete-filled and mandrel-driven tube piles, precast concrete piles, or mandrel-drilled, concrete-filled, step-tapered piles, or (3) base slab on drilled piers.
- If piles are required, the types, required depth of embedment, capacity, whether batter piles are needed, and whether pile load tests are required.
- If drilled piers are feasible, the elevation of the bottom, whether the drill shaft needs to be lined, and whether the piers should be belled.
- If a base slab on grade is the solution, (1) allowable bearing pressure, (2) probable settlement, and (3) coefficient of friction between the base slab and soil.
- The possibility of unequal lateral forces and the resultant tendency for sliding if adjacent areas are excavated in the future.
- The water table: Install a permanent well point in one boring to determine the static water levels because levels obtained when borings are made are unreliable.

The foregoing information is the responsibility of the geotechnical engineer. The investigation is made with the help of preliminary design information furnished by the structural engineer.

Hydrogen Sulfide Investigation

Hydrogen sulfide (H_2S), a product of stale wastewater, is a deadly, odiferous (it has a rotten-egg smell) gas slightly heavier than air. Certain bacteria convert H_2S to sulfuric acid, which is very corrosive to electrical equipment and to concrete, iron, and steel. Large wet wells exacerbate the H_2S problem because the quiescent conditions allow solids to settle and, because they are not readily resuspended, the long retention time increases H_2S production. Always investigate the concentration of H_2S at the beginning of the project because a substantial (more than, say, 1 ppm concentration of H_2S or a combination of high ($> 22°C$ or $< 72°F$) wastewater temperature and long (> 5 h) residence time in the sewers, may:

- be a deciding factor in choosing variable-speed drives;
- require a sealed wet well or pump sump;
- alter the location of the pumping station site;
- require odor-control units on the ventilation exhaust system to meet state requirements (In

California, the concentration of H_2S in air at the property line is limited to 30 ppb.);

- increase the capital and operation costs of the ventilating system;
- influence construction details, such as the kind of cement used or the type and lining of the wet well and sewers; and
- require a specific regimen of wash-down operations to keep acid-producing bacteria in check.

25-3. Architectural and Environmental Considerations

Before the pumping station design is begun, make sure all the environmental considerations are met and public hearings have been held. The site should be checked for archaeological findings, and local and state regulatory agencies that may have jurisdiction should be consulted. Become acquainted with the pertinent codes. Local planning agencies are likely to require review and approval of plans. Discussions with the owner, the local residents, and the planning agency are useful in determining the style of building to be used. Consider using a consulting architect if one is not available "in-house." In general, the cost of architectural treatment is minor; one expert estimated the cost to be less than 2% of the construction cost. Others maintain that it is often difficult to determine whether architectural treatment costs anything.

Aesthetics

For the most part, utility installations should not be seen, heard, or detected in any way that would degrade the local environment. To that end, the design should include the following features:

- Architectural design that blends into the neighborhood and neither establishes a presence that declares the purpose of the installation nor degrades the property value of its neighbors. Pumping station superstructures, with minimal effort and expense, can be easily crafted to appear to be homes, farmhouses, or restaurants. See Figure 25-1 for alternative architectural concepts.
- As alternatives, pumping stations can either be concealed by burial (with the exception of electrical and instrumentation equipment, which must be protected from flooding), partly concealed by landscaping, or made into attractive architectural features. Landscaping plans that minimize maintenance should be developed by a landscape architect.
- Noise and light emissions should be suppressed to avoid broadcasting the function of the installation. Special construction features can be included to virtually eliminate the high-frequency noise from motors. Hospital-grade silencers are available for engines at little additional cost. With diligence, the noise from ventilation equipment can be suppressed. Lighting fixtures in a variety of architecturally acceptable designs are available to illuminate exterior areas without suggesting a commercial or industrial facility.
- For wastewater stations, odor control and treatment are perhaps the most challenging environmental management tasks. The design should incorporate every technique for avoiding the conditions that generate odors. Many cost-effective and simple treatment techniques are available. The alternative is to design a cheap structure that will invariably look like an ugly pumping station.

Hazardous Areas

An unventilated or sealed wastewater pumping station wet well is rated as a *Class A Confined Space*. Because of the extreme danger of explosion, flammability, toxic gases, and oxygen deficiency, it can be entered only under stringent regulations (see Section 23-1). If the wet well contains electrical equipment, it is rated by NEC standards as a Class I, Division 1 hazardous space when flammable or explosive gases are normally present, as in an unventilated space. However, if flammable gases are absent except under abnormal conditions (e.g., failure of ventilation system), the wet well is rated as a Class I, Division 2 space. All electrical equipment and devices in Class I, Division 1 spaces must be explosion-proof, whereas in Class I, Division 2 spaces, only arc-producing equipment (e.g., switches with external contacts) must be explosion-proof. (A squirrel-cage motor in a Class I, Division 1 space must be in an explosion-proof enclosure, whereas it can be in an open, drip-proof enclosure in a Class I, Division 2 space as long as it is not equipped with switches.) Division 1 spaces in wet wells can be avoided (except below the grating) by installing a ventilation system that always operates at 12 air changes per hour or more (see Section 23-2). The ratings of the various areas must be decided in conjunction with the authority having jurisdiction—the local or state electrical inspector.

A dry well with access to a wet well also becomes a Class I, Division 1 or Division 2 hazardous space, and such access should never be tolerated in new

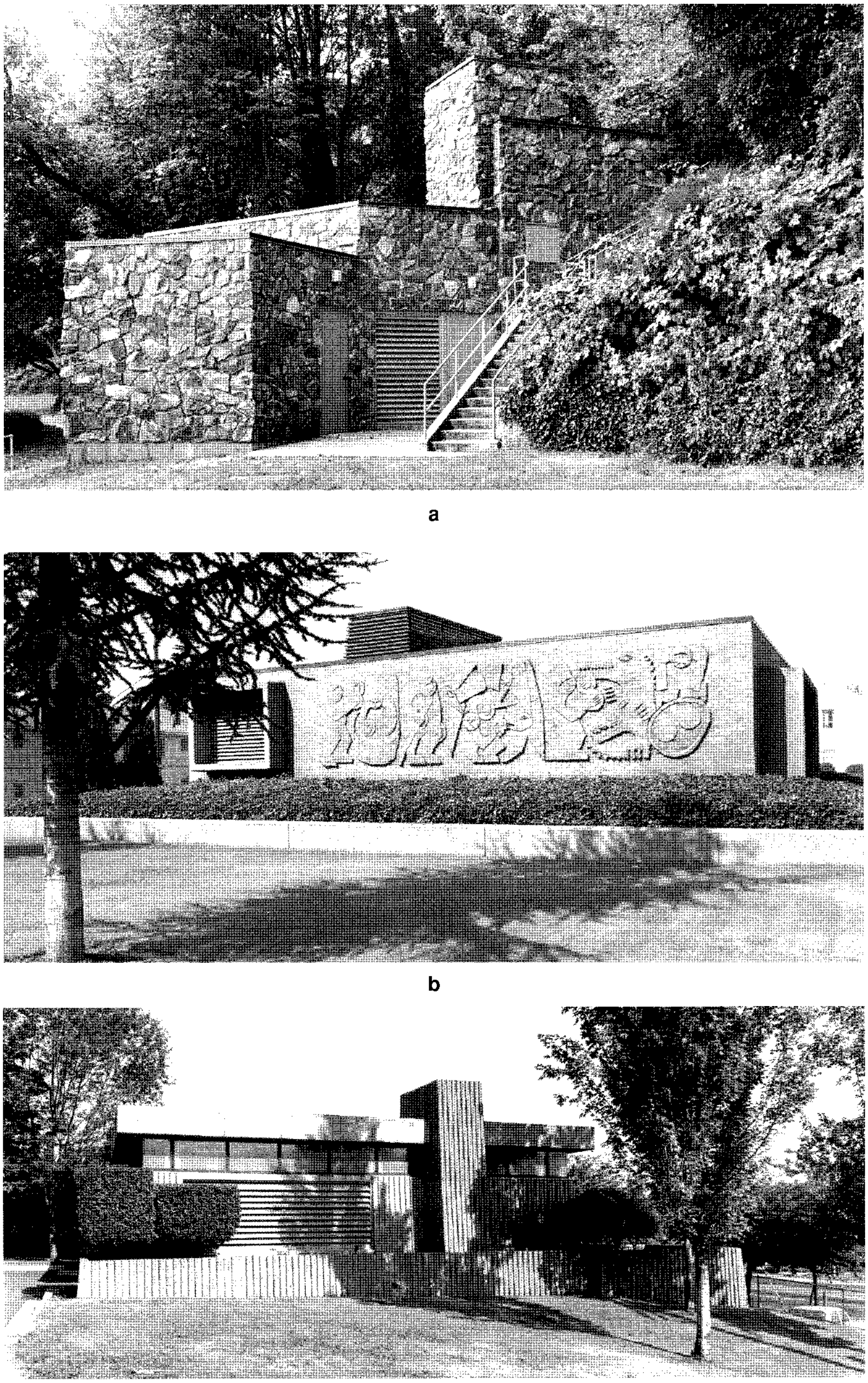


Figure 25-1. Seattle Metro wastewater pumping stations designed by Metropolitan Engineers (a consortium of Brown and Caldwell; R. W. Beck; Kramer, Chin and Mayo; and URS). (a) Trees and shrubbery conceal the station from the road above; (b) architectural enhancement of a pumping station; (c) a pumping station in a residential district. Photography by George Tchobanoglous.

construction nor should it be allowed in an existing pumping station being remodeled. Access to a wet well should be possible only through an outside door.

Fire Safety in Pumping Stations

Several codes (NFPA 820, NFPA 70, NFPA 101, to name a few) provide guidance for fire protection for installations such as pumping stations. In addition to these documents, the designer should consider the following:

Choice of Materials

- Be careful in the choice of materials that could add to the fuel load (the amount of burnable material present) in a structure. Plastic, although useful for corrosion protection and resistance, will burn if ignition occurs. The products of combustion from plastics are often toxic and corrosive, and become more so when they contain fire retardants.
- Be suspicious of materials that could auto-combust, such as some forms of activated carbon. If these materials are used, make certain that they are isolated from other combustible materials.

Fire Suppression Systems

- Storage areas, such as those for records, lubricants, and fuels, should be separated from other areas of the structure by two-hour rated fire walls and should be served by independent ventilation systems. Appropriate fire suppression systems should be considered for these areas.
- Cable trays and other locations where electrical conductors are not protected by conduit should be monitored by rate of temperature rise, particle ion, and smoke detection systems.
- In large installations, consider dividing the electrical systems into identical halves and placing the switchgear in separate, fire-rated rooms to avoid the prospect that an electrical fire will disable the entire facility.
- In critical facilities, install sprinklers over cable trays.

Ventilation

Sewer gas is deadly. The safety of personnel who must enter pumping stations should not be compromised, especially if they must enter wet wells. Codes require a certain number of air changes per hour, and minimums are shown in Table 25-1. However, some states require a higher rate of ventilation and, if the wastewater is stale, so should prudent engineers.

Some codes allow a lower rate of ventilation while the station is unattended if two-speed fans ventilate a hazardous area at high rate before a worker enters. The use of lower ventilation rates should be discouraged because they do not guard against explosions, corrosion, or the impatience of workers. Air to wet wells should be *both supplied and exhausted* through ducts by powered blowers so that a slight negative pressure is maintained (refer to Sections 23-1 and 23-2).

In stations housing large motors (whether for pumping wastewater or water), the rate of ventilation required for cooling the motors may be a controlling factor, especially in hot climates where the ventilation rates can be surprisingly high. Heat dissipation in stations equipped with engine drives can be of even greater magnitude. The ventilation problem is one to be solved by an experienced heating and ventilating engineer.

Requirements for the design of ventilation systems for wastewater installations are provided in NFPA 820. Designers should be particularly aware that control and electrical equipment require an environment that has proper temperature control and protection against atmospheric conditions such as hydrogen sul-

Table 25-1. Suggested Minimum Ventilation Criteria for Wastewater Pumping Stations

Space	Ventilation criteria
Wet wells with access for maintenance	For continuous operation, 12 to 30 air changes per hour. Supply air at the ceiling. Withdraw air at the floor near wastewater channels. Use powered blowers on both intake and exhaust.
Motor rooms	Sufficient air to control the buildup of heat.
Engine rooms	Sufficient air to control the buildup of heat of the air supplied. Exhaust at least 10% at the roof to control buildup of smoke and fumes.
Pump rooms	Use 10 air changes per hour. Supply air at the ceiling; withdraw air at floor.
Motor control centers	Provide 6 air changes per hour of filtered air, and air condition if heat would otherwise be a problem.
Dry wells	For continuous operation, 6 air changes per hour.

fide gas. Because overall station reliability greatly depends on the proper operation of these devices, environmental control is of vital importance to the success of the installation. See Chapter 23 for information on proper design for purging and scavenging toxic, dangerous, and corrosive gases from an enclosure.

Odors

Unpleasant odors are emitted from some wastewater pumping stations—a serious problem for stations in or near a residential area. Bad odors are often caused by grease and scum buildup on the wet well walls, so good housekeeping practices may eliminate the nuisance. Steps can often be taken upstream from the station to minimize hydrogen sulfide and reduce odor problems. By sealing the wet well, odors can be largely confined, particularly if the drivers are variable-speed (V/S) units that eliminate the "pumping" action produced by the rise and fall of the wet well levels that occurs with constant-speed (C/S) pumping units. If it is impossible or impractical to eliminate odors, then air-scrubbing units may be required in the air exhaust system—an expensive solution. If freezing can occur, dry scrubbing units may require thermostatically controlled heaters to avoid plugging by frozen condensate.

Access

Provide safe, adequate access for personnel and adequate space and facilities for replacing any or all machinery. The considerations include the following:

- Pump rooms (dry wells) and wet wells must be completely separated. Therefore, provide separate ground-level access to each. *Never* interconnect wet and dry sides even above the flood level.
- Entrance depths of 3.6 m (12 ft) or more should have stair landings or ladder platforms.
- Ladders should have nonslip, serrated, or splined-edged rungs of aluminum and should be surrounded with a cage. Aluminum can corrode at the concrete interface, so coat the aluminum at the point of contact or consider plastic-coated steel or stainless steel.
- Conventional stairways and landings are by far the best. It is difficult to carry tools and nearly impossible to carry injured workers up ladders or circular stairways.
- Hatches, doors, and other openings must be large enough to remove the largest piece of equipment.
- Large access covers are difficult to lift, so specify standard spring-loaded covers made skidproof by using checkerplate or an equivalent.
- Individual hatches over each submersible pump or a duplex split hatch over two pumps is preferable to a single large lid over two or three pumps.

Storage

In larger pumping stations, provide space for the storage of spare parts inventory, tools, and maintenance equipment.

Lighting

Uniform lighting of about 200 to 300 lux (20 to 30 ft · cd) at the floor is adequate except for repairs, which require about twice as much light. Install additional ceiling lights that can be switched on when needed (required in some states for energy conservation). An alternative is GFCI outlet receptacles for portable lights. Outlet receptacles are also required for small power tools. Supply 120-V power for lights, outlets, and small loads from a small, dry-type transformer (25 kVA or less) supplying a single panelboard.

Personnel

Personnel requirements for pumping stations vary over the broad range given in Table 25-2. Some owners allow major pumping stations to be unmanned, while others may require 24-hour, two-person operation for a similar facility. Factors to be considered for operational personnel requirements include: (1) the capital investment, (2) the degree of reliability needed, (3) security and the risk of vandalism, (4) safety, (5) union requirements, and (6) the owner's policy. The extent of automation and remote telemetry required, as well as many of the amenities of the station, depend on whether it is to be manned or unmanned.

Scheduled daily inspection, including maintaining a daily log, is normally allocated to unmanned pumping plants. The facilities should be adequate for regular scheduled maintenance as well as for corrective and emergency unscheduled maintenance, which may require working crews of several persons for several days. To avoid problems later, discuss the station layout with the owner's maintenance personnel before beginning the design.

Table 25-2. Operational Requirements of Pumping Stations

Type of facility	Operational personnel	Comments
Water pumping stations of less than 750 kW (1000 hp); wastewater lift stations with flow rates less than 0.2 m^3/s (5 Mgal/d)	Unattended	Daily inspection, automatic controls, alarm and fail-safe features
Major water and wastewater stations with engine drives, screens, chlorination facilities, or odor-control facilities.	Staffed, 8 h/d, 5–7 d/week	Automatic controls, local and remote alarms, fail-safe features
Major regional water and wastewater stations for communities of more than 250,000	Staffed, 8–24 h/d, 7 d/week	Automatic controls, local and remote monitoring

Convenience

Items providing for convenient operation and maintenance include the following:

1. The building:

- Materials on the walls, floors, and ceilings easy to maintain with no rough finishes.
- Coves at the base of all walls.
- Good floor drainage (slope the floor at ≥ 1% to the drains so that water cannot form puddles).
- Easy access to all windows, louvers, and equipment to be serviced.
- Sufficient space to service and maintain equipment—a minimum of 1.1 m (42 in.) clear around machinery.
- Loading platform and ramps.
- Restrooms, a slop sink, and a janitor's closet for all but the smallest stations.
- Room for storing miscellaneous small tools and maintenance items such as impellers, wearing rings, packing, and other materials as specified by the owner or manufacturer.
- Workbench, tools, and log books (for all but the smallest stations).

2. Access:

- Convenient roads.
- Convenient hatches for equipment removal.
- Pipe basements instead of crawl spaces.

3. Equipment:

- Cranes or eyes for handling the debris from screening equipment.
- Drainage to sump pumps for dewatering and washdown.
- Eyes, monorails, or cranes for equipment handling.

4. Miscellaneous:

- Hose valves (bibbs) with a backflow preventer for cleaning and for yard irrigation (freeze-proof them if necessary); check state regulations for the separation of potable water from wash water.
- A desk, chairs, and file cabinets for storing records, maintenance manuals, and plans.
- An O&M manual for reference and troubleshooting.
- Special tools.
- Site maintenance equipment and storage rooms.
- A drinking water fountain or bottled water and drinking cups.

Safety

Safety considerations include the following:

- Nonslip walkways of checker or diamond plate or covered with nonskid "sanded" paints
- Handrails and kick plates around openings and stairways
- Safety cages on ladders and intermediate platforms for ladders longer than 3.6 m (12 ft); straight stairways are far more convenient and safe than are ladders
- Air purging systems
- Chlorine detectors and gas masks
- Two sets of self-contained air-breathing tank and mask apparatus
- Explosive air detectors
- Hydrogen sulfide detectors
- Eye washers and showers if hazardous chemicals are used
- Personnel equipment such as hard hats, safety harnesses, first-aid equipment, ropes, and life vests or rings

- Fire extinguishers
- Telephone or two-way radio communications—unnecessary if service trucks are always so equipped
- Grounding of all electrical equipment including GFCI receptacles for hand tools
- Backflow (cross-connection) preventers or an air gap between the potable water supply and the pumping station water supply systems
- No open-style personnel lifts; all lifts and elevators to have solid walls, ventilation, roof escape hatch, telephone, and no gates.

Security

Consider the following as security items:

- Fencing and exterior lighting
- Unauthorized entry alarms with signal transmission to some authority (e.g., the police).

25-4. Future Expansion

Designers must have the vision and presence of mind to plan for expansion beyond the design life. If a future demand for increased flow is likely (nearly always true), consider provisions for future expansion and weigh the cost of such provisions against the future cost of remodeling or abandonment. The size and importance of the pumping station, the owner's requirements, and the uncertainties of need, time lapse, and expected flow increase must be taken as intangible but essential and sometimes overriding considerations.

The method for achieving the future increase in capacity should be included in the specifications and the plans. Future expansion can be facilitated in a number of ways that include:

- Pumps with medium-sized impellers that can be changed for larger ones.
- Generous space for each pump so that a larger pump can be installed. The extra space will be appreciated by the maintenance staff.
- Suction and discharge piping large enough to permit the increased flow.
- Space for additional pumps. Terminate suction piping and pipe connections to the header with valves and blind flanges.
- Provision for enlarging the building by adding steel waterstops and dowels for connecting future walls and floors; bend the dowels to the face of the concrete and protect exposed steel with lean mortar (note that rubber or plastic waterstops can be too easily damaged by future construction).
- An electrical system designed so that the entire system will not have to be rebuilt; use larger conduit, for example, and leave extra space where needed.

Plan the operations necessary to allow the changes to be made while the station remains in operation. Consider the following:

- The pump manufacturer should know whether larger impellers are to be used so that bearing frames, bearings, motor, and other associated parts are sized properly.
- The removal of blind flanges on the discharge side of a future pump requires either (1) a means for quickly draining pressurized pipelines, or (2) a valve at the flange. Note that valves are likely to become inoperative unless regularly exercised. Gate valves are the worst, lubricated plug valves are good (and the most expensive), and stop plates or slide gates are the best, the most reliable, and the least expensive when they can be used for isolating pump inlets.
- The removal of blind flanges on suction pipes requires dewatering the wet well (or a portion thereof) in a way that minimizes (or eliminates) downtime or prevents overflows in wastewater pumping stations. The wet well may be compartmentalized by inserting stop plates into grooves provided for that purpose.
- Considerable storage is available in sewers upstream from most wastewater pumping stations—storage that is sufficient to make the needed connections if planned properly and executed at the lowest flow.
- Because blind flanges are heavy, require lifting handles to speed removal.
- Ensure convenience for the lifting equipment needed to install new or replacement facilities, and provide access for the installation of new equipment.

The capacity of an existing pumping station can often be increased. To avoid blunders, first make field measurements of flow, static head, friction head, true pipe diameter, and roughness coefficient. Trust no plans—not even record drawings. Measure the station, especially the piping connections, so that the new facilities will fit exactly. Various methods of increasing the capacity are:

- Cleaning and relining transmission pipelines in place
- Laying new transmission lines or force mains

- Substituting larger impellers in existing pumps
- Substituting larger pumps and motors if the suction and discharge piping permits
- Adding submersible pumps in the existing wet well
- Converting the dry well to a wet well and adding submersible or VTSH-type pumps
- Adding a new wet well with submersible pumps
- Constructing a new pumping station
- Adding a booster pumping station.

25-5. Hydraulic Constraints

The hydraulic constraints include the flow rates (and the range of flow rates); the discharge characteristics required of the pumps (the range of both the flow rate and the head); the suction head; and the type of fluid pumped.

Wastewater Pumping

Pumping stations should be designed to handle flows from the entire service area. Obtain owner input regarding the size of the service area. Install a pumping capacity that is adequate for the fully developed service area, or install a pumping and force main capacity that can handle both the current needs and those of the immediate future with an adequate provision for economically adding to the capacity in the future. Consider both average and peak flows, which may be based on (1) existing and future land use, (2) per capita flow estimates, or (3) actual field flow rate measurements. Consult the literature before making design decisions. See Davis and Sorenson [1] and ASCE Manual 37 [2]. When making flow rate projections,

- Choose a realistic design period, but make it possible to meet flow demands beyond the design period.
- Design the station layout to allow for planned increases in the capacity at specific times within the design period.
- Always use the larger pump size when choosing between two sizes.
- Optimize the size of force mains. A good rule is $0.61 \leq$ velocity ≤ 2.4 m/s ($2 \leq$ velocity ≤ 8 ft/s). Velocities as low as 0.48 m/s (1.6 ft/s) are tolerable with two daily flushes.
- If velocities are less than 0.76 m/s (2.5 ft/s), a daily flush at 1.2 m/s (4.0 ft/s) long enough to sweep out the entire volume of the force main is desirable. The maximum velocity should not exceed about 2.4 m/s (8 ft/s) because of high headlosses and the possibility of water hammer. If these constraints are mutually exclusive, use two force mains and block either as needed for achieving wanted velocities.

Screens

Only a small percentage of pumping stations require screens. At many stations, manual screens have been abandoned because they must be raked daily, and operators prefer to clear an occasionally blocked pump than to tolerate onerous daily chores. Mechanical screens require a series of ancillary facilities for conveying the screenings, washing and then dewatering them, temporarily storing them in bins, and trucking them to a landfill. Odor control is often necessary. Maintenance is labor-intensive. If, nevertheless, screens must be added, pay heed to the many references to them in Chapter 24. See Pankratz [3] for more information.

Water Pumping

For the probable population to be served and the estimated flow needs, choose a realistic design period and allow for future expansion beyond the design period.

- The station layout should permit plans for periodic increases in flow capacity.
- Carefully consider the maximum daily and hourly flow rates, the requirements for fire flow, and a combination of the two.
- If head and flow are expected to change significantly over the years, choose pumps with medium-sized impellers so that impellers only—not pumps—need to be changed to match the flow.
- Choose a range of roughnesses and not a single, "conservative" value (see Section 3-2).
- Optimize the size of transmission mains. Where velocities exceed about 2.4 m/s (8 ft/s), water hammer may become a concern and may require additional, expensive control equipment and structures.

Sludge Pumping

Sludge is usually confined to treatment plants and transported for only short distances—a few hundred meters or yards at most. Centrifugal pumps can be used for thin sludges in large quantities. However, positive-displacement pumps are usually required for thick sludges because these sludges are thixotropic

and, after immobilization, high pressures are required to start their movement. For wastewater sludges, consider:

- Sludge properties, which are entirely different from those of water (see Chapter 19).
- Sludge pipelines longer than 1.6 km (1 mi) must be designed on the basis of *measured* friction. (The associated problems are difficult and beyond the scope of this book. See Section 19-5.)
- Comparisons of plunger, progressing cavity, and lobe pumps based on both first cost and maintenance.
- Pipelines of no less than 150 mm (6 in.) in diameter, although glass-lined piping down to 100 mm (4 in.) may be suitable.
- Glass lining for sizes below 250 mm (10 in.) and cement-mortar lining for any size (but particularly for larger pipe).

Water plant sludges are of two general types: (1) those resulting from sedimentation and flocculation, and (2) those from lime softening. The former are gelatinous, bulky, and difficult to dewater. It may be desirable (although difficult) to decant them, but they are not difficult to pump. The characteristics of softening sludges vary greatly, and it is difficult to predict a conservative design that operates well over the range of necessary conditions. It is wise to:

- Choose pumps that can be easily and relatively quickly disassembled and cleaned and locate them for ready accessibility;
- Use straight pipelines that can be rodded out or, better, use open troughs that are easily accessible for cleaning;
- For a more extensive discussion of sludge pumping, refer to Chapter 19.

25-6. Types of Pumping Stations

The following discussion about the types of pumping stations is necessarily abridged. Because greater detail is given in Chapters 11–15, 17–19, 26, and 29, the discussion in this section is limited to a few selected essentials relating to choice. Develop alternatives only to the point where realistic, comparative cost estimates can be made (e.g., as shown in Example 29-1).

Wastewater Pumping Stations

Decisions should be based on the required capacity, cost, reliability, the owner's preference, and aesthetic considerations. Reliability is more important than efficiency. Wastewater pumps must pass stringy materials and other solids, and it is frequently specified that they must be able to pass a 75-mm (3-in.) sphere. But because many relatively large pumps contain restrictions, always require certification on the size of solids to be passed *entirely through* the pump.

Before choosing the type of pumps and pumping stations, consider the following:

- Minimum 100-mm (4-in.) piping for wastewater may dictate the pump size required to maintain minimum velocities. Many authorities and some codes allow 75-mm (3-in.) piping, but if this small pipe is used, make certain the pumps (or screens) cannot pass solids 75 mm (3 in.) in diameter.
- Provide enough headroom, floor space, and access to replace the pumps, motors, valves, and piping.
- Single-volute pumps and single-vane impellers are asymmetrical and cannot be hydraulically balanced. Many manufacturers cannot trim single-vane impellers. If a single-vane impeller is trimmed, it should be balanced dynamically (often by using attached counterweights) after trimming.
- A V/S drive allows substitution of a small pump sump for the large wet well, but V/S pumping increases cost, complexity, and maintenance, and it might even decrease efficiency. Many V/S installations seem to have been misapplied, so read Sections 12-5 and 15-1 and Example 29-1 before deciding to use V/S.
- For flat system head curves, multiple C/S pumps alternated "first on—first off" may have most of the advantages of combined V/S and C/S pumping.
- Provide one spare standby unit of the largest capacity and consider the need for other spares, depending on the critical effects of station outage.
- Supply standby power. (See Sections 8-4 and 26-9 for engine-generator requirements and Example 9-10 for engine-generator sizing. Engine requirements are discussed in Chapter 14. Note that a separate electrical utility service can also be used for standby.)

Submersible Pumps

Large (75- to 750-kW or 100- to 1000-hp) submersible pumps are gaining market acceptance, although a number of utilities seem to agree that (due to unfortunate experiences and possibly inadequate facilities) they should be installed only in dry pits. In one large utility, no submersible pumps larger than 75 kW (100 hp) are allowed in wet wells. In another large utility, it required a dozen people working nearly a

whole morning to remove a large submersible pump from a storm water pumping station because it was thought necessary to disconnect the power cable from the motor before pulling the pump. Hence, two workers were required to enter a wet well—a task that is not recommended by any pump manufacturer. As the wet well is a confined space, the large crew was necessary for safety.

Contrast the above scenario with the experience of still another large utility that likes large submersible pumps. The pump maintenance supervisor at that utility stated that 190-kW (250-hp) submersible pumps can be lifted and removed in half an hour by a crew of three (boom operator, electrician, and mechanic) using a truck-mounted boom. Larger pumps require a mobile crane, but the job is otherwise just as easy. (Workers never enter the wet well except to work on guide rails or brackets.) The pumps are cradled on their sides for access to impellers. The same utility has mechanics who are certified for repairing submersible pumps and who can even balance impellers to tolerances more precise than factory specifications. Minor servicing (such as changing the oil and checking the shaft and bearings for play) can be performed while a pump is in the cradle. When a pump is to be sent to the shop for major repairs, such as replacement of seals and bearings, the power cable is disconnected at the motor starter and not at the motor. The cable is pulled out of the conduit and accompanies the pump. This procedure allows the connection to the motor to be tested for shorts. After repairs and before approval for returning the pump to the wet well, the motor is run for a final check. Thus, large submersible pumps in wet wells *can* provide good service when the facilities are specifically tailored for their use and maintenance crews are well-trained and motivated. Handling these large pumps is, however, entirely different from handling small ones because of their weight and the weight, size, and stiffness of their electrical power cables. Both cable and pump must be maneuvered with the assistance of cranes, whereas the cables for smaller pumps can be handled manually.

Special care must be taken in both the design and handling of large submersible pumps to prevent cables from being cut or rubbed on projections. At least one edge of the hatch should be round and smooth so the pump can be swung inboard without damage to the cable. Cables should not be allowed to swing freely in currents, because the movement may eventually crack the insulation or wires. Pulling the slack out of the cable is usually sufficient, but discuss the problem with the pump manufacturer,

One technique for removing large submersible pumps is to lift the pump while allowing the cable to hang freely in a U-shaped loop like an elevator cable. After the pump is swung inboard and placed on the floor (or in a cradle), the cable can be retrieved with the aid of the crane or boom. If the cable is to be disconnected at the motor cap (cable entry), the disconnected end can be held at floor level with, for example, a Kellems® Support Grip. This method eliminates the need for separate handling of the cable and for special arrangements for storing the cable while the pump is being repaired. On the other hand, disconnecting the cable at the motor starter does not disturb an attachment that must remain absolutely waterproof.

When a pump is pulled up in the above manner, the cable bends in an arc above the motor cap. This bending does not harm short cables for small pumps, but it may overstress long, heavy cables for large pumps. The cable can be protected (1) by means of semiflexible stainless-steel cable sheathing designed to control the radius of the curvature, (2) by the use of cable saddles, or (3) by emerging vertically downward from the motor cap. The need for special protection should be discussed with the pump manufacturer. An alternative method is to lift both pump and cable together, thereby eliminating bending near the motor cap. A motor-driven cable reel mounted close to the wet well is of great help for handling and storing cable. The cable is unwound from the reel when the pump is installed, and a predetermined tension keeps slack out of the cable. When the pump is removed, both it and the cable should be washed with a high-pressure hose for sanitation.

The tension in long, heavy electric power cables for large pumps may require relief. One relieving method is to clamp the cable at intervals to a stainless-steel wire rope that carries the weight of the cable. Another method is to use a Kellems® Support Grip to attach the midpoint of the cable to a stainless-steel wire rope held by a hook device at the hatch of the wet well.

The unprotected end of the power cable is not submersible and should not even be exposed to water. Submersible pump control and power cables should never be spliced. The pump should be ordered with sufficient cable to avoid the need to splice. Water that does enter the ends of a control cable can cause false signals in the control panel and possibly enter the motor itself. Connecting a flexible cable to another in a junction box, however, is acceptable if the junction box is properly located. Water has entered power cables through junction boxes located improperly (for example, recessed into floors or mounted in regions of condensing humidity).

Electric power cables for submersible pumps in trench-type wet wells are particularly vulnerable to

movement, because the pumps (being on the axis of the trench) are located directly in the path of the strong currents from the inlet. The cables will also collect stringy material. Exposure to currents and rags can be a serious problem that deserves careful evaluation by both the designer and the pump manufacturer. If the distance from the top of the motor to HWL is short, a rigid conduit rigidly fastened to the motor and extending to HWL would shield the cable from all currents. The cable emerging from the conduit must, however, be protected from excessive bending.

The above method does not apply to deeply submerged pumps. A large (150-mm or 6-in.) plastic conduit buried in the concrete wall or a stainless-steel conduit attached to the wall and ending near the discharge coupling is a possible solution if the cable can slide easily up and down the conduit. Again, excessive cable flexure at the ends of the conduit must be prevented. Note that the cable would have to be very long and be payed out from a cable reel as the pump is lifted. Note also that this method is not cheap. Another method is to attach the power cable at intervals to sliders or rollers held in a track (like fastening sails to a mast) that is bolted to the wet well wall out of the way of strong currents. These methods have not been tried, however, and there is no guarantee of success, so a full-scale test should be made.

Designers who have any doubts should describe the cable environment to manufacturers of cables and seek their advice. There may be no problem at all, or perhaps simple measures will suffice. Otherwise a summary of options is (1) to employ one of the methods above; (2) to invent some adequate alternative way to protect the cable; (3) to do nothing special and replace cables from time to time; (4) to locate the pumps in a dry pit—expensive in first cost but less costly to maintain, so the life-cycle cost may be reasonable; or (5) to substitute a different type of self-cleaning wet well wherein cables are not subjected to strong currents.

Refer to Tables 25-3 and 25-4 and Sections 24-10 and 27-6 for further discussion.

Comparison of Different Types of Wastewater Pumping Stations

Different types of pumping stations (Table 25-3) are interrelated with different types of pumps (Tables 25-4 and 25-5), so all three tables must be considered together. Be alert, because a comment in Table 25-3 is not repeated in Table 25-4.

Table 25-3. Comparison of Wastewater Pumping Station Types

Advantages	Disadvantages
Dry well (as compared to wet well)	
Easy access for maintenance.	Greater cost due to excavation and building below grade—expensive if groundwater is high, if soils are very poor, or if blasting is required.
Wider range of head and capacity.	Greater risk of outage due to flooding; dry well must be kept dry.
Wider choice of driver arrangements.	Flood-protected motors (in the dry well) are expensive.
Possible to use flood-protected motors.	Long leads to the motor (in the dry well) from the control panel if the motors are frame mounted to the pumps.
Flooded suction improves reliability.	
With long shaft and flood-protected motors at grade, electrical leads are short.	
Wet well pumps with above-grade drivers	
Less excavation (eliminates the dry well).	Difficult to remove for servicing, especially the pumps with separate discharge pipe.
Small superstructure.	Difficult to service in place, wet well must be drained or pump removed for major disassembly to reach many parts.
Above-grade drivers protected from flooding.	High superstructure needed to permit the removal of the pump.
	Submerged bearings are subject to overloads and frequent failures.
	Difficult to keep lubricated and protected from grit.
	Should not be used for raw wastewater applications because of clogging.

Table 25-3. Continued

Advantages	Disadvantages
Wet well with self-priming pumps above grade	
Least construction cost, easiest maintenance.	Reduced reliability due to the need for priming.
Convenient access to pumps above ground.	Distance from the low wet well level to the pumps is limited by NPSH and the priming system to about 8 m (25 ft).
Eliminates dry well.	If operated infrequently, putrefaction with gas production can occur to break prime and/or to make wastewater boil at 60°C (140°F), so an external priming system must then be used.
No flood hazard to motors.	V-belts require proper adjustment and periodic replacement.
Self-priming pumps do not have trimable impellers, so they are usually belt-driven to reach design point.	
Belt drives are easily changed to meet a new condition point.	
Wet well submersible pumps and motors	
No dry well; excavation and concrete reduced.	Valves and headers must be accessible (1) in an adjacent vault, (2) in a small above-grade superstructure, or (3) by exposing the header above grade.
No superstructure required except for engine-generator or cabinet for motor controls.	Pump must be removed and disassembled for inspection and maintenance. Requires a hoist or crane and specially trained mechanics.
No seal water system, no long shafts with steady bearings required.	Hazard of pumps jamming on guide rails or not seating properly.
Reduces the land area needed.	Often more difficult to remove pumps than manufacturers admit.
All of the above reduces construction costs (see Figure 29-9).	Special motors, seals, and moisture monitoring required (but moisture probes are useless for leaks via power cable).
Quick removal and replacement in emergencies.	Makes differ greatly with respect to satisfactory performance, and quality of pump/driver unit has a very high impact on maintenance and life-cycle cost
Well adapted for increasing the capacity of a pumping station using existing wet and dry wells.	Vibration has occurred with some makes of pumps larger than 22 kW (30 hp).
No daily nor weekly maintenance (but overhaul needed every 1 or 2 years).	Guarantees often are valid only if repairs are made by authorized service centers.
Units removable for shop servicing, minimizes field work.	At least one uninstalled spare unit (in addition to standby units) in each size is needed to permit shop servicing, which adds to the cost trade-off for submersibles.
Quiet operation.	Pumps and motors are generally not well-suited mechanically for V/S operation (although AFDs can be used).
Safety from flooding.	Inaccessibility of the pump and motor for routine preventive maintenance and the need for expensive overhaul every 1 to 2 years leads some engineers to refuse to countenance their use.
On a life-cycle basis, balance the lower first cost of submersible pumping station with its lack of regular, frequent maintenance chores against the cost of an annual or biennial complete overhaul by specially trained mechanics (or at manufacturer's service center).	Large units tend to break down more often than do small ones and tend to require high maintenance costs.
Elimination of (1) the dry well, (2) frequent preventive maintenance, and (3) the seal water systems that, coupled with simpler design and low first cost, lead some engineers and many operators to prefer submersible pumps.	Choice of pumps for wet well service should depend almost entirely on the utility itself–attitude, experience, and commitment to train and to support maintenance workers (see Sections 1–13, 24–11, and 27–6).
Submersible pumps and motors in dry wells	
Excellent reliability.	More costly than pump and separate motor, although elimination of seal water system and reduction of wet well size because of more frequent allowable starts may make price competitive.
Maximum protection against dry well flooding.	

(*Continued*)

Table 25-3. Continued

Advantages	Disadvantages
No seal water. No intermediate shafting to disassemble for work on pumps. No steady bearings nor supporting structure. No daily maintenance. No oiling, greasing, or leaking packing glands. Reduced housekeeping. Air fin or water jacket cooling permits much greater pump cycling frequency. Minimal vibration. Quieter than ordinary dry pit pumps. Becoming popular. Conditions favoring use High risk of dry well flooding. Deep stations. Reduced daily maintenance requirement. Above advantages may make submersibles the most cost effective.	Air cooling is inadequate for most machines, so cooling jackets (using pumped fluid) are required. Water jackets may require occasional flushing, and/or disassembly, and they sometimes clog with solids in pumped fluid at low ($\leq$40 Hz) speeds (usually low enough for 0.35 BEC). Fresh water cooling is expensive. Variable-speed control is limited to AFD, which may increase cooling problems especially at lower speeds (see Section 15-11). Repairs in place may void manufacturer's guarantee. If delays are likely for shop repairs, must have uninstalled spare unit. Access to the pump (the part of the unit requiring the most maintenance) is costlier than with a conventional pump.
Immersible motors in dry wells	
Have almost all the advantages of submersible pumps. Pumping units are less expensive than submersibles. Motors are premium grade high efficiency—more efficient than submersible pump motors. Can be submerged 9 m (30 ft) for two weeks. Eliminates need for long drive shafts.	Motor more expensive than TEFC motor. Cannot exceed guaranteed inundation head and time.
Horizontal vs. vertical configuration	
	Horizontal pumps
Some double suction pumps available in large sizes. Easy access for maintenance. Motors less costly. More convenient for belt drive. High headroom not required. Permits a high head design with a double-ended shaft motor and two pumps coupled with series piping.	Requires a dry well. Requires a large floor area. Long leads from the control panel to the motor.
	Direct-connected vertical pumps
Less floor space and a smaller superstructure. See also Table 25-4.	Same as for horizontal pumps (above) plus. Double suction not available. High headroom for lifting motor. Supporting the motor by the pump casing is poor practice in seismic areas.
	Extended shaft vertical pump
The same as direct-connected vertical pumps plus. Avoids flood hazard to motor. Permits work on pump without disturbing motor. Short electrical leads.	Long shaft requires a structure to support the intermediate bearings plus added lubrication and maintenance. Consider a stiff, hollow shaft without intermediate bearings (which may be impractical because limitations are severe). High headroom needed for replacing motors, shafts, and pumps Flexible couplings required at pump and motor. Must ensure that manufacturer analyzes shaft for torsional vibration—especially for variable-speed drivers.

Table 25-3. Continued

Advantages	Disadvantages
Field construction vs. package plants	
Field construction	
No limitations of head or capacity.	Higher construction cost.
Complete flexibility of layout.	More engineering time and skills.
Special features can be included, such as bar screens, comminutors, better access and working room, overhead hoists, and spare parts storage.	More difficult to avoid blunders.
In smaller sizes, may be faster to build than package stations.	
Greater ease of maintenance due to less crowding.	
Package types—general	
Low construction cost (see Figure 29-9).	Alarming number of fatalities and injuries in can-type prefabs due to asphyxiation, H_2S, flooding, and the difficulty of escape.
Consulting engineers' cost is lower for an "off-the-shelf" unit.	Access is often poor and rescue almost impossible.
Equipment can be factory assembled and tested prior to shipment.	Ventilation is usually woefully inadequate.
Standardization reduces chances for blunders.	Layout design is inflexible and special features are limited.
Especially suitable for small stations.	Cramped working space makes maintenance and repairs difficult and more costly because of added labor.
Available in following types: underground wet well–dry well, wet well with suction pumps, submersible pumps in wet well, pneumatic ejector.	Flow rate usually limited to about 110 m^3/h (500 gal/min), although larger ones have been built.
	Corrosion and buckling are potential problems with steel shells, which often corrode quickly (in 2 yr) due to stray electrolytic currents, and hence cathodic protection is required.
	FRP is difficult to use where soil loadings are significant; it is also vulnerable to corrosion if the protective resin coating cracks.
	Discuss hazards and disadvantages with owner.
Package type: wet well–dry well	
The same as the dry well–wet well type above plus it can be less conspicuous with only hatches aboveground.	Inconvenient access.
	Layout design and special features are limited.
	Crowded space adds maintenance difficulty.
	Forced draft ventilation is often inadequate, so the atmosphere is somewhat corrosive.
Package type: wet well with self-priming pumps	
Same as the wet well with suction pump above plus it is less expensive than the package wet well–dry well.	Maximum capacity is about 90 m^3/h (400 gal/min)
Package type: pneumatic ejector	
Suitable for basements and for small capacities and high heads.	Relatively high construction cost.
	Low efficiency.
Can scour force main at low flow rates.	Consider cutter pump (for low head) followed by a centrifugal pump (for medium head) or by a positive displacement lobe or progressing cavity pump (for high head).
	Maintenance of auxiliary equipment is expensive.
Package type: submersible pumps in wet well	
Low cost.	Pump characteristics somewhat limited.
	See wet well submersible pumps and motors above.

Table 25-4. Comparison of Nonclog Wastewater Pumps

Advantages	Disadvantages
Self-priming centrifugal pumps	
Sizes to 250-mm (10-in.) suction/discharge. Flow rates to $0.15\,m^3/s$ at 24 m (2400 gal/min at 80 ft) TDH. Heads to 38 m (125 ft) at less discharge. Low required NPSH. Suction lift to 7.6 m (25 ft). Dry well is unnecessary if suction lift is not excessive. Quick, easy access for impeller cleanout.	Typical efficiencies are 10–20% below other nonclog pumps. Priming depends on the volume of suction line, which must be short. High dynamic suction loss.
Close-coupled centrifugal pumps (common pump and motor shaft)	
Lowest first cost. Most compact. Conditions favoring use. Limited finances. Temporary stations. Operating time less than 2000 h/yr.	Motor bearings must carry the radial and thrust loads of pumping. Some pumps require special motors with a 3-month delay after ordering. Maintenance problems arise because the motor must be removed to replace packing or mechanical seals (not true with some models).
Horizontal frame-mounted centrifugal pumps	
Pump bearings can be selected for long life and suitability for continuous operation with high radial and thrust loads. Does not need a special motor. Coupling between the motor and pump reduces lateral vibrations, permits work on the pump without disturbing the motor. Wide range of pump sizes and characteristics is available.	Higher cost than close-coupled pumps.
Vertical frame-mounted centrifugal pumps in dry well	
Same as for horizontal frame-mounted pumps (see also Table 25-3).	Same as for horizontal frame-mounted pumps plus motor must be lifted to work on the pump or to replace mechanical seals (but not to replace seal packing).
Vertical frame-mounted centrifugal pumps in dry well with long shafts for motors on floor above	
Same as vertical frame-mounted pumps (see also Table 25-3).	Highest cost; shafting adds 5–7% to pump cost. Must analyze the shaft for torsional vibration. Impractical for large ($1.3\,m^3/s$ or 20,000 gal/min) pumps.
Submersible pumps (with motors)	
Wide range of sizes and characteristics—up to $5\,m^3/s$ (80,000 gal/min) for a vertical propeller pump. Conditions favoring use. Need for low first cost. Many other units in same system. See also Table 25-3.	V/S control limited to AFD. Limited U.S. experience with large units. Pump and motor are more expensive. Maintenance requires removal (a messy operation) from wet well and disassembly by trained workers, so factory service may be required at substantial expense.
Screw centrifugal pumps (e.g., Hidrostal®)	
Superior rag-passing capability. Does not clog where other pumps would. High efficiency. Good configuration for dry pit. Operates well at right of BEP.	Single vane impeller. Vibration severe when operating at left of BEP and at very low NPSHA.

Table 25-4. Continued

Advantages	Disadvantages
	Propeller pumps
High efficiency.	
Large flow rates, up to $57\,m^3/s$ ($2000\,ft^3/s$).	Unsuited for raw wastewater flow rates less than $1\,m^3/s$ (15,000 gal/min) and unsuitable for high heads.
Low cost, low maintenance.	Wastewater should be screened for small pumps.
Once available with variable pitch blades, which reduce motor horsepower and starting torque. Considered impractical now.	Variable pitch blades may not be reliable. Check with users.
	Limited to well-screened wastewater or storm water.
Conditions favoring use.	
Large flow rates at low head (9 m or 30 ft) per stage for axial flow and up to 24 m (80 ft) per stage for mixed flow.	
	Vertical column, wet well pumps
Axial- or mixed-flow pumps permit high flow rates and, with multiple stages, reach high head.	Non-clog volute pumps limited to small capacities.
	Radial loads on submerged bearings limit bearing life.
	Diffusion vaned pumps cannot pass stringy solids, not useful for raw wastewater in smaller sizes.
Avoid centrifugal volute-type pumps for wet well applications Use submersible or vertical turbine, solids-handling pumps instead.	Limited stable capacity range.
	Entire pump with column must be removed for service.
Conditions favoring use.	
Large flows following primary treatment or screening.	
Propeller pumps for intermittent service on screened storm water.	
	Vertical turbine pumps for solids handling
Eliminates need for a dry well.	Pump is expensive
True non-clog design; reliably pumps long stringy solids and passes 75-mm (3-in.) spheres (see Section 11-4).	Sizes smaller than 750 mm (10 in.) not available
	Repairs to pump require lifting entire unit out of wet well. Hence, the superstructure must be very high or a hatch must be located for access by a mobile crane.
Flow rates of $0.08-1.9\,m^3/s$ (1200–30,000 gal/min) at 6–21 m (20–70 ft) TDH.	
Motor above ground floor is protected from flooding.	When a pump is removed for service, the wet well is interconnected with the pump room which, depending on the ventilation system (see NEC Art. 500), becomes a Class 1, Group D, Div. 1 or 2 hazardous space.
Capacity of existing pumping stations can be increased by converting dry wells to wet wells.	TEFC motors are acceptable if the room is ventilated at 15 air changes per hour. At lower ventilation rates, explosion-proof motors are strongly recommended (and may be required in the future).
High efficiency over a broad operating range.	All electrical equipment should be protected against corrosion and labeled for hazardous environments. All electrical panels must be in a separate room.
Ideal for increasing the capacity of existing pump stations and for V/S.	
Can operate at very low speed.	As with any column pump, the column must be vented; a conventional air release valve is unreliable. Use open vent with a return to the sump or a solenoid-controlled, pilot-operated sleeve or diaphragm valve.
At least three manufacturers.	
Excellent for trench-type wastewater wet wells.	

Table 25-5. Comparison of Special Design Wastewater Pumps

Advantages	Disadvantages
	Open Archimedes screws
Available in diameters of 0.6 to 3.7 m (2 to 12 ft) and flow rates to 3.8 m^3/s (60,000 gal/min). Maximum typical efficiency is 75% over wide delivery range. Constant-speed drive produces equivalent of V/S pumping; self-adjusts to match inflow Conditions favoring use Low lift, as at treatment plants Little variation in flow level at discharge. Can handle large solids so screening and grit removal unnecessary. Need for V/S pumping with C/S drive. Adequate area outdoors.	Does not pressurize fluid Pumps only to a fixed discharge level Requires critical clearance between trough and screw Efficiency (1) depends on slip due to clearance (so it cannot be pretested), (2) is reduced by *cascade* at exit receiver, and (3) decreases at lower flows due to slip Maximum lift 9 m (30 ft) Lower bearing is submerged in the sump and inaccessible unless the sump is drained Concrete trough is difficult to construct Large units and high lift are subject to binding and excessive backflow. Requires more space than other lift stations. Ice is troublesome at the inlet. Pump may require an insulating cover in cold climates. Releases odor if wastewater contains H_2S. High cost.
	Enclosed Archimedes screws
Same as open screws except Guaranteed efficiency is 85% and only decreases to 80% at low flow. No concrete trough. No leakage. All bearings are in the open and readily accessible. Flow rates from 115 to 8000 m^3/h (500 to 35,000 gal/min).	In freezing weather the pump must be run nearly dry (in reverse) if it goes on standby. Largest size is 3 m (10 ft) in diameter. High cost.
	Pneumatic ejectors
Flow rate range is from 0.5 to 140 m^3/h (2 to 600 gal/min). High head. Automatically variable volume. Batch discharge velocity scours force main. Operation is usually trouble free in small sizes even without screening or comminution. Rugged, reliable, long life. Conditions favoring use. Where centrifugal pumps unsuitable. An air supply is already available. Low (about 11 m^3/h or 50 gal/min), inconsistent flow and high head. Single receiver is acceptable and a standby is not needed (but avoid unless fully justified). Adequate basement for installation.	Maximum efficiency is 60%—typically less due to compressor efficiency and blowdown losses. Some designs are unsuitable for raw wastewater due to the use of conventional swing check valves. No suction lift; fills only by gravity. Maintenance (for some makes) includes unclogging check valves (which are susceptible to plugging) and cleaning electrodes for level controls. Noisy and creates high ambient temperatures in an enclosed space. High capital cost, especially in large sizes where a standby receiver is included. Settling in the discharge pipeline may occur unless the unit is sized properly. Electrode-controlled units corrode. Two compressors and two pressure vessels are needed to obtain a semblance of continuous flow. Flow cannot be increased in the future except by adding ejectors. Adequate basement required for installation.
	Grinder pumps (centrifugal or positive-displacement types)
Many pumps can be connected to a single, small force main. Positive-displacement types can produce flow at heads to 36 m (120 ft) or more with flow rates up to 1.8 m^3/h (8 gal/min) and motors up to 7.5 kW (10 hp). Conditions favoring use. Residences below the sewer main with a long force main. High head, low flow.	Costly. Unsuitable for municipal application. Usually installed one per residence, which mandates prompt repair service at any hour in any weather. Reliability is improved and high pump cost is slightly reduced by a single sump containing two pumps serving two or more residences. Centrifugal types may not pump when the force main is pressurized.

Table 25-5. Continued

Advantages	Disadvantages
Cutter pumps	
Sizes available are 75–150 mm (3–6 in.).	Only three-phase motors are available at 1150 or 1750 rev/min.
Flow rates are $11-340\,m^3/h$ (50–1000 gal/min).	Cost is double that of a comparable centrifugal pump.
Power range (dry pit) is 2.3–30 kW (3–40 hp).	Efficiency is low (30 to 60%) because of the cutting required.
Power range (submersible) is 2.3–15 kW (3–20 hp).	Maintenance of submersible styles requires disconnection and removal, or work must be done in a hazardous area.
Comminutors and bar screens may be eliminated.	
Very effective on all (e.g., overalls and disposable diapers) but the most unusual solids (e.g., panty hose, which are too thin to cut).	
Long life (many years) in severe service.	
Effective with relatively low-horsepower motors.	

Water Pumping Stations

All pumping stations (except wells) require redundant pumping equipment so that while the largest pump is removed for maintenance, the remaining pumps meet all the demands on the station.

Raw Water Pumping

Some influencing considerations that are not under the designer's control include variation in the water level, potential flooding, trash and debris, fish protection, aggressive water, silt and turbidity, microorganisms and slimes, and ice—frazil, surface, and anchor. Head and discharge requirements influence the type of pump and, perhaps, the type of station.

Prior to selecting the pumps, a basic decision must be made on the use of the following:

- Vertical wet well pumps
- Horizontal dry well pumps in a dry well–wet well design
- Vertical dry well pumps in a dry well–wet well design
- Horizontal pumps that are floor-mounted above the suction well, with a priming system.

In some installations, the most economical design is obvious, but in others, alternative designs must be prepared on the basis of first cost and operation and maintenance costs.

- If a deep pump setting is required to reach the water surface and if construction is difficult due to poor soils or groundwater problems, a wet well station design with vertical pumps is always the most cost-effective.
- If the setting is shallow and if the design of the intake works eliminates the need for a separate suction well, a horizontal pump installation with the pumps (below low water) in a dry well is usually the most economical.
- If the distance between the pump room floor and the low water surface is such that a horizontal pump can handle the required suction lift, an economical design is horizontal pumps installed at the ground-floor level. Complete reliability depends on the proper functioning of the pump priming system—particularly in an unmanned station.

A small pumping station may well use a single operating pump and an identical standby unit to reduce the spare parts inventory. The electrical design should preclude the two pumps operating together and should also ensure that each operates alternately. In multiple-pump stations, pumps should be started one at a time. The hydraulic transient or surge analysis should be based on the simultaneous shutdown of all operating units as a result of power failure.

Well Water Pumping

Pumps used for well water are almost always vertical mixed-flow or radial-flow (compared in Table 25-8). If flooding can occur and station reliability is essential, all electrical machinery must be above grade, which might require vertical pumps. If space is limited, vertical turbines can be mounted directly above a clear well or reservoir, although protection against contamination is a significant factor for pumps in a clear well.

The types of wells include:

- Shallow or caisson wells for depths up to 60 m (200 ft)
- Horizontal collectors and infiltration galleries
- Deep, cased (drilled) wells for depths of 6 to 600 m (20 to 2000 ft) and diameters of 100 to 750 mm (4 to 30 in.).

Most situations can be met with the variety of pumping stations compared in Table 25-6 and pumps compared in Table 25-7.

Table 25-6. Comparison of Water Pumping Station Types

Advantages	Disadvantages
Vertical wet well pumps	
Wide selection is available: axial-flow (single-stage), mixed-flow (single- and multistage), Francis turbine (single- and multistage), and radial-flow (turbine in single- or multistage)	High superstructure to pull pumps or pumps can be pulled through a hatch in the roof of a low superstructure by a truck crane parked outside, which may be cheaper than inside crane
Small floor area and small superstructure	Requires disconnecting the motor and pulling the pump for inspection or repair
Ground floor pump motors above flooding	Requires more shaft bearings
Pump suction always flooded; no priming	Priming may be required if air in the pump column causes problems
NPSH easily met	If idle pumps collect air in pump column, either (1) an automatic air vent valve at the discharge elbow or (2) priming must be installed
Ideal for deep installation and large variations in water level	
Vertical turbines are especially flexible in meeting head requirements by adding stages	
Wet well–dry well, horizontal pumps below water level (flooded suction)	
Pump types available include split-case centrifugal, end-suction overhung-shaft, and horizontally mounted axial-flow (propeller and mixed-flow) types	Large pump room floor area
	Greater excavation
	Motors are subject to flooding
	Longer electric conduits
Eliminates priming and air problems	Pumping stations are more costly due to additional floor area, access, lighting, ventilation, etc.
Low headroom requirement	
Easy maintenance and accessibility	Greater forced ventilation needed to cool motors
Service can be accomplished without disconnecting motor	
Horizontal pumps on floor above suction well	
Easy maintenance and accessibility	Requires priming equipment
Short electric conduits	Dependability (due to priming equipment) is reduced
Motors above flood level	Limits the choice of pumps to those for negative suction
Ventilation minimized	
Reduced excavation and below-ground construction. Similar to wet pit	

Table 25-7. Comparison of Water Pumps

Advantages	Disadvantages
Axially split-case pumps vertically or horizontally mounted	
Easy to maintain and to inspect	Requires a housing to prevent freezing and protect bearings and packing from dust
Axially balanced, so there is no axial thrust	
Vertical axial-flow, mixed-flow, and radial-flow (turbine) pumps	
Can be housed in low, flat-roofed buildings with roof hatches for pulling the pump with a crane parked outside	If outdoors, standby units may freeze, noise is a problem in residential areas, and the station is easier to vandalize
Quiet, especially with submersible motors	
Submersible motor style needs no housing except for the control panel and valve vault	
Virtually no maintenance except for pulling the entire unit every 2 to 5 yr	

Table 25-8. Comparison of Well Pumps

Advantages	Disadvantages
Self-priming centrifugals	
Flow rates to 0.25 m^3/s (4000 gal/min)	Limited to caisson, gallery, or wells with a water table less than 4.6 m (15 ft) below the pump
Vertical turbines (V/Ts)	
Most common type of well pump	Efficiency as low as 50% depending on the size and application
Motor- or engine-driven	
Diameter: 50 to > 1200 mm (2 to > 48 in.)	
Flow rates exceeding 0.6 m^3/s (10,000 gal/min)	
Use for finished water and booster pumping is increasing	
Can be tailored for specific head by adding bowls	
Vertical turbines, lineshaft	
Minimum maintenance	Unsuited to crooked wells
Driver accessible	Somewhat noisy
Excellent for wells less than 90 m (300 ft) deep	Suitability is marginal at depths over 300 m (1000 ft). Even at depths less than 180 m (600 ft), shaft stretch wears impellers and bowls unless the thrust bearing is kept carefully adjusted
Vertical turbines, submersible	
Quiet, so they are suitable near hospitals, schools, and residences	Maintenance requires pulling the entire unit
Only solution for crooked wells	Regular maintenance is required
Practical at depths over 210 m (700 ft)	Seal problems may be severe
Water cooling is very effective	Long electric cables
No long-shaft problems	

Booster Pumping

Selecting the type of booster pump is somewhat complicated. The following discussion is oversimplified and should be supplemented by reviewing Chapter 18.

Booster pumping stations fall into two general classes: (1) distribution boosters, which increase pressure to serve a water distribution system at a higher elevation; and (2) in-line boosters on a transmission main.

Boosters should be constructed only when justified by sound economic and operational reasons. The advantages and disadvantages of booster pumping stations, which are shown in Table 25-9, make it

Table 25-9. Comparative Advantages of Booster Pumping Stations

Advantages	Disadvantages
Allows suction side pipeline to be designed at a lower pressure rating and may reduce pipeline construction cost.	Additional construction costs, O&M, and replacement costs.
May avoid designing primary pumping station for abnormally high discharge pressure with resultant cost savings.	Increases operational complexity.
May reduce maximum system pressures over large service areas and reduce energy costs and leakage.	Offsite facilities (access roads and power lines) may be required.
Useful in increasing flow in existing pipeline systems.	Complicates analysis and control of water hammer.
Useful in eliminating substandard pressures in zone extremities.	Requires matching the flows of primary and in-line booster stations.
	Complicates design because head-capacity curves cannot be independently established for either station.
	Large fire flow requirements and small minimum domestic flow needs require a wide range in pumping capability.

necessary to conceive alternative system layouts, some of which might exclude a booster. Make a choice on the basis of a present-worth analysis of capital, O&M, and energy costs as well as adaptability for meeting the requirements of the situation.

The comparative advantages of the different types of pumps for booster stations are the same as for raw and finished water stations. In general, most booster pumps are best served by either a horizontal, split-case, double suction pump or a vertical turbine can pump. The personal preferences of the engineer and owner are important factors. During the past three decades, the trend has been toward the increasing use of turbine pumps, especially in the western United States. However, the selection of the pumping units requires considerable analysis because of the many factors that must be weighed against each other. Some of these are as follows:

- A smaller number of larger pumps is less costly.
- More favorable flow rates with less storage are obtained with more small pumps.
- Pumps of different sizes (proportional to flow ratios of 1:2:4) give the greatest number of flow combinations, but they (1) violate the desirability of standardization, (2) cause one pump to do most of the work, and (3) increase the cost of the standby pump.
- The use of V/S pumping improves flow matching, but it (1) adds complexity, (2) increases the difficulty of the engineering, and (3) reduces reliability.
- Before resorting to V/S, explore all other possibilities first, such as (1) balancing storage at either or both ends of system, (2) using complementary C/S pumps of two or more sizes, (3) bypassing a portion of the discharge, or (4) adding a throttling valve.
- Because of maintenance, capital cost, and inefficiencies, V/S is justified only (1) when feeding large systems with inadequate storage and large demand fluctuations, or (2) in an intermediate zone where both the upper and lower zones contain sources of supply and variable speed is needed to balance pressures.
- Two philosophies in selecting the size of booster pumps are (1) the use of pumps of the same size (which is often better for small stations), and (2) the use of two groups of pumps—small ones for average demands and large ones for maximum demand (which is often better for large stations with heavy fire demands).

Sludge Pumping

Sludge pumping is usually a part of a water or wastewater treatment plant. Hence, sludge pumping stations per se are uncommon and pumps are usually housed in the dry well. Pumps are compared in Table 25-10. Sludge pumping evaluation and design is complex and not straightforward due to variations in the characteristic behavior of sludge as a non-Newtonian fluid, in addition to other factors that render traditional design approaches inappropriate. For an extensive discussion, refer to Chapter 19.

Table 25-10. Comparison of Sludge Pumps

Advantages	Disadvantages
Air lifts	
Suitable for return activated sludge, raw wastewater, and sandy waters. Also used for pumping shallow wells	Limited to low lifts—about 2 m (7 ft)
Maximum flow rate is about 550 m^3/h (2500 gal/min)	Does not pressurize fluid, so they are limited to free discharge
Simple and cheap	Maximum efficiency is 35% or less
No moving parts, very little maintenance	Difficult to regulate discharge, and pumping is unreliable if air is throttled for low flow rates
Conditions favoring use:	Requires a large submergence
Return activated sludge	Sensitive to small changes in head and viscosity
Decanting digester supernatant	Suitable only for thin sludges without large solids
Sometimes for shallow wells where:	Use for flow measurement is unwieldy and inaccurate
Close control of flow is not critical	Requires air compressor or blower
Unusual cost for alternative system	
Air supply is already available	
Digester gas mixing systems	

Table 25-10. Continued

Advantages	Disadvantages
Plunger pumps (positive displacement)	
The most reliable of the positive-displacement types Flow rates to 120 m^3/h (540 gal/min) at pressures up to 13,800 kPa (2000 lb/in.^{2}) or more Can pump very thick sludge, large solids, stringy materials, and grit Slow speed and long life Suitable for suction lift Major repairs can be made in a local machine shop Flow rate is constant despite large changes in head Conditions favoring use: Sludge pumping for digesters High and/or variable head High suction lift Viscous fluids	Maximum efficiency is typically 40 to 50% depending on the type Needs extra attention for lubrication and routine maintenance Requires sizable floor space Solids size is limited by check valve clearance Check valves must be dismantled to locate trapped solids; dual checks are helpful Check valve spheres are short-lived Low capacity; primarily useful only for sludge Air chamber is usually required to prevent water hammer Requires bypass pressure relief or shear pin protection Head depends on peak of sinusoidal flow characteristics. Air chambers are needed to smooth out flow pulsations
Rotary lobe pumps (positive displacement)	
Efficiency near maximum (~75%) at all speeds and pressures Quick, easy replacement of moving parts (in the cantilever type) Check valves are not required Pressures to 690 kPa (100 lb/in.^{2}) Flow rate to about 450 m^3/h (2000 gal/min) High tolerance for rags and stringy materials and can pass solids to 120 mm (4 in.) in diameter; easy access for cleanout Long life at low speed Meters the flow accurately despite changes in TDH Good for viscous sludge; creates the least shear rate Can run dry or in reverse without damage Smooth, relatively nonpulsating flow Conditions favoring use: Where trouble-free operation and low maintenance are more important than capital cost Where floor space or building cost is at a premium	High first cost (about 130% of the cost of a progressing cavity pump) Efficiency is low for thin liquids Comminution, cutting, or grinding solids and grit removal (for moderate to high grit content) is an essential pretreatment In usual service, replace lobes yearly, seals every 2 years at a typical annual cost, depending on the model, of $1200 to $3000 in 1997 Suction lift due to slip is limited to 3 m (9 ft) Side clearances are abraded by grit, so avoid where grit loads are heavy Bypass relief pipe is required to prevent damage if discharge is plugged or valve is closed Limited to a maximum solids content of about 5%
Diaphragm pumps (positive displacement)	
Simple field repairability High pressure Suitable for suction lift Moderate shear rate Low cost Can meter flow Handles viscous liquids, sandy and muddy waters Ease of repair Conditions favoring use: Temporary use, especially with a gasoline engine outdoors Low flows Dewatering excavations Slurry pumping High grit loads	Instant shut-down on component failure. Check valves are required Solids handling ability is limited by ball check Poor for rags, sticks, and string

(*Continued*)

Table 25-10. Continued

Advantages	Disadvantages
Progressing cavity pumps (positive displacement)	
Typical maximum efficiency is about 75%	High maintenance cost. Typically, stators replaced yearly, rotors rebuilt every 2 yr
Good solids handling for abrasive and viscous liquids	Large floor space and clearance are required—especially to pull rotor
Generates pressures to 7000 kPa (1000 lb/in.^{2})	Unsuited to heavy grit loads
Maximum flow rate is 110 m^3/h (480 gal/min)	Cannot pump solids larger than 45 mm (1.8 in.). May require in-line grinding as pretreatment
Rotor acts as a check valve	Cumbersome piping
Smooth, relatively nonpulsating flow	Instant failure when run dry or if the discharge plugs. Safety devices for (1) high pressure, and (2) no flow are recommended
Low capital cost	Elastomer (material in stator) limits the type of liquids pumped
Handles gas inclusion	Slow speed drive required
Conditions favoring use:	
High head, variable head	
High suction lift	
Grit-free sludge with a high solids content	
Nonclog centrifugal	
Wide selection of flow and head applications	NPSH considerations may be critical
Vertical and horizontal configurations are available	May be difficult to size large enough to pass solids without clogging, yet small enough to avoid dilution by drawing in overlying liquid (rat-holing)
If applied accurately, pump efficiency is high and pumping is economical	Requires capacity adjustment (variable-speed drives)
Conditions favoring use:	Needs separate flowmetering for process control and speed control
Thin sludges without debris (waste and return activated sludge)	Pump clearances must be adequate to pass the material typically contained in sludge
High volume of sludge to be pumped	
Need for economic pumping outweighs maintenance costs	
Vortex pumps (recessed impeller, centrifugal)	
Readily passes solids (debris, grit, rags, stringy materials)	Efficiency only 35 to 50%
Good longevity	Very flat H-Q curve
Materials for gritty slurries available	Usually requires variable-speed drive
Excellent reliability	Must be accurately sized to avoid excessive recirculation
Conditions favoring use:	Cannot trim impellers of U.S. makes
Primary sludges	Increase of head greatly reduces capacity
Undegritted sludges	
Need for compactness	

25-7. Power, Drivers, and Standby

Early decisions about drivers and standby units affect the type, configuration, and physical size of the pumping station. The preliminary design considerations should include plans for the electrical system, such as load estimating (power and lighting), load data collection, load characteristics, selection of the power source, plans for load growth and change, selection of the best voltage, and selection of the best distribution system for reliability, flexibility, safety, and maintainability (see Chapters 8 and 9).

Motors

A generalized guide for selecting C/S drivers is presented in Table 25-11, and a guide for selecting V/S drivers is given in Table 15-2. About 90% of all drivers are electric motors, and the squirrel-cage induction motor, by far the most common, is available in a wide variety of casings, windings, insulation, allowable temperature rise, shafts, and bearings. Synchronous motors are occasionally used for very low-speed applications or for drivers larger than 375 kW (500 hp). Wound-rotor motors are sometimes used for V/S applications, but they are expensive and maintenance is greater than with squirrel-cage motors. Direct-current machines are rarely used in pumping stations.

NEMA Designations

NEMA *design letters* A, B, C, and D are used for designating the starting torque and starting

current of general-purpose motors up to 150 kW (200 hp).

- Design A (never used in pumping stations) is normal starting torque and normal starting current.
- Design B is normal starting torque and low starting current (the most common type in pumping stations).
- Design C is high starting torque and low starting current.
- Design D is high starting torque, low starting current, and high slip.

NEMA *code letters* are different from NEMA design letters. The code letter identifies the starting conditions in ''kilovolt-amperes per horsepower'' when the motor is started on full voltage. These data are primarily used to determine if the capability of the electric supply or on-site generator is sufficient for starting and picking up the load.

Starting in-rush current requirements are needed for sizing the emergency generators, conductors, and controllers and are of interest to the power company. The motor controls should permit starting only one unit at a time and, if drivers of different sizes are used, the emergency generator size can be minimized by starting the largest motor first.

Hazardous Locations

A Class I, Division 1 wastewater wet well requires full containment of electrical conductors and equipment. A nonexplosion-proof submersible motor can be made compliant by using a redundant low-level float switch to disconnect the motor and sound an alarm when the liquid level falls to the top of the motor. Class I, Division 2 allows the use of non-explosion-proof wiring and equipment provided it does not produce arcs (see NEC 501-3).

Close-Coupled Motors

Close-coupled motors have a long shaft that carries the pump impeller. Radial loads for multivaned impellers pumping clear water are relatively small, but radial loads for impellers pumping raw wastewater are large; they fluctuate (twice per revolution for a two-vane impeller) and, with large solids present, the fluctuations are irregular. Shafts or bearings and seals in a close-coupled unit pumping wastewater may be short-lived, particularly at high discharge heads unless the unit is specially designed for severe service.

Motor Specifications

Motors located indoors and above grade can have open drip-proof enclosures. These are the least expensive, and they allow motors to run cooler than other enclosures. Some designers suggest splash-proof enclosures for some environments so that the entire area can be washed with a hose. To provide protection from flooding, specify installations below grade to be either (1) open, drip-proof enclosures with encapsulated motor windings, or (2) totally enclosed, fan-cooled motors. Outdoor installations should be specified as Weather-Protected II enclosures. For Class I, Division 1 hazardous locations, specify that ''motors shall be rated Class I, Division 1,'' which is a totally enclosed, explosion-proof motor. For Class I, Division 2 locations, the open, drip-proof enclosure is acceptable for squirrel-cage motors if there are no switches or other arc-producing components in the motor.

Other specifications for motors should include:

- Class B temperature rise
- Class F insulation
- 480-V, three-phase power for motors from 2 to about 60 kW (2 to about 75 hp)
- 2300-V, three-phase power for motors of 75 kW (100 hp) or more
- A service factor of 1.15
- High energy efficiency and a high power factor (because these terms have no accurate definitions, state the actual values in the specifications)
- Blower, fan, and centrifugal pump motor torque characteristics that fit NEMA Standards Design B
- Antifriction ball bearings with a life of 100,000 hours for normal loads and 30,000 hours for severe loads. For motors up to 300 kW (400 hp); ball or roller bearings can be used.
- NEMA code letters F or G for motors of 150 kW (200 hp) or less (see Chapter 13).

Variable Speeds

Use V/S drives only if necessary and do not use complex control systems (such as AFDs) for remotely located pumping stations, except with the client's full understanding of the problems stated in the subsection entitled ''Adjustable-Frequency Drives'' in Section 15-11. Specify unit responsibility (see Chapter 16) for the drive and pump, but, if an AFD is to be used, either (1) specify that guaranteed service and critical parts will be available within a reasonable distance (say, 160 km or 100 mi), or (2) make sure that the client will have trained electronic technicians on the staff or that there is a reliable, qualified service organization

Table 25-11. Guide for Selecting Drivers

	Application criteria		
Type	Power limits	Reliability	Capital cost
Electric motor (direct drive)	No practical limit	Good, depending on utility	Low
Electric motor (belt drive)	600 hp	Good to poor depending upon frequency of belt and pulley replacement	Low
Electric motor (gear drive)	No practical limit	Good (depending upon utility), but has additional maintenance cost	Low
Engine (direct drive)	No practical limit	Excellent, especially with on-site fuel storage or production	Moderate to high
Engine (gear drive)	No practical limit	Same as direct-drive engines	High
Engine/motor (with clutch or declutching)	Generally less than 200 hp	Superior, especially with onsite fuel storage or production	High
Engine-generator standby duty	No practical limit	Superior	Very high

nearby. If V/S is necessary and AFD cannot be justified, and if slightly higher power losses can be tolerated, consider the simpler slip drives such as the eddy-current coupling (see Sections 15-1 and 15-11 and Example 29-1). One way to "improve" the efficiency of slip drives is to design so that a pump rarely operates at less than about 85% of full speed. Also consider the almost 100%-reliable combined squirrel-cage motor/diesel engine drive where the engine with a larger rating is used both for peak flow and for standby power. If V/S is used for one pump, there must be a V/S standby. A combination of a V/S pump and a C/S pump is only satisfactory if the discharge capacity of the V/S pump exceeds the capacity of the C/S pump by at least 50%. If the two pumps have equal capacities, the V/S pump will "starve" on too little flow and will wear out quickly for the reasons given in Sections 10-6 and 15-5.

Flat H-Q curves for both the pump and the system must be avoided because, with an acute angle of intersection, small changes in head (due to air in the force main, changes in the wet well level, or a change in speed due to fluctuating voltage) cause highly unstable operation. The best operational approach is to start a second V/S pump when the lead V/S pump capacity is slightly exceeded and to run the two pumps at the same speed in a "load-sharing" operation, which is more efficient than operating the two pumps at different speeds in "staggered operation" (see Figure 15-8).

Miscellaneous but Important Considerations

The following miscellaneous considerations are also important:

- For flood protection in dry wells, consider (1) submersible pumps and motors, (2) vacuum and pressure epoxy encapsulation of the motor windings with a flood level float to disconnect the motor automatically before inundation, (3) vertical pumps with extended shafts, or (4) immersible motors.
- For high-service duty, select the slowest speed that can perform satisfactorily. Slow-speed motors and pumps cost more, but wear is a function of speed to about the third power.
- Consider slip or high-slip (8–13%) motors with positive-displacement pumps for high static heads or long force mains.
- Keep the motor control panels in a clean, dry environment above grade and above the flood level, but locate them as close to the motor as possible to keep the leads short.
- Calculate the number of motor starts per year and obtain motors that will last for many years by

Table 25-11. Continued

Application criteria		
Operating cost	Speed limits	Reliability
Moderate to high	Essentially limited to 1800, 1200, 900, or 720 rev/min for 60 Hz current; variable speed possible	Varies depending upon size and system configuration-synchronous drives can be complex
Moderate to high below 1720 rev/min	No practical limit to mechanically adjustable speed	Moderate, but if speed changes constantly, belts wear quickly. If speed is seldom adjusted, grooves worn into cone pulleys make adjustment impossible, so specify chromium-plated pulleys
Moderate to high	No practical limit either up or down; variable speed (through motor speed adjustment)	Moderate
Moderate to low, depending upon fuel cost. Can be very low if engine waste heat can be recovered for heating or cooling	Limited to available engine speeds; variable speed easily achieved	Moderate to very complex
Same as direct-drive engine	No practical limit either up or down; variable speed easily achieved	Moderate to very complex
Moderate to low, depending upon fuel cost	Practical limit is 1200 rev/min; variable speed is easily achieved with engine	Moderate to very complex
Moderately low in standby duty. Must exercise frequently	Same as electric motor	Very complex

optimizing life-cycle costs. Specify the minimum cycle times and the minimum life required for motors and obtain the manufacturer's guarantee in writing (never verbally). The best economy is often found in a special, more expensive motor.

- Limit the permissible deflection of the impeller shaft at the wearing rings under the worst specified continuous operating condition (see Sections 10-6, 11-3, and Appendix C).
- Specify the bearing grade. Some designers require certified calculations of pump bearing life at the design operating condition.

Engines

Internal combustion engines are used sparingly as prime movers. Nevertheless, they are more reliable, often more economical (especially if operating on biogas), offer improved protection against surge damage (because of the increased inertia of moving parts), and are easily controlled in V/S operation. But they are noisy, complex, and require skilled mechanics for maintenance and repairs. If engines are seriously considered, consult the application engineering departments of equipment manufacturers—certainly before proceeding with the design. *Never* install engines below grade.

Fuel

Consider diesel, natural gas, biogas, and LPG (propane) fuels. For engines over 560 kW (750 hp), consider dual fuel. Gasoline is unsatisfactory because its storage properties are very poor, and it has a low flash point, which increases the danger of explosion and fire. Diesel is much less hazardous, and it can be stored for approximately a year. An advantage of LPG is its unlimited storage life.

Duty

The term *duty* means the time-based utilization of the driver and the specifics of the application such as direct drive or electrical generation and emergency or standby duty.

- Direct drive is the most efficient. Gear reducers permit any pump speed at the best engine revolutions per minute. If gears are required, they should have a nonreverse mechanism to prevent reverse rotation of the engine and the interruption of lubrication.

- Continuous duty engines should be designed with conservative rating factors to meet the objectives of reliability, low operating costs, and long service life.
- Standby duty engines should be selected for rapid starting and an ability to develop full load quickly. Specify starting aids, such as two-stage trickle chargers for batteries, jacket water and lube oil heaters, and high torque versus speed.
- Electrical generation requires a careful analysis of the loads (see Example 9-1) and thoughtful application to the generator (see Example 9-10 and Section 26-11). The manufacturer should confirm the analysis and make the final equipment selection.

Standby

Standby may be supplied in four ways:

- Dual electrical power systems. Dual power does not guarantee that a general failure of interlocked systems (which has occurred over very wide areas) or that a failure in the main switchyard (as by a lightning strike) will not occur.
- Standby engines with either clutches and direct drives or an engine-generator.
- Use of only engines as prime movers with one engine and pump standby. Diesel fuel provides the utmost in reliability, and natural gas is much more reliable than electrical power. For example, the only interruption in natural gas to an area in southern California occurred in the 1932 Long Beach earthquake.
- Storage tanks or reservoirs for the time anticipated to restore power. The time cannot be known, so consider a suitable safety margin.

If standby electrical generation is required for a pumping station with electric motors and V/S drives, using engines and direct drives (with microprocessors to trace the system head curve) is more economical in both capital and operating costs (assuming a 56-kW (75-hp) station and energy at 6¢/kW · h for electricity and 36¢/therm for gas). Maintenance for engines is much greater than for motors, but the savings in capital and operations is more than adequate for the maintenance and ultimate replacement of engines (see Section 8-4 for engine-generator requirements, Example 9-10 for engine-generator sizing, and Chapter 14 for engine requirements).

Pumping Units versus Station Capacity

Optimize the size of the wet well and the number of pumping units so that the permissible number of starts per hour is never exceeded. Note, however, that the use of solid-state soft starters in combination with inverter duty motors allows frequent starting (hence, smaller wet wells) as explained in Section 8-3 and in "Critique of Example 12-2." Consider these factors:

- Present average and peak flow rates
- Ultimate design average and peak flow rates
- Time period until the ultimate design flows are obtained
- Desirability of adding pumping units in the future
- Size and number of pumping units required
- Size of standby power unit(s) needed to operate the station.

In general, the minimum number of pumping units is as follows:

- One pump and one standby for small stations [e.g., $< 160\,m^3/h$ (<700 gal/min)]
- Two or three pumps plus a standby for medium-sized stations [e.g., $160–450\,m^3/h$ (700–2000 gal/min)]
- Three to five (or more) pumps plus standby for large stations. Consider V/S or a combination of V/S and C/S pumps (see Chapter 15).

25-8. Application-Engineered Equipment

Consider specifying application-engineered equipment for those facilities considered to be critical to the overall function of the installation. Included in this category are the main pumps and motors, engine-generators, and engine drives for pumps. The hallmark of application-engineered equipment is the design of machines for the specific installation and application. Generally, a greater level of conservatism is employed for more robust equipment (and therefore more reliable with longer life) than that obtainable from standard machines. The following are suggestions for improving the performance of typical machines.

Pumps

- A maximum permissible pump shaft deflection of 0.05 mm (0.002 in.) at the pump seal is suggested for any operating condition specified. Pumps can usually be modified to accept larger bearings and shafts.
- Specify bearings with L-10 ratings of 100,000 hours for loads imposed at the BEP and 30,000 hours at the worst operating condition to be expected. See

Figure 10-18 for an approximate relationship between best and worst conditions. Some engineers specify L-10 ratings of 100,000 hours for the worst conditions, but others insist that the oversize bearings have a tendency to induce failure from skidding. Discuss this problem thoroughly with the pump manufacturer.
- Specify either (1) austenitic cast iron (ASTM A439), or (2) 2 to 3% nickel in iron castings for improved resistance to corrosion and erosion. Alternatively, consider applying ceramic coatings such as Thortex Cerami-Tech® [4] to internal pump surfaces to reduce wear and erosion (see Section 24-10 Item 19).
- Specify cast stainless steel or aluminum bronze for impellers where NPSHA margins are borderline to reduce erosion due to cavitation.
- Require the pump rotor unbalance to be no greater than grade G2.5 per ISO Standard 1940/1.
- Require the pump and driver supplier to perform a torsional analysis for all systems that are to operate at variable speed and all engine-driven systems, including constant-speed systems such as engine-generators.

Motors

- Specify NEMA Class H insulation, but require the motor be designed for a Class B temperature rise. Note, however, that the thicker insulation requires wider slots and thereby reduces the amount of iron and the efficiency. (Otherwise, specify an inverter duty motor.)
- Require a 1.15 service factor over nameplate rating, but prohibit the use of the service factor for driving the machine at any specified operating condition.
- Specify inverter duty motors for all variable-speed applications.
- Specify IEEE efficiency-compliant motors for cooler operation.
- Consult the vendor for derating motors at altitudes above 1000 m (3000 ft).

Engines

- Derate engines to allow no more than 85% of the manufacturer's continuous duty rating.
- Pay attention to elevation and temperature conditions to make certain engines are selected properly.
- Insist upon a thorough mass elastic system analysis.
- Do not allow controls to be mounted on the engine.

Generators

- Specify the same construction as detailed above for motors.
- Require solid-state field-forcing regulators.
- Specify generator loading criteria specific to the application—that is, if the generator is to start 75 kW (100 hp) motors in sequence, so state. Require the manufacturer to produce a voltage transient study to demonstrate that the generator and engine have been selected for the actual starting duty.

Machine Foundations (Baseplates) and Installation

Commonly accepted practices for preparation of machine foundations and for machine alignment are often inadequate. Improper preparation of the foundation results in poor support of the equipment, progressive misalignment of the machines, excessive vibration, and high maintenance costs. Sometimes, even manufacturers' recommendations for equipment supports are found wanting. See API 610 for excellent specifications for baseplate installation details. The key elements of good practice are given in Section 12-8, Items 26 through 29. Although the cost of that procedure is somewhat greater than that of commonly accepted practices, the results can be startling in terms of reduced vibration and lengthened equipment service life. One company was able to reduce the magnitude of equipment vibration by more than half by using it [5]. Another company credits improved equipment support for the major part of an increase in mean time before repairs are needed from six months to 57 months—an increase of 950% [6].

In addition to improved equipment support design, owners have found that advanced alignment techniques, using laser- and computer-based equipment, have resulted in improved power train alignment and increased operating efficiency. This equipment is also capable of detecting support imperfections on both driving and driven machines. Piotrowski [7] stated that even small amounts of misalignment result in significant increases in the loads applied to bearings, shafts, and seals. Machine alignment should be the responsibility of specialists using a variety of tools to align all elements of the power train accurately.

25-9. Station Auxiliaries

Just as much care is required in the engineering and equipment selection for station auxiliary systems as

for the main pumping equipment. In retrospect, station reliability, as determined by the ability of the main pumping equipment to perform their intended function, is greatly dependent upon these systems. A host of benefits flow from properly engineered station auxiliary systems:

- A wholesome environment, free from condensation and dangerous gases, and affording protection for personnel, equipment, and structures alike (see "Ventilation" in Section 25-3).
- Reliable protection against flooding (sump pumping system).
- Convenient removal and replacement of heavy equipment components (hoisting equipment).
- Reliable supply of housekeeping and shaft seal water for cleanliness and protection of equipment (utility water supply).
- Warning of probable system trouble before it becomes a crisis (system controls and alarms).

Some insights into the considerations that enter into design and equipment selection for these systems to provide the best investment are considered in the following.

Ventilation

Guidance for the design of ventilation systems for wastewater pumping stations is given in Chapter 23 and in NFPA 820.

Sump Pumping System

Often given short shrift in the design of many installations, sump pumping stations provide a vital service: *They keep the floor dry*. To perform this function satisfactorily, they must be able to accomplish the following:

- The sump pumping station force main must discharge with an air break above the highest possible water surface at the point of disposal for station drainage. The best is at some location where isolation from the station is possible—for example, upstream from the influent manhole, where a sluice gate has been provided, or into the main pumping station discharge manhole.
- Sump pumping stations should contain two pumps, and each should be capable of accommodating all normal operating conditions.
- The pumps must be capable of passing solids of the size commonly encountered. For an example, in a wastewater pumping station, the sump should be capable of passing a 50-mm (2-in.) solid.
- The operating conditions should be close to the BEP to avoid damaging vibration and radial thrust.
- Large rates of flow should be able to enter the sump through a grated opening. Avoid sumps sealed with a solid cover plate.

If all of the above rules are followed, the sump pumps will have a capacity of $34\,m^2/h$ (150 gal/min) or more, because the requirements to pass large solids and to discharge to a higher elevation dominate all other considerations. However, since the pumps will not be required to operate very often (leakage and other contributions under most conditions are quite small), the sump itself need only be of sufficient size to accommodate the installation and removal of the pumping equipment. If a grated sump cover is used, ample capacity is available if a pipe coupling comes loose or breaks and large quantities of liquid enter the room.

Hoisting Equipment

Electrified hoisting equipment is recommended for all but the very smallest stations. Large items of equipment or subassemblies from large units are awkward to move and place. As the persons charged with this work typically are not skilled in using hoisting equipment, specify crane trolleys, bridges, and hoists with the slowest possible speeds. On cranes and hoists of more than 3600 kg (4 tons) rated capacity, specify inching motors to assist in placing and removing large, bulky objects. Inching motors are necessary when precision alignment is needed because of coupling requirements.

Cranes in wastewater pumping stations are subjected to varying concentrations of hydrogen sulfide. Such cranes should be constructed of corrosion-resistant material, and all the equipment should be explosion-proof if located in the wet well area within an enclosure. Power to all electrical devices must be supplied by festooned cables.

Cranes should have full, vertical access to heavy equipment in lower rooms through removable cover plates or supports for equipment located overhead (such as motors). Do not expect that items of equipment can be jockeyed under an opening from an equipment position using tie-back tackle or "coffin hoists"—a dangerous practice. Design doors that are wide enough and high enough for trucks to be driven under the crane for removal of the bulkiest equipment. *Do not* assume equipment can be placed on wheeled carts moved by hand outside the structure

for mobile cranes or jib cranes to hoist onto trucks. Cranes require rigid frames or diagonal bracing to withstand lateral and longitudinal forces.

Specify cranes and hoists to conform to the standards of the Crane Manufacturers Association of America (CMAA) and the ANSI/ASME HST series of specifications. The exact CMAA and ANSI/ASME standards or specifications for the particular type of crane or hoist should be identified in the project specification. In addition, see OSHA CFR 1910.179 for U.S. federal government requirements pertaining to the required safety requirements, inspections, and operational testing requirements for cranes and gantries. Specify that the installation must either (1) be tested and licensed by the local state entity providing this type of service (identify the specific state agency that requires and verifies the tests); or (2) conform to state requirements regarding crane owner self-certification for operational testing and examination. Check which approach is used in your project state. In addition, the specifications should include crane operation and safety training.

For small stations, lifting eyes (or hooks) over the equipment are feasible (although awkward and somewhat dangerous) for all items capable of being lifted by manually operated, portable hoists. Design and specify lifting eyes to be strong and substantially anchored into the overhead structure as shown in Figure 25-2. Be sure to design adequate headroom for the hoist and the machinery so that dismantling is unnecessary. Lifting eyes are completely inadequate for medium-sized and large stations. The design maximum capacity should not exceed 1800 kg (2 tons) and preferably no more than 25% of that capacity.

Monorails, which may be adequate in medium-sized stations, are often supported by bolts in concrete roof slabs. Design the bolts and the slab to resist impact as well as static loads. Overloading the anchorages is frequently a problem that violates OSHA requirements (see ANSI B30.2.0, B30.11, B30.16, and B30.17). Avoid the use of threaded inserts (into which bolts are screwed), whether cast in place or drilled. Instead, use bolts with adequate mechanical anchorage. An alternative is a low superstructure with a large hatch in the roof for access by a mobile crane parked outside (see CHMA standards).

The lifting effort of cranes or truck-mounted booms actually used in lifting submersible pumps should preferably not exceed about 150% of the weight of the pump and must not exceed the allowable strength of the discharge elbow and its eccentrically loaded connection to the concrete base. In at least one known accident, the discharge elbow and pipe were pulled up with the pump, thus leaving the owner with a very expensive repair job. Anchors for discharge elbows should be buried deep enough to develop the ultimate tensile strength of the anchors, and the anchors should be surrounded with a strong cage of reinforcing bars to prevent the gradual cracking and deterioration of concrete caused by vibration of the pump.

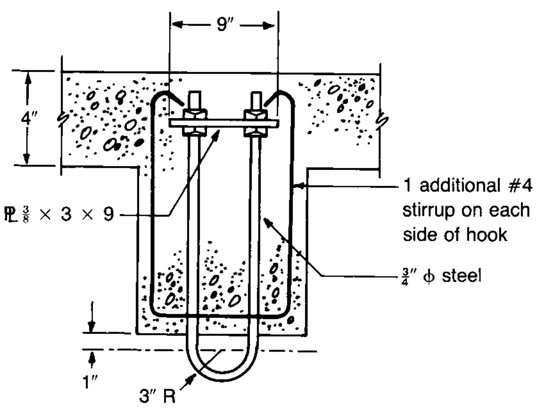


Figure 25-2. Lifting eye or hook for a maximum capacity of 2 tons and a desirable capacity of half a ton.

Compressed Air Systems

Air compressors, particularly the positive-displacement type, can be high-maintenance items. Be conservative about estimating air demand, then double it for leakage at joints, etc. Select compressors on the basis of at least 10 times that figure. Be generous when calculating distribution system losses, and avoid selecting a compressor at the extreme limit of its range. For example, do not select an 80-ft^3/ min, 100-lb/in.^{2} compressor for an application where the calculated figure is 75 ft^3/ min at 90 lb/in.^{2}

Furnish at least two compressors and plenty of receiver capacity. All distribution piping should contain sediment and water traps at low points. If it is a large system, consider using terminal receivers in addition to the control receivers at the compressor location.

Instrument air systems require non-lubricated compressors and some means of removing water vapor from the compressed gas stream. Refrigerated dryers work well for this purpose if the piping system is not exposed to freezing temperatures. If freezing temperatures may be encountered, use a heat-reactivated, desiccant-type downstream from the refrigerated dryer to produce a pipeline dew point of −73°C (−100°F). (See also Section 20-3.)

Water Systems

Usually, two water systems are required in a pumping station; one to provide a pressurized supply for shaft seal purge and lubrication, and the other for housekeeping purposes. The seal water supply requirements (small demand, high pressure) can usually be satisfied by using a pair of regenerative turbine pumps (two are required for redundancy) operated on the pressure in a hydropneumatic tank. To avoid air binding, use a diaphragm-type tank for control and storage. The multistage centrifugal pump is an alternative to the regenerative turbine. (See Section 11-10.)

Housekeeping water supply requirements of 2 to 4 L/s (30 to 60 gal/min) at 600 kPa (90 lb/in.^{2}) gauge pressure or more can be satisfied by providing start/stop controls at each utility station. An overriding timer can be provided to shut the pump down after a timed interval (say, 15 min), thereby avoiding the possibility the pump may be operated against a closed valve.

State regulations for protection of potable water systems against backflow require at least the installation of a reduced pressure principle backflow prevention device on the water supply to the station premises. Some states require an air break tank for absolute protection. Locate these in an area with good free drainage to the dry well. When these devices fail, they frequently release considerable quantities of water.

Alarms

Alarm functions should be from devices independent from controls systems because the problem could very well be caused by a malfunctioning control system. Alarms should be configured to warn of abnormal system behavior before the problem becomes a crisis. For example, in a two-unit system designed for ample performance with one unit in service, furnish the following features:

- Use a broad-band alarm switch with two pairs of contacts.
- Set the alarm to initiate at a value less than that selected to start the follow unit.
- Set the second pair of contacts to open at a value below that which will shut down the follow unit.
- Wire the switch contacts to maintain the alarm condition until the second set of contacts has opened.
- Provide the system with a manual lead/follow selector switch.

With the above arrangement, if the lead unit should fail or an unusual demand is placed upon the system, an alarm will be initiated. The second unit will operate, but it will not be possible to cancel the alarm condition until the controlled variable is within the range of the lead unit. The alarm condition serves as a reminder to the operating staff that something is wrong and encourages investigation. An example of this arrangement, portrayed as the controls for a sump pumping station, is shown in Figure 21-4.

25-10. Instruments and Control

A good basic rule is to avoid too much instrumentation because it increases costs and adds complexities that can compromise the entire system.

Typical Minimum Instrumentation and Control

Every pump should have a pressure gauge (or a tap for one) on the discharge piping. It is the "stethoscope" that is used to check the condition of the pump and force main or transmission pipeline—conditions such as worn impellers, blockage in the force main, deposits in the transmission pipeline, and partly clogged pumps. The gauges can be permanently mounted (which is convenient but costly if there are many pumps) or portable and attachable with quick-connects (which is less expensive, more easily calibrated, and less subject to wear, but less convenient and less apt to be frequently used). A good gauge installation is shown in Figure 20-6.

Other instruments and controls include the following:

- *Incoming power feeders:* voltmeter, ammeter, and watt-hour meter; the watt-hour meter readings may differ somewhat from those of the utility's meter.
- *Motors:* elapsed time meter; on large motors [$>$ 400 kW ($>$ 300 hp)], individual ammeters and thermal protectors.
- *Engines:* instruments and controls as recommended by the manufacturer for the particular engine, and its auxiliary systems such as fuel, cooling system, and storage tanks.
- *Wet well:* level sensors (bubbler or float switches) with low- and high-water pump controls and low- and high-water alarms.
- *Wet well storage tank:* level sensor (pressure transducer, capacitance, or ultrasonic) with high and low levels and alarms.

- *Dry well:* water level sensor for flood alarm.
- *Boosters:* pressure switches.
- *Water wells:* pressure switch, automatic timer, or on-off switch actuated from a central station.
- *Chlorination equipment:* if necessary.
- *Flowmeter (optional):* choose reliability and simplicity and allow enough room for installation according to the manufacturer's specifications. Most flowmeters require a straight entrance of 15 pipe diameters, and even magnetic flowmeters require two to five pipe diameters.
- *Multiple pumps:* controllers to start one at a time on a "first on-first off" basis; some experts object to automatic duty rotation unless the number of starts per hour would otherwise be excessive. Automatic alternators do add complexity, and wear can be equalized (if desired) on a periodic basis by operating personnel.
- *Variable speed:* controls and instruments as required by manufacturer.

Automatic Control

Entirely manual control is not economically efficient. Properly designed automation

- reduces operator workload;
- optimizes operational efficiency;
- avoids potentially dangerous occurrences.

Automatic controls can be anything from a simple float switch that turns motors off and on to a microprocessor or even a computer. Standardized (or "off-the-shelf") microprocessors are efficient, effective, reliable, and relatively inexpensive. Like computers, they are getting better and less expensive. Microprocessors, in particular, have several advantages over simple switches even for simple process controls such as the level in a wet well or storage tank (see Section 20-11 and Chapter 21 for additional discussions of automatic controls).

Recording

Recording instruments vary from simple, circular chart recorders to strip chart recorders that document many operational variables at once, either at the station or at a centralized location. Consult the owner to determine whether recorded data are needed. Unless the data will be used, recorders should be omitted. However, consider using digital data storage (tape cassettes) that can be recycled, or consider digital recorders that automatically produce a properly annotated chart when abnormal conditions occur. Some information could be invaluable if a malfunction occurs in an unattended station.

Complex Instrumentation

Sophisticated instrumentation can take many forms such as: (1) local monitoring and control, (2) local control with central monitoring, (3) local analog control with digital monitoring, and (4) distributed digital control. The total initial costs vary from about \$300,000 to about \$1,000,000. In the late 1990s, the total cost per month (including amortization, energy for instruments and control, and salaries) ranged from about \$20,000 to \$40,000. Thus, sophisticated control instrumentation is very expensive (see Chapter 21).

25-11. Structural Design

Many parameters differentiate the design of a pumping station from other commercial or industrial-type structures. The two most important are:

- The structure must be watertight
- The structure will be exposed to a potentially corrosive environment.

Pumping stations, as with most sanitary structures, are usually required to be continuously operated for their entire service life of 30 to 50 years. Hence, durability and serviceability are overriding considerations. Conventional concrete structures are designed to crack—the very assumption upon which strength design is based. Although it is virtually impossible to design conventionally reinforced concrete structures without cracks, crack depth, width, and number can be reduced by modifying the design procedures of ACI Building Code 318. When cracks are reduced, the permeability of the concrete is also reduced and the durability of the structure is increased. To achieve these ends, ACI 350 (Concrete Sanitary Engineering Structures), produced in 1983, should be used instead of ACI 318 for pumping station concrete design. As compared with ACI 318, ACI 350:

- Lowers the allowable tensile stress of reinforcing by requiring either the use of working stress design or by requiring higher load factors for strength design.
- Emphasizes serviceability requirements and lowering of the "z" factor.
- Suggests guidelines on mix design components and proportions.
- Increases minimum reinforcement requirements.

- Increases limitations for construction and expansion joint spacing.
- Guides concrete coating use and selection.
- Discusses and guides design for impact and dynamic loading.

Abbreviations and Definitions

Consult ACI Building Code 318 for a more complete list of abbreviations and definitions. Those given here are primarily for the figures in this chapter.

A_s	Area of tension reinforcement.
A_{vf}	Area of shear friction reinforcement, in.^{2}.
b	Breadth.
C_3A	Tri-calcium aluminate.
f_c	Specified compressive strength of concrete, lb/in.^{2}.
f_y	Specified yield strength of reinforcement, lb/in.^{2}.
h	Height or total thickness, in.
ℓ_d	Development length; in. $= \ell_{db} \times$ modification factors.
ℓ_{db}	Basic development length, in.
U	A factor (multiplier) used on the live load for ultimate strength design.
V_n	Nominal shear strength, lb/in.^{2}.
V_u	Factored shear force at a section, lb.
Z	A distribution factor for flexural reinforcement. See Equation 25-1.
ϕ	Strength reduction factor.
μ	Coefficient of friction.

$1\,\text{k/in.}^2 = 1000\,\text{lb/in.}^2 = 6.9\,\text{MPa} = 6900\,\text{kPa}$
$1\,\text{k/in.} = 1000\,\text{lb/in.} = 0.175\,\text{kN/mm}$
$1\,\text{in.}^2 = 645\,\text{mm}^2$.

The factor Z is defined by ACI 318 as

$$Z = f_s(d_c A)^{1/3} \qquad (25\text{-}1)$$

where f_s is the computed tensile stress in a steel reinforcing bar, d_c is the depth of cover over the bar, and A is a cross-sectional area of concrete equal to $2d_c \times$ bar spacing.

Geotechnical Considerations

The purpose of most wastewater pumping stations (raising the hydraulic grade line to minimize pipe depth) means that these structures are usually founded quite deep—frequently below groundwater. The importance of obtaining good geotechnical information prior to designing the structure cannot be overemphasized. A geotechnical engineer should be consulted, and the following information and recommendations should be solicited.

- Safe construction slopes for excavation.
- Design high groundwater level, including seasonal and tidal effects.
- Recommended dewatering methods.
- Feasible foundation types.
- Characterization of site soils.
- Active, at-rest, and passive lateral soil pressures.
- Seismic loadings including dynamic soil loading (if UBC indicates site is in Zones 3 or 4).
- Corrosivity of soil or groundwater (water-soluble sulfates and chlorides).
- Soil unit weight and buoyant soil unit weight.

The determination of type of excavation should be based upon both cost and feasibility. Open cutting with a 2:1 slope (2 horizontal to 1 vertical) is the cheapest method where it can be used, but it is usually impractical in areas with high groundwater, or where there are adjacent structures or property line restrictions. These and other factors may require sheeted excavation with bracing or tie-backs.

Buoyancy can be a major factor in the determination of both foundation types and structural section. The groundwater level may be subject to seasonal or tidal variation, and the level at the time that the geotechnical investigation was performed may be inappropriate. The design high groundwater level should be used in designing the structure for buoyancy. The factors that can be used for resisting buoyancy are:

- The weight of the structure.
- The uplift resistance of piers or piles.
- The buoyant weight of soil over the toe of the footing.
- The soil shearing stress acting on the soil prism.

The latter three factors require evaluation by the geotechnical consultant. The proportion of soil mobilized by the footing to resist uplift can be determined by taking the buoyant weight of the prism of soil over the toe of the footing and adding the effect of the soil shearing stresses acting on this prism. A simpler and more common practice is to use the buoyant weight of the soil contained in an envelope defined by the toe of the footing and a 30-degree projection (from the vertical) to the ground surface. When conventional spread footings are used, it is usually more economical to generate uplift resistance by extending the toe of the footing than to thicken the walls or slab. Depending upon the confidence in the design high groundwater level, a factor of safety of 1.25 to 1.5 should be used for buoyancy.

A mat foundation is usually the preferred foundation type unless conditions exist that would preclude its use or make the mat foundation cost-prohibitive. Unstable soils, expected high settlements, expansive soils, or high groundwater with a limited footprint are all conditions that may lead to consideration of alternative schemes such as:

- Piers
- Piles
- Caisson
- Slurry walls

A caisson is a heavy structure consisting of the outer walls of the pumping station built above ground and sunk under its own weight, often with the aid of water jets. By excavating soil from the inside of the caisson, a controlled soil failure occurs under the bottom "cutting edge" accompanied by a subsidence of the caisson. After sinking the structure to the required depth, a concrete floor is poured to form the wet well and pump room floor. A caisson can be an economical pumping station design in certain soil conditions or where adjacent structures or property lines are a problem. However, this type of construction is somewhat inflexible for layout, and it must allow for some larger-than-normal tolerances on plumbness.

Slurry walls are short, deep, separated ditches excavated through a slurry of bentonite to support the earth walls. A cage of reinforcing bars can be lowered through the bentonite and concrete can be placed in the slurry ditch by tremie. After concrete has filled the first set of ditches, the intervening spaces are excavated and filled with reinforced concrete in the same manner, after which the enclosed earth can be excavated. It is usually best to allow the contractor to choose either caisson or slurry wall.

Design Criteria and Analysis

Loads

The following loads should be considered in the design of pumping stations:

1. *Lateral earth*. Either at-rest or active lateral earth loads should be applied to the below-grade walls of the structure. Values and criteria for determining which to use should be obtained from the geotechnical consultant.
2. *Hydrostatic*. Lateral loads from water contained in the wet well and/or from groundwater.
3. *Seismic*. In addition to seismic forces applied to the above-ground structure (dictated by UBC), hydrodynamic and dynamic soil lateral loading should be evaluated for structures in Seismic Zones 3 or 4. Dynamic soil loading should be obtained from the geotechnical consultant. Hydrodynamic loading should be calculated as described in Chapter 6 and Appendix F of *Nuclear Reactors and Earthquakes* [8].
4. *Buoyancy*. The buoyant forces created from the volume displaced by the portion of the structure below high groundwater (see subsection "Geotechnical Considerations").
5. *Surcharge*. If traffic or construction vehicles are anticipated adjacent to the below-ground structure, the associated lateral loads should be applied to the walls.
6. *Floor live load*. Such loading is usually higher than that specified by building codes. Depending on the intended activities, a live load between $4.8\,\mathrm{kN/m^2}$ ($100\,\mathrm{lb/ft^2}$) and four times as much should be used. Operation and maintenance activities involving equipment break-down, replacement, or use of portable gantry cranes can easily generate the higher loads.
7. *Equipment*. Most large pumps and motors generate dynamic or vibration loading in addition to their own dead weight. These dynamic loadings must be carefully evaluated to ensure that no unforeseen amplification occurs due to resonant frequencies. In general, structural natural frequencies must be at least 25% above equipment excitation frequencies.

These loads must be combined so that the result is a conservative design in which both the construction sequencing and the potential future conditions are recognized. The following design load combinations are suggested:

- Load combination I: full lateral earth + surcharge + hydrostatic high groundwater + dynamic soil + floor dead load. This loading condition should produce maximum negative vertical moment at the base of the walls and maximum positive vertical moment near the midpoint of the walls.
- Load combination II: same as above + floor live load. This loading should produce maximum negative vertical moment at the top of the wall as well as maximum negative moment in the floor.
- Load combination III: one-half lateral earth + low lateral groundwater + floor dead load + floor live load. This loading should produce the maximum positive floor moment. Using half of the lateral earth load is rational because the development of full design earth pressures may take a long time or may never occur at all.

- Load combination IV: no lateral earth pressure but full hydrostatic loading from the wet well. Such a loading should produce maximum positive and negative wall moments in the opposite faces to that produced by soil and groundwater loading. This load combination is meant to account for the hydrotest of the wet well prior to backfilling the excavation. Designers should determine whether this test can be performed with or without the top slab of the wet well and note any restriction accordingly on the drawings.

Wall Analysis

Pumping stations are often quite deep and the loading of the below-grade walls is much greater than in conventional buildings. With these high loadings, one way to limit wall thickness is to recognize the potential for two-way action (wall spanning both horizontally and vertically). If the aspect ratio of the below-grade wall is 3 to 1 or less, there is probably an economic advantage in designing it as a two-way wall.

Even if walls are designed as two-way structures, shear instead of flexure can often become the controlling parameter for determining wall thickness because walls are not traditionally reinforced for shear. With deep walls, especially in high groundwater, this type of design can result in extraordinarily thick walls. If, without reinforcing, shear criteria results in a wall thickness that is over 25% thicker than is required for flexure, shear ties in the walls should be considered.

There are several ways that walls can be analyzed as two-way structures. One way is to use a structural analysis computer program that permits element loading perpendicular to the element face. If this program is not available, refer to "Rectangular Concrete Tanks" [9], Moody [10], *Nuclear Reactors and Earthquakes* [8], and Housner [11, 12].

Engineers should pay special attention to the support conditions at the wall perimeter. As discussed above, the floor slab may not be in place at the time the structure is backfilled or hydrotested, so there may be a free condition on the top of the wall for some loading conditions. Additionally, caution should be used in assuming fixed conditions where rotation at the support could occur and result in underestimation of the mid-height, positive moment.

Dynamic Analyses of Buildings

The presence of rotating machinery requires an analysis of the effects of dynamic loading or vibration on buildings—especially of equipment supported by elevated floors. The principal focus of this analysis is to determine the natural frequencies of the structural support system. The lowest fundamental natural frequency must be compared with the excitation frequency of the machinery to guard against the potential for resonance. As a rule, resonance is unlikely if the fundamental natural frequency falls outside a range of 0.5 to 1.5 times the operating frequency of the equipment.

Most finite element structural analysis programs now have the capability of calculating the natural frequencies and corresponding mode shapes of the structural support system. These programs can be used to find the potential resonant conditions or to aid in detuning the structure to avoid resonance. Avoiding resonance problems by addressing these issues during design is considerably easier than trying to solve them once they have arisen in a constructed facility. An extensive discussion of vibration theory is presented in Chapter 22.

ACI 350 Code

The use of either working stress design or a modified strength design is allowed by ACI 350. An additional "service factor" by which the normal load factors in ACI 318 are multiplied is used in the modified strength design. The result is a larger, more heavily reinforced section than if designed per ACI 318. However, the modified strength design usually results in a somewhat more economical section (thinner with less reinforcement) than one designed using working stress.

ACI 350 also prescribes an additional 12-mm ($\frac{1}{2}$-in.) concrete cover over the reinforcing steel because it is recognized that these structures function in a potentially aggressive environment.

Crack control and serviceability (receiving greater attention in ACI 350) is accomplished by reducing the value of the reinforcing distribution factor, Z, from a maximum of 25 kN/mm (145 k/in.) for exteriors to a maximum of 20 kN/mm (115 k/in.) for a normal sanitary exposure—even down to 17 kN/mm (95 k/in.) for severe sanitary exposure. The practical effect is that, for a given required area of steel, smaller reinforcing bars at closer spacing are required. In SI units, for example, if the required area of steel in a flexural section is $37.5\,\text{cm}^2/\text{m}$, the use of number 35M bars at 265-mm spacing would satisfy the strength requirement, but d_c is 71 mm, so $Z = 22.2$, a number that exceeds the maximum of 20. However, for number 30M bars at 180-mm spacing, d_c is

68 mm, so $Z = 19.0$ and, therefore, the second choice for reinforcement satisfies both provisions. In U.S. customary units, the required area of steel is 1.78 in^2/ft, and although number 10 bars at 8.5-in. spacing satisfies the strength requirement, d_c is 2.72 in., and $Z = 118.4$—more than 115. Number 9 bars at 6.5-in. spacing with d_c at 2.62 in. yield a Z of 103, which satisfies both requirements.

One other modification in ACI 350 aimed at crack control is a significant increase in minimum temperature and shrinkage reinforcing. In ACI 350 the minimum reinforcing is correlated to the spacing of movement joints by allowing less reinforcing where joints are spaced more closely. The primary effect of this provision would occur in sections where flexure is not a controlling parameter. As shown in Figure 25-3, even with movement joints at a 9-m (30-ft) spacing, 50% more reinforcing steel is required than in nonhydraulic structures.

Concrete Mix

The concrete mix design for any hydraulic structure subject to the same environment as in pumping stations should be modified over that specified for building construction. Design 28-day compressive strengths should be 28,000 kPa (4000 $lb/in.^2$) minimum for concrete exposed to freezing and thawing cycles and 24,000 kPa (3500 $lb/in.^2$) minimum for concrete not exposed to these cycles. Again, the goal of these modifications is to produce a more durable and watertight structure.

Cement

In structures exposed to wastewater or wastewater effluent, the cement should have a C_3A content of less than 8%. Where possible, therefore, ASTM C 150, Types II or V Portland cements should be used. In some parts of the country, requiring Type II or V cement can significantly increase the concrete cost. Substituting a pozzolan, ASTM C618 (fly ash), for a portion of the cement so that the total C_3A content does not exceed 5% can offer similar sulfate resistance with Type I cement. The pozzolan should not exceed 20% by weight of the cement plus pozzolan.

Engineers should be aware that the use of Type V cement or the addition of fly ash frequently suppresses the early strength gain of concrete and is therefore not always popular with contractors. The long-term concrete strength is not affected. Additionally, a fly ash mix can sometimes be more difficult to trowel finish than a normal cement-only mix. These measures should be used only where they are warranted for sulfate resistance, and other mix designs should be allowed elsewhere.

Admixtures

An air-entraining admixture conforming to ASTM C260 should always be specified. Additionally, it is frequently desirable to use a water-reducing admixture and/or a superplasticizer to maintain a workable mix with the reduced water-cement ratio required in hydraulic structures. These admixtures should conform to ASTM C494.

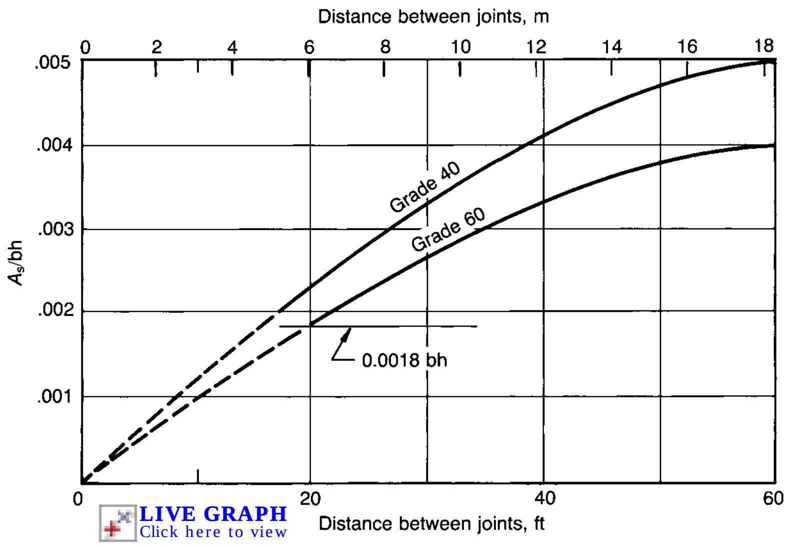


Figure 25-3. Wall shrinkage reinforcement.

Because of the concern for watertightness, a water reducer-retarder is sometimes specified to reduce the potential for a cold joint during a large concrete pour. However, the retarding effect can become a problem for contractors during cold weather—air temperatures of less than 4°C (40°F). A concrete mix using a retarder with set times of a couple of hours in the summer can increase to 8 to 10 hours on cold days. When temperature becomes a problem, allow for the use of an alternative mix with the water reducer but without the retarder.

Mix Proportioning

The two most important parameters for producing watertight, durable concrete are the cement content and the water-cement ratio.

A minimum cement content should always be specified for concrete used in hydraulic structures. Guidelines for minimum cement content are given in ACI 350. These cement contents frequently produce concrete strengths significantly exceeding the specified 24,000 to 28,000 kPa (3500 to 4000 lb/in.^{2}). Bear in mind, however, that the high cement content is not only for strength but also helps produce a denser, more chemically resistant concrete.

Controlling the water-cement ratio is paramount in producing watertight structures. Generally, the less water used, the less shrinkage cracking will occur in the cured concrete structure. The maximum water-cement ratio should not exceed 0.45. Water-cement ratios of 0.43 to 0.38 are not uncommon when water-reducing admixtures or superplasticizers are used. When pozzolans are used, the water-cement ratio should be calculated as the water weight divided by the cement plus pozzolan weight.

It is desirable to use the largest aggregate size practical—up to 38 mm (1.5 in.) for concrete in hydraulic structures. Large-sized aggregate also helps to reduce shrinkage. As aggregate size decreases, the cement content should increase and the water-cement ratio should decrease. The maximum size of aggregate allowed should be determined by the following:

- One-fifth the thickness of the wall, or
- Three-fourths the clear spacing between reinforcing bars, or
- One-third the depth of the slab.

Watertightness

In addition to the modified concrete design and mix design, other elements are required to produce a watertight structure. These are discussed in this section.

Waterstops

Waterstops (see Figure 25-4) are often a troublesome source of problems because an amazing number of contractors and workers do not realize that, to be effective at all, waterstops must be fully continuous with all splices carefully welded and not simply lapped. Corner splices and intersections should be shop-made. Waterstops between floors and walls should not be placed in a groove but in an upturned key as shown in Figure 25-4a. Waterstops should never be permitted to be bent out of shape by reinforcement. Waterstops can be made of rubber dumbbells, ribbed plastic, or steel plate.

In addition to the traditional rubber, plastic, or steel strip-type waterstops, there are numerous "second generation" waterstops available. Rather than being a strip that is cast into the first and then the adjacent pour, they are usually placed on the face of the joint between pours by adhesive or mechanical fastening. They rely on either adhesion to the concrete or expansive compression to produce a watertight joint. These types of joints are slowly gaining acceptance. Their advantages are:

- They are often the only practical waterstop for tying into existing structures with a watertight joint.
- Some have a lower installed cost than traditional strip waterstops.

Their disadvantages are:

- Those that rely solely upon adhesion may leak if any subsequent movement occurs in the joint.
- Those that rely on their expansive properties must be confined adequately by the concrete. At least 3 in. of cover must be provided or the expansion of the waterstop may break the concrete.
- The adhesives used to attach the waterstop to the first pour face tend to perform poorly in cold or wet weather.
- Some have a *higher* installed cost than traditional strip waterstops.

Joints

As discussed above, location and spacing of construction and expansion joints affects the amount of cracking that occurs. The locations of these joints should either be shown on the plans or parameters for their placement should be given in the specifications. Typical details are shown in Figures 25-4 and 25-5.

Walls exceeding 15 m (50 ft) in length should have construction joints spaced at no more than 12 m (40 ft). It is desirable to wait at least five days be-

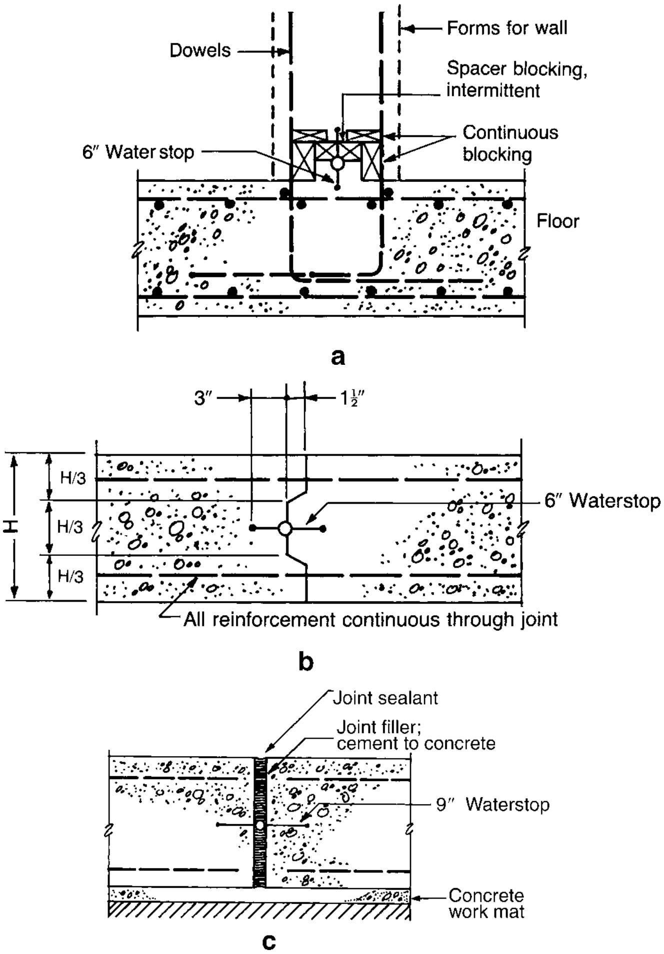


Figure 25-4. Details of construction joints with waterstops. (a) Upturned key; (b) wall joint; (c) expansion joint in a base slab.

tween pouring adjacent panels. This waiting period does not delay progress unduly if alternating panels are poured. Walls or slabs longer than about 37 m (120 ft) should have expansion joints spaced no more than 31 m (100 ft) apart.

Slabs should be cast in panels not to exceed 12 m (40 ft) in length or 95 m^2 (1000 ft^2) in area. Furthermore, the panels should be cast in a checkerboard pattern with at least three days between adjacent panel pours. An exception to these rules are slabs that can be cured by water-ponding. Good success has been obtained with very large monolithic pours cured by ponding water at least 50 to 75 mm (2 to 3 in.) deep for at least six days. Shrinkage appears to be minimized by the unlimited supply of free water during hydration.

If concrete is placed from a height, a tremie should be used to limit the free drop to 1.5 m (5 ft) maximum to reduce the potential (1) for segregation of the concrete, and (2) for dislodging waterstops or other inserts. Pumping, however, is preferable to placing concrete by tremie because the result is likely to be more uniform, with fewer rock pockets.

Exterior Coatings for Groundwater Exclusion

Exterior coatings are used in pumping stations for two reasons: (1) corrosion protection, and (2) waterproofing for keeping interior walls dry. Many of the corrosion coatings can serve both purposes. See Section 25-11 for the application of coating and linings inside the wet well.

The three main types of waterproof coatings are:

- Sprayed, brushed, or rolled-on coatings
- Membrane systems
- Cementitious crystalline waterproofing.

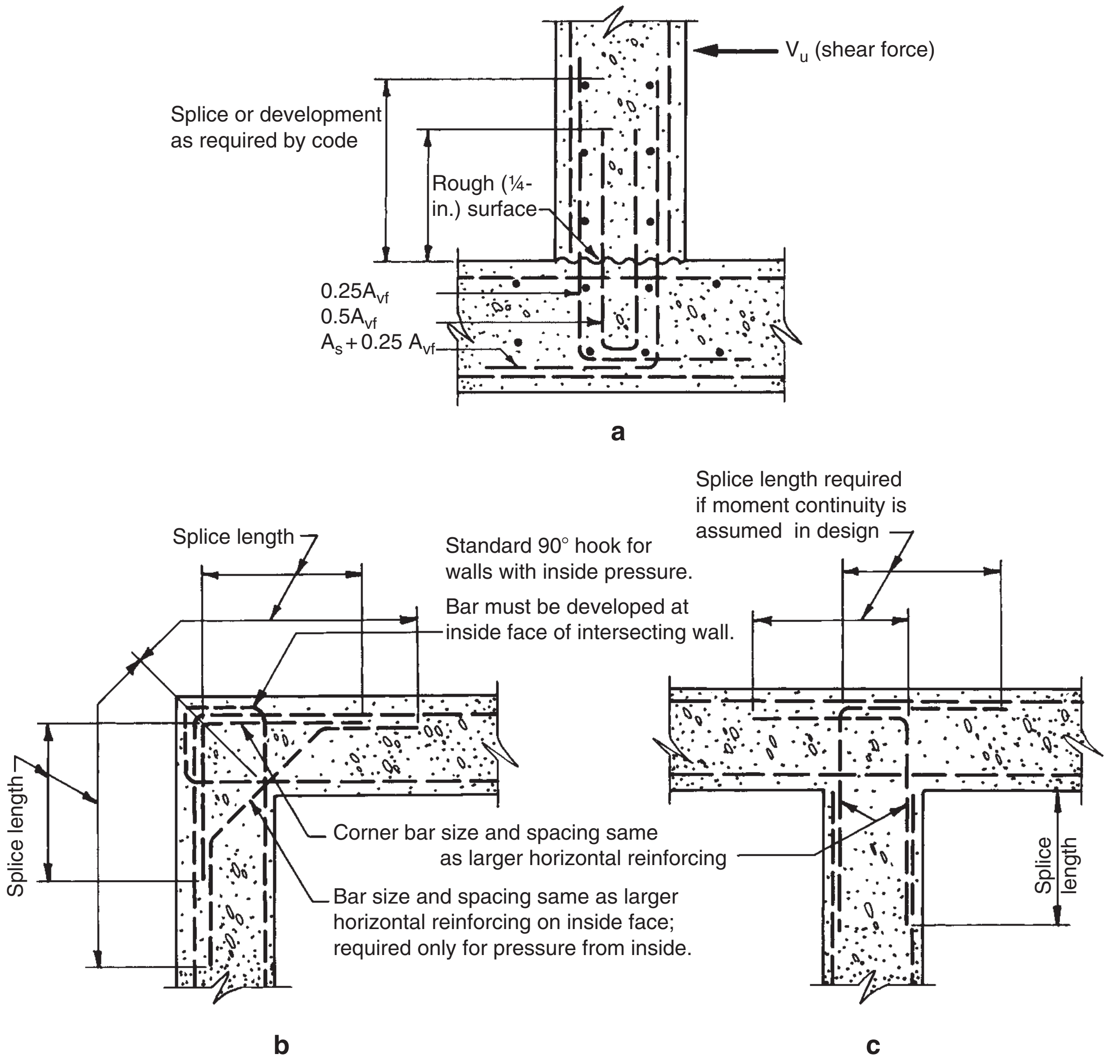


Figure 25-5. Concrete joint details. (a) Shear friction joint; (b) horizontal corner bars for walls. $V_u \leq \phi V_R \leq 0.2f'_c A_c \leq 800A_c$; $\phi = 0.85$ for shear tension; $V_n = A_{vf} f_y \mu$; $\mu = 1.0$ for rough surfaces.

With new regulations on volatile organic compounds, the sprayed or rolled-on coating field is changing rapidly. Spray-on, coal-tar coating has traditionally been the low-cost damp-proofing used where groundwater was either not present or only intermittently present. The new higher-solids epoxies that have taken their place are of higher quality at higher cost. Unless there is a potential for corrosion, other methods for waterproofing are desirable.

Cementitious crystalline waterproofing is sprayed or brushed onto new concrete prior to complete curing. It reacts with free water and forms a crystalline matrix that seals the pores and capillaries of the concrete. These systems perform quite well in most conditions at relatively low cost.

Membrane systems consist of sheets or panels that are applied to the concrete surface to act as an impermeable barrier for groundwater. Some systems actually employ clay panels that swell with contact to water providing an impermeable barrier. In general, membrane systems are more expensive but more effective than crystalline waterproofing.

Details

This section contains miscellaneous aspects of details common to pumping stations. As with any facet of structural design, there are many variations of these basic details, and designers should use only those that are appropriate for the specific application.

Detailing Reinforcement

One of the more important considerations in designing reinforcing at corners and other intersections is maintaining adequate clearance between bars to permit concrete placement that produces a watertight structure. Honeycombing or rock pockets promotes leakage. Vertical construction joints should usually be located at least 1 to 1.5 m (3 to 5 ft) from corners to allow good bar placement without interference between waterstops and corner reinforcing. Splice and development lengths from ACI 318 are applicable.

The reinforcing for walls designed as two-way elements must reflect the support conditions assumed at the corners. Support conditions are especially critical for walls resisting inside pressure because the interior horizontal reinforcing must be developed at the face of the intersection wall. See "Rectangular Concrete Tanks" [9] and Figure 25-5 for typical corner reinforcing at intersecting walls.

Anchors

Anchoring mechanical equipment to concrete is one of the most troublesome tasks in construction, and it requires careful detailing and superior inspection during construction to avoid problems. Nowhere else are there greater rewards for care or costlier penalties for carelessness. The environment for anchorages is often hostile due to submergence, high humidity, and the corrosiveness of wastewater and some other waters. Generally, it is more economical to use stainless steel or appropriate coatings on mild steel than to oversize the material to allow for rust and corrosion. The labor of replacement far exceeds the cost of prevention. Anchors are compared in Table 25-12. Refer to ACI's *Guide to the Design of Anchor Bolts and Other Steel Embedments* [13] for design procedures.

Table 25-12. Anchorages

Type	Advantages	Disadvantages
Physical embedment of equipment	Justified only for pipe sleeves in walls	Cannot be realigned nor replaced Use only if unavoidable
Embedded bars with anchor bolts	Excellent anchorage—the best Can be stainless steel. See Figure 12-21	Costly Must be accurately located Field welds must be expertly made Difficult to realign or replace
Embedded bolts	Slight realignment possible when set in sleeves Can use stainless steel or other corrosion-resistant materials	Must be precisely located in plan and elevation. Use templates
Bolts in holes through thin slabs	Good for thin slabs such as roofs Can develop full strength of bolt	Usable only if there is access to both sides of the slab Protruding bolt head may be unacceptable
Bolts with expansion shields	Inexpensive Suitable for concrete already cast	Suitable only for light loads and no vibration Unreliable for tension loads
Bolts set with two-component system (glue) in drilled holes	Suitable for concrete already cast Easily located and easily changed Reliability of two-component systems is excellent Good in tension and shear Inexpensive due to ease of installation Stainless steel (or other material) is suitable	Drilled hole may strike reinforcement Hole must be (a) thoroughly cleaned of all dust and (b) dry; otherwise, failure is likely. Some experienced engineers do not trust these fastenings at all because of the dependence upon responsible workmanship If used for critical service, bury bolts deeper than required by manufacturer and specify proof tensile tests for, say, 10% of the bolts

Adjacent Structures

There are often ancillary structures on shallow foundations adjacent to the pumping station. They might include small generator rooms, hydropneumatic tanks, or valve vaults. Their foundations should be designed to accommodate the mechanical or piping connections between the two structures. The pumping station is usually deep, whereas the adjacent structures usually rest upon backfill, thus creating a perfect scenario for differential settlement. Consider either: (1) designing flexible mechanical or piping connections between the structures or (2) supporting the ancillary structure with the pumping station itself by using, for example, a cantilevered counterfort.

Future Expansion

When designing pumping stations, always consider the likelihood of future expansion and discuss the potential for increasing the size of pumping equipment with the owner. Provisions can be made in the original design that greatly simplify expansion.

Motor or pump floor systems can be readily designed for greater future loads, whereas retrofitting an existing marginal support system for higher loading can be costly and can disrupt the operation of the pumping station.

If walls or slabs will be extended in a planned expansion, waterstops and dowel inserts to facilitate the new connection can be cast into the original structure and protected with lean concrete. Sliding can be a significant problem if the expansion involves excavation of soil at the wall face while the remaining structure remains backfilled. A shear key (like a retaining wall) cast below the floor prevents sliding. An alternative is to excavate both ends of the below-grade structure to equalize the soil loadings.

25-12. Concrete Protection: Coatings and Linings

Concrete structures exposed to the atmosphere of wastewater, such as the area above low water in sewers and wet wells, are attacked and corroded by sulfuric acid generated by bacteria. The mechanism is described in Section 4-7, subsection Lining. The extent of attack is indicated by the pH of the surface. Surface scrapings of new concrete in distilled water yield a pH range of 11 to 13. As concrete weathers, calcium hydroxide is converted to calcium carbonate, which then slowly dissolves to calcium bicarbonate. Carbon dioxide in the atmosphere can lower the pH to less than 7. Sulfate-reducing bacteria in stale wastewater reduce the sulfate ion in the water to hydrogen sulfide, and sulfide-oxidizing bacteria on the walls above the water convert the hydrogen sulfide to sulfuric acid, which dissolves components in the Portland cement. The surface pH can be as low as 1, but any pH of 4 or less indicates active corrosion. A protective covering of inert and impenetrable material is required for longevity. It is better and far cheaper to apply such protection to new structures than to repair existing ones.

The County Sanitation Districts of Los Angeles [14, 15] have tested a wide variety of protective systems for many years. Systems are screened by: (1) allowing a maker or supplier to coat or line a short, vertical concrete pipe with a product, (2) partly filling the pipe with 10% sulfuric acid, and (3) observing the effects after a long period of time. Products that pass this test are then tested under actual field conditions.

Inspections of field applications by the County Sanitation Districts of Los Angeles indicate there are no fail-safe coatings, even though some (polyester mortar, polyurea, and a sulfur concrete) have withstood the 10% sulfuric acid test well. The only system that has a long-term (more than four decades) *history* of protection in a corrosive environment is formed-in-place (ribbed) PVC liners, but proper welding of the seams is critical for success. Combinations of both coating (polyurethane mastic) and PVC lining have a shorter history of success. Mastic with polyethylene linings have failed in bond. Evaluations are continuing, and it would be premature to draw irrefutable conclusions yet.

In recent accelerated testing work sponsored by Tnemec Company [16], Nixon and Briand [17] found that permeability resistance to sewer gases and acids is more important than sulfuric acid resistance. In these tests, coated steel coupons were exposed to high concentrations of H_2S gas, acid baths, and CO_2 gas. The coupons were examined with electrochemical impedance spectroscopy prior to and following test cabinet exposure. Several products were found to have permeable films as evidenced by blistering-type failures. Accelerated testing is being used to develop better coating formulations for biogenic sulfide corrosion exposure. This laboratory test program is continuing and is supported by a comprehensive field testing program conducted by Corrosion Probe [18] at the Deer Island Treatment Plant for the Massachusetts Water Resources Authority. More information on the field findings were published in 2006.

Concrete Pipe

See Section 4-7 for a discussion of protection for concrete pipe.

Wet Wells

Wet wells are so costly that they should be protected above LWL even though such protection is initially expensive. It is much less expensive, however, to line or coat new construction than to rehabilitate an older wet well. Three imperatives for achieving long-term (50 years) protection are: (1) adequate surface preparation, (2) *skillful* application of either coatings or linings, and (3) *competent* inspection during application.

Waterblast new concrete at 55,000 to 69,000 kPa (8,000 to 10,000 lb/in.^{2}) to remove laitance, open the surface, and remove all debris, form oil, and any aggregate particles not tightly bonded. Allow the concrete to dry sufficiently so that coatings can penetrate into the matrix for a strong bond. (Coatings applied hot tend to enhance migration of moisture to the surface.) After drying, sandblast the surface in a pattern sweep to expose a new fresh surface. The waterblast can be omitted in favor of a vigorous sandblasting.

The general contractor should never be allowed to choose the applicator because the lowest bid (ending in the poorest workmanship and shortest life) will be accepted. Instead, prequalify bidders on the basis of reputation and successful projects. Obtain the services of consultants who specialize in corrosion prevention in wastewater structures and can assist in prequalifying applicators. The corrosion consultant can either supply competent inspectors or name some of those qualified.

Coatings

Coatings are brushed, troweled, or sprayed on wet well walls. Coatings have received bad publicity because the older synthetics, such as coal tar epoxy, did not protect the concrete for more than a few years. The newer epoxies, such as Tnemec Series 434 [16] among others, do afford long-term protection, presumably for 50 years or more. Epoxy is hard and brittle, however, so it must not be used across construction joints where the slightest movement cracks the coating. Epoxy coatings should be stopped short of such joints, and the area between should be coated with polyurethane. Some polyurethanes can be stretched 100% without losing integrity.

Coatings have two advantages over linings: (1) odd corners or shapes are no problem at all, and (2) they are less expensive. At 2005 prices, a rough estimate for coating a wet well would be 76 to 130/m^2 (7 to 12/ft^2).

Linings

Linings are of two types: ribbed (such as Ameron T-Lock®) and bonded (such as Linabond®). Ribbed liners are placed against the forms, stapled to them, and the forms themselves are fastened together with snap ties. When the forms are removed, there are holes in the liner from both snap ties and staples. If there is any hole in a ribbed lining, H_2S and sulfur-oxidizing bacteria can seep along the full length of the ribbed section and corrode the concrete beneath. Every hole must be sealed with a PVC patch perfectly welded to the parent material. The high quality of workmanship required cannot be assured today. Another problem is the joints, which are difficult to fabricate perfectly gas-tight. As a consequence, there are many failures of ribbed liners. For wet wells, bonded liners are better, although for concrete pipe, nothing surpasses ribbed liners.

Bonded liners are applied after the concrete has cured and the surface preparation of sandblasting, described above, is completed. In the Linabond® system, for example, a two-component primer is applied followed by trowelling a 1.5-mm (60-mil) layer of polyurethane mastic. An activator is applied to a sheet of PVC and allowed to cure for a few minutes. The sheet is then applied to the mastic, and air bubbles are rolled out. Joints are overlapped 100 mm (4 in.). There are two types of PVC sheets (semi-rigid and flexible), and for sheets that must be bent, specify the flexible type. Once again, corners present a difficulty, but premolded pieces of PVC are available. PVC is more impervious than epoxy, and bonded liners offer excellent protection for wet wells. The cost in 2005 is roughly 130 to 162/m^2 (12 to 15/ft^2) installed.

Rehabilitation

It is necessary to remove all damaged concrete to hard gray concrete with a surface pH of at least 7 by using a water blast at preferably 55,000 to 69,000 kPa (8000 to 10,000 lb/in.^{2}) pressure or by sandblasting. If there is a delay after waterblasting, a sweep sandblast is needed. The surface can then be rebuilt to its original contour with, for example, a polymer grout that is

chemically impervious to acid. With Linabond®, the polyurethane mastic is applied in a layer twice as thick as for new construction to allow for the wavy surface.

Other Products

There is a myriad of products for protecting concrete, and engineers should be alert for good, new ones. But be wary and very skeptical of sales claims because systems that are excellent in the laboratory may be disappointing in the field. Furthermore, the traditional low-bid approach for choosing a contractor is not conducive to good field work. However contractors are chosen, competent inspection is important in producing quality work.

It is best to investigate field installations that have been in place several years before specifying a "new" product.

25-13. Corrosion of Metals

A wide variety of metals, including carbon steel, ductile iron, and cast iron, as well as stainless steels and copper and aluminum alloys, are used for piping and its supports, and for gates, gate frames, structural elements, and related facilities in pumping stations. Their long-term corrosion-resistant performance is the main consideration in this section. See Chapters 4, 5, and 11 for corrosion resistance of pipelines, pumps, and valves.

Types of Corrosion

The term *corrosion* describes the oxidation process by which a native metal is converted to an oxide. The principal types of metallic corrosion are summarized in Table 25-13. Metallic corrosion can occur both slowly and rapidly. When corrosion occurs slowly, the metal on the surface is typically converted to an oxide of the metal and remains on the surface, usually as a protective layer. If the corrosion is rapid, as often occurs in galvanic and in pit and crevice corrosion, the oxidized metal *can* remain on the surface, but more typically is released to the solution in the form of a metallic ion. As reported in Table 25-13, the characteristics of water are extremely important in the corrosion of metals. As a consequence, it is difficult to generalize as to which metal will be most suitable in any given application, according to Crittenden et al. [19]. Because of the variety of issues that affect the kind and rate of corrosion, it is wise to consult a corrosion expert for critical applications.

Because dissimilar metals are often used in pumping station applications, it is useful to review the formation of galvanic cells (also known as COMP cells). A galvanic cell is formed when two metals with dissimilar COMP come in contact with each other in the presence of an electrolyte (water in pumping station applications.) When two metals such as iron, zinc, or aluminum come in contact, the metal that is higher on the electromotive series, as reported in Table 25-14, becomes the cathode and the other metal becomes the anode. Because of the potential difference between the metals, current flows from the higher- to the lower-voltage metal. At the anode, the current is released to the electrolyte and the metal undergoes an anodic reaction such as $Fe \rightarrow Fe^{2+} + 2e^{-}$ and, thus, corrodes. The arrangement of the selected metals in Table 25-14 is based on their presence in seawater and, as such, is only meant to illustrate the nature of the electromotive series. The specific galvanic arrangement of metals in contact with water depends on the chemical characteristics of the water.

Carbon Steel

Carbon steel, with the exception of steel mortar-lined conduits discussed at the end of this section, is not widely used in pump station design today except in some internal piping uses and as structural steel framing for associated buildings. Effective corrosion control measures include adequate ventilation to eliminate condensation, premium coatings and linings selected for both corrosion control, and erosion resistance. Carbon steel suffers from electrolytic corrosion in both immersion and in highly humid head spaces in pumping stations. When submerged in water, the corrosion of carbon steel is mostly driven by the amount of oxygen reaching the bare metal surface. As pH falls, the corrosion rate of carbon steel accelerates. Carbon steels immersed in water are also susceptible to galvanic corrosion, especially where the steel is connected electrically to stainless steel. Here, the carbon steel becomes anodic to the stainless steel and corrodes preferentially to protect the stainless steel.

The use of coated carbon steel in water immersion generally results in localized pitting corrosion of the steel wherever the coating is breached. Here, the area relationship between anodic and cathodic areas affects the corrosion rate. The bare metal at localized

Table 25-13. Principal Types of Corrosion of Metals Used in Pumping Stations. Adapted in part from Crittenden et al. [18]

Types of corrosion	Description/occurrence	Important factors
Concentration (crevice corrosion)	Crevice corrosion is a localized form of corrosion which occurs in stagnant microenvironments (e.g., shielded areas such as under gaskets, between nuts and bolts, under coatings which have disbonded)	Changes in local chemistry within crevice leading to depletion of inhibitor, depletion of oxygen, the creation of acid conditions, and the accumulation of aggressive ions (principally chloride)
Electrolytic	Corrosion occurs where stray direct electrical current from an outside source enters a pipe or other metal structure (either internally or externally) and then leaves to return to the source. In the process, metal is removed at the anode.	Strength of electrical current, internal and external conditions, water pH, conductivity, and velocity; the amount of oxygen reaching the bare metal surface
Erosion	Corrosion of a metal caused or accelerated by the motion of a corrosive fluid, especially one containing abrasive material. The two most common types are impingement and cavitation (see discussion in Section 10-4).	Velocity of fluid, presence of abrasive particulate matter such as grit in wastewater, presence of imperfections in metal surface, presence of other types of corrosion Very low pressure resulting in formation of vapor bubbles
Galvanic	Galvanic corrosion occurs where two dissimilar metals are placed in contact with each other	Difference in the electrical potential between the two metals, difference in the relative areas of the two metals, conductivity of water
Localized (pitting)	A small portion of the pipe surface becomes a permanent anode and the surrounding pipe serves as a cathode	Oxidizing potential of the water, the presence of aggressive ions (principally chloride), condition of metal surface in service (presence or absence of films), manufacturing defects (e.g., holidays)
Microbiologically induced (MIC)	Enhance corrosion kinetics by accelerating the rate of redox reactions, most commonly in conjunction with the other types of corrosion cited in this table	Bacteria and other microorganisms form micro-zones on the pipe surface which contain high acidity or concentrations of corrosive species
Stress corrosion cracking	Induced by tensile stress and a corrosive environment	Residual stresses caused by welding, improper alignment, repeated cycling coupled with a corrosive environment (principally the presence of chloride)
Uniform (metal conduits)	Metal conduits exposed to water react with constituents in the water to corrode or become oxidized uniformly	Oxidizing potential of the water, the presence of aggressive ions (principally chloride)

pinholes or "holidays" (small anomalies in the coating resulting from damage during shipping or installation) becomes anodic and represents a small amount of area relative to the rest of the coated (cathodic) surfaces. Hence, the anodic or corroding sites corrode more aggressively. Although galvanized carbon steel can be used successfully for hard-to-recoat, immersed equipment, there is considerable risk if the water chemistry is not just right. Note, however, that mortar-lined carbon steel pipe has proven to be relatively long-lived.

Galvanized Steel

Galvanized steel pipe has a hot-dipped coating of zinc that is composed of five distinct layers: (1) the base steel pipe, (2) the γ layer (75% Zn, 25% Fe), (3) the δ layer (90% Zn, 10% Fe), (4) the ζ layer (94% Zn, 6% Fe), and (5) the η layer (~99% Zn). Corrosion of zinc in water is mostly related to the pH of the water. Other factors include the specific impurities present in the water, time of exposure, temperature, and motion or agitation. The corrosion resistance of zinc in

Table 25-14. Galvanic Series for Metals Immersed in Seawater at 25°C (77°F)[a]

Metal	Corrosion potential,[b] V
Anodic or active end of series, most likely to corrode	
Magnesium and magnesium alloys	−1.60 to −1.63
Zinc	−0.98 to −1.03
Aluminum alloys	−0,76 to −1,00
Mild steel (~96% Fe; 0.2% C; < 1.6% Mn; < 0.6% Si; < 0.6% Cu)	−0,60 to −0,71
Wrought iron (~96.6% Fe; 2.8% slag; 0.06% C; 0.18% P; 0.04% S; 0.2% Si; 0.06% Mn;)	−0.60 to −0.71
Cast iron (~93% Fe; 3.5% C; 2.4% Si; 0.05% S; 0.5% P; 0.22% Mn)	−0.60 to −0.71
Type 410 (11.5–13.5% Cr; 1% Mn; 1% Si; 15% max. C) stainless steel—active in water	−0.46 to −0.58
Type 304 (18–20% Cr; 8–12% Ni; 0.08% max. C) stainless steel—active in water	−0.46 to −0.58
Type 316 (18–18% Cr; 10–14% Ni; 2–3% Mo; 0.08% max. C) stainless steel—active in water	−0.43 to −0.54
Inconel (78% Ni; 13.5% Cr; 6% Fe)—active in water	−0.35 to −0.46
Aluminum bronze (92% Cu; 8% Al)	−0.31 to −0.42
Yellow brass (65% Cu; 35% Zn)	−0.30 to −0.40
Red brass (85% Cu; 15% Zn)	−0.30 to −0.40
Tin	−0.31 to −0.33
Copper	−0.30 to −0.57
Lead-tin solder (50%–50%)	−0.28 to −0.37
Admiralty brass (71% Cu; 28% Zn; 1% Sn)	−0.28 to −0.36
Aluminum brass (76% Cu; 22% Zn; 2% Al)	−0.28 to −0.36
Manganese bronze (58.5% Cu; 39% Zn; 1% Sn; 1% Fe; 0.3% Mn)	−0.27 to −0.34
Silicon bronze (96% Cu; 0.80% Fe; 1.50% Zn; 2% Si; 0.75% Mn; 1.60% Sn)	−0.26 to −0.29
Type 410 (13% Cr) stainless steel—passive in water	−0.26 to −0.35
Lead	−0.19 to −0.25
Nickel 200	−0.10 to −0.20
Inconel (78% Ni; 13.5% Cr; 6% Fe)—passive in water	−0.14 to −0.17
Monel 400 (70% Ni; 30% Cu)	−0.04 to −0.14
Type 304 (18–8) stainless steel—passive in water	−0.05 to −0.10
Type 316 (18–8, 3% Mo) stainless steel—passive in water	0.00 to −0.10
Titanium	−0.05 to +0.06
Hastelloy C (~66% Ni; 15–17% Mo; 14.5–16.5% Cr; 2.5% Co)	−0.03 to +0.09
Platinum	+0.19 to +0.25
Graphite	+0.20 to +0.30
Cathodic or passive end of series, least likely to corrode	

[a]Adapted from www.bodrum-bodrum.com/vorteks/arsenal/articles/galvanic series.htm.

[b]The galvanic corrosion potential is reported relative to a standard hydrogen electrode (SHE), assigned a value of 0.0 V at 25°C (77°F). The corrosion potential values can be used to determine which of two metals in contact will become the cathode and anode.

Note: Active corrosion is accompanied by the formation of non-protective, soluble corrosion products. Passive behavior is accompanied by the formation of a protective, insoluble corrosion product film.

water depends largely on its initial ability to form a protective layer or film. For example, in distilled water, there is no chance of formation of a protective scale. Hence, the access of oxygen to the zinc surface is unimpeded and corrosion is quite severe.

The scale-forming capability of water depends, for the most part, on three factors: (1) pH, (2) total calcium content, and (3) total alkalinity. If the pH of the water is above or below where the water is in equilibrium with calcium carbonate ($CaCO_3$), the water will either deposit a scale in the form of a film or will dissolve the scale. The potential to deposit or dissolve a film can be estimated by means of Caldwell Lawrence diagrams [20]. While hardness and alkalinity are very important and harder waters are better than softer waters, other factors must also be considered. For instance, it is known empirically that waters high in free carbon dioxide are aggressive toward zinc. It has generally been shown that zinc performs well at pH values between 6 and 12, assuming no other corrosion contributors are present. Galvanized carbon steel in non-submerged applications must not be exposed to constantly wet, condensing conditions.

The corrosion resistance of zinc exposed to the atmosphere is due to the formation of an insoluble, basic zinc-carbonate film. Initially, when exposed to

the atmosphere (where not corrosive), zinc reacts with air to form zinc oxide [ZnO and later zinc hydroxide $Zn(OH)_2$]. Ultimately, the zinc in the outer coating will react with atmospheric CO_2 to form zinc carbonate (the most protective film). If the zinc surfaces are constantly wet and very slow to dry, the zinc will corrode. If the atmosphere is not especially aggressive, the zinc forms a white, powdery corrosion product called "white rust." It is voluminous and does not permit the formation of a tightly adhered oxide film. Although not necessarily a major corrosion concern, the formation of white rust does prevent the formation of a good barrier and keeps moisture present. As a consequence, because the zinc layer is wet more of the time, more corrosion occurs.

Zinc corrodes actively if the moisture present is acidic, as with exposure to H_2S. When carbon steel is electrically connected to an aluminum alloy in immersion service, aggressive corrosion of the aluminum occurs because the aluminum is electromotively less noble than the steel in the galvanic series (see Table 25-14). As an example, if bolts holding an aluminum gate to a concrete wall touch the reinforcing steel in the concrete, the aluminum will corrode in an attempt to protect the more noble carbon steel.

Cast and Ductile Irons

The use of cast iron and ductile iron components essentially follows the same rules as the use of carbon steel. In immersion applications, these iron-based (non-alloyed) metals corrode almost identically to carbon steel where oxygen is the main driver of corrosion. The same applies to these iron-based metals in corrosive, non-submerged exposure conditions. These metals corrode if constantly wet or if exposed to low pH conditions such as high H_2S concentration and sulfur-oxidizing bacteria. Hence, they must be coated and/or lined to resist corrosion. An exception involves the immersed use of high-nickel, austenitic (non-magnetic) cast irons which perform reasonably well in acidic vapor environments such as wet well head spaces. High-silicon cast irons are an even better choice.

Where relatively high concentrations of chlorides exist (such as in coastal collection systems having high infiltration rates of seawater or brackish water), the use of iron-based metals should be considered carefully along with the use of other materials. Cast iron alloyed with nickel (3–4%) improves its corrosion resistance in waters with chloride ranging from a few mg/L to seawater concentrations (about 19,000 mg/L). Nickel at about 3% also improves the erosion and cavitation resistance of cast iron. Nickel plus 1 to 3% chromium further improves resistance to corrosion. Generally, however, cast irons are coated for improved corrosion resistance.

Stainless Steels

The use of stainless steel (ss) in municipal water applications has grown substantially in recent years due to over 40 years of excellent historical performance. Common applications include slide gates, gate frames, and piping, as well as in pumps and fittings. The most common types of stainless steels are 304 and 316 austenitic ss. Austenitic ss typically contain 18% Cr and 8% Ni (see Table 25-14). Despite a very good performance record, stainless steels can be susceptible to several forms of corrosion, including pitting (most common), uniform corrosion, galvanic corrosion, crevice corrosion, and stress corrosion cracking, as well as other corrosion mechanisms (see Table 25-13).

Corrosion Resistance

The corrosion resistance of ss involves "passivity," a term used to describe the formation of a protective oxide film. Passivity occurs under certain conditions for particular exposure environments. The range of conditions in which passivity can be maintained is dependent on the exact environment and on the COMP of the stainless steel. If passivity is maintained under the right conditions, the rate of corrosion of stainless steels is very low. If the passive film is damaged and cannot be restored under given conditions, the stainless steel corrodes readily, usually in a localized manner by pitting or crevice corrosion.

Various factors are critical to passive film formation. For example, oxygen must be present for the formation of the passive film. Also, the surface of the stainless steel must be kept free of deposits by a flowing solution. If parts of the surface are covered by coatings, gaskets, interfaces with other stainless surfaces, or biofouling, oxygen-poor zones are formed at these covered or creviced locations. These zones or regions become anodic to the aerated exposed surfaces and localized corrosion occurs. This localized corrosion can be accelerated depending on other aspects of the environment, including conductivity of the solution or electrolyte, temperature, pH, anode to cathode area ratios, and other factors.

Stainless Steel Corrosion

Types 304 and 316 austenitic ss generally perform well in exposure to hydrogen sulfide and to the low concentrations of sulfuric acid (H_2SO_4) formed by sulfur-oxidizing bacteria in aerated head spaces. But those same grades of ss have varying passive film resistance to chloride and are therefore susceptible to crevice corrosion, because chloride ions are readily able to penetrate the passive film and cause pitting. Crevice corrosion occurs in 304 ss at chloride levels of less than 300 mg/L, and it occurs in 316 ss at chloride levels just below 1000 mg/L. These chloride levels can be present in coastal collection systems due to infiltration of seawater or brackish water.

If crevice corrosion is suspected in Type 316 stainless steel due to higher chloride concentrations, consider upgrading to Type 317. Its higher molybdenum content enhances its chloride pitting resistance. It performs well up to 3000 to 4000 mg/L chloride in neutral pH waters up to 32°C (90°F). If chloride levels approach that of seawater but velocities are high [above 1.5 m/s (5 ft/s)], Type 316 and 316L may perform adequately. If chlorides are at seawater levels and velocities are lower, stainless steel alloys, containing 6 to 7% molybdenum, such as AL6XN or 254 SMO (also known as 6Moly stainless steels), are excellent choices.

Types 304 and 316 stainless steels are more likely to develop corrosion in low-flow or stagnant conditions. Microbiologically induced corrosion (MIC) at welds or at the heat-affected zones of welds in stainless steels has been observed in wet wells where stagnant conditions and sludge buildup has occurred. Removal of heat tint scale left at welds on stainless steel is critical to preventing localized corrosion of stainless steels at or near welds. Also, avoiding the presence of crevices in stainless steel design and construction helps to reduce the occurrences of crevice corrosion and MIC. Another example of stainless steel corrosion in wastewater applications involves return activated sludge (RAS) piping. In this application, if scouring velocities are not achieved, severe pitting corrosion can occur in stainless steel piping due to the presence of sulfate-reducing bacteria under tubercle deposits. If low-flow conditions are unavoidable, MIC can be eliminated through the use of plastic pipe.

Immersed austenitic stainless steels are typically resistant to the normal chlorine concentrations encountered with the disinfection of water. Both 304L and 316L ss have very good corrosion resistance in water having up to 2 mg/L chlorine. If the chlorine concentration is higher (e.g., 3 to 5 mg/L near chemical feed stations), these same grades of stainless steel exhibit crevice corrosion. Consider using fiberglass-reinforced plastic (FRP), PVC, HDPE, or Hastelloy C-276 at these high chlorine concentrations.

Copper

Although copper has been used for instrumentation and utility piping in pumping stations, its use has largely been dropped following a history of significant corrosion problems due mostly to the poor resistance of copper to hydrogen sulfide in both moist or dry environments. Copper piping or tubing should be replaced with either plastic (PVC) or stainless steel.

Aluminum Alloys

Aluminum has been used extensively in pumping station applications for covers, gratings, and similar applications because of its high strength and relatively good fatigue resistance. Furthermore, aluminum and its alloys are corrosion-resistant to a wide range of exposure conditions as a result of the formation of a barrier oxide film tenaciously bonded to the metal surface. This film is readily restored when damaged under normal weathering conditions. The presence of the natural oxide film maintains the metal in a passive state (non-corroding) over a relatively wide range of exposure conditions. For example, aluminum alloys remain passive between a pH range of about 4.0 to 8.5. Above or below this pH range in a wetted environment, most aluminum alloys corrode. Corrosion at very high or low pH values can be uniform or can take the form of localized pitting corrosion.

Pitting corrosion of some aluminum alloys can also occur in the passive pH range between a pH of 4 and 8.5 if exposed to high chlorides in an aerated environment. Corrosion of aluminum alloys in municipal water immersion is typically related to being galvanically coupled to a more cathodic or more noble metal like carbon steel or copper or even stainless steel. Here, the aluminum becomes anodic and corrodes preferentially to protect the carbon steel or other metal. In aluminum to stainless steel couples, however, the stainless steel is usually easily polarized cathodically such that the corrosion current is so low that corrosion losses in the aluminum are very small.

In atmospheric exposures (head spaces) in wet wells, aluminum alloys perform well if the condensates to which they are exposed do not develop pH values of 4.0 or less. Low pH values are typically related to the concentration of H_2S. If H_2S levels

are relatively high and/or fairly constant, the aerobic bacteria *Thiobacillus* will metabolize the sulfides to form dilute H_2SO_4. If the wetted surfaces of the aluminum reach a pH of below 4.0, corrosion (generally pitting) occurs in the aluminum structures or components. As examples, aluminum gates in the open position often corrode actively in wet well head spaces if H_2S gas levels are high or constant. When immersed, aluminum gates and gate frames corrode galvanically when electrically continuous with carbon steel (like rebar). Therefore, the expected H_2S gas levels should be considered when determining whether to use aluminum alloys. H_2S gas levels can be reduced to very low concentrations by frequently cleaning the wet well and still further reduced by good ventilation. Fiberglass reinforced plastic decking or grating should be used in lieu of aluminum where such conditions may exist or could develop.

Metal Mortar-Lined Pipes

Both steel and ductile iron pipes are commonly furnished with cement mortar linings for corrosion protection. These linings are usually very long-lived (see Subsection "Linings" in Section 3-2). Unfortunately, the cement lining can deteriorate under the continual exposure to aggressive waters. The principal methods of deterioration involve (1) the solubilization and release of the free lime in the mortar under low-pH conditions, and (2) the chemical attack of the mortar by aggressive ions such as sulfate and chloride. Low-alkalinity waters enhance the above mechanisms, and very soft waters are also aggressive to cement mortar. High concentrations of sulfate (above 400 mg/L) can cause swelling which can also lead to the deterioration of the mortar coating. Where low-pH conditions and aggressive ions are present, mortar made with greater amounts of aluminum and iron has proven to be more resistant than conventional mortar mixes, but a conservative engineer would specify a plastic lining or plastic pipe. Always analyze the water before specifying the pipe lining.

25-14. Force Main Design

The force main is not part of a pumping station, but because the station system curve depends on the friction head as well as the static head, the discharge piping system cannot be ignored. Furthermore, the design of the force main affects water hammer or hydraulic transients, which, in turn, has an influence on the design of the pumping station.

Hydraulic Transients

Transient control receives far too little attention. Hence, there have been many dramatic disasters from the cracking of valves or pump casings to the rupture of pipes with flooding and, at times, loss of life. Pump start-up with a force main full of water causes surges that may become major for pumps with a high shutoff head. Broken mains, damaged check valves, distorted pump shafts, or a multitude of small leaks caused by loose joints and cracks resulting from repeated water hammer impacts are the result.

Transient analyses are usually required when the following conditions exist:

- The TDH is greater than 12 to 15 m (40 to 50 ft) and the flow rate exceeds about $110\,m^3/h$ (500 gal/min).
- Pipelines have high points or "knees" near the midpoint. Power failures can cause a partial vacuum at the knee that can result in column separation (with only water vapor in the pipe at the knee)—a problem to avoid at all costs.
- The static pressure differential for an in-line booster pump is more than 12 m (40 ft).

Transient analyses are not usually required if (1) the total TDH is less than 12 m (40 ft), (2) the flow rate is less than $23\,mm^3/h$ (100 gal/min), (3) the pipe velocity is 0.6 m/s (2 ft/s) or less, or (4) if the facility is a domestic water network that has many branches and loops to dissipate the transients. See Chapters 6 and 7 for more explanation.

Simplified transient analyses may be incorrect or misleading. Computer modeling is the most effective method, and the cost for an experienced consultant can range from about $1000 for a simple system to $10,000 or more for a complex problem. Some manufacturers and suppliers offer computer modeling as a free service, but as such modeling may be incomplete or biased, the user had better know how to review the assumptions and the results.

There are many methods for controlling transients, but none can be used indiscriminately. For example, the pipe profile dictates both the kind and placement of hardware. Only an expert should make the analysis and design the transient control system.

Size

Optimize the size of the pipe based on the life-cycle cost, including power and capital costs. A practical maximum velocity is about 2.4 m/s (8 ft/s). Higher flow results in greater headlosses and may result in

excessive water hammer. The lowest design velocity that should be used for raw wastewater is 0.6 m/s (2 ft/s) to keep grit moving, and a peak daily velocity of 1.1 m/s (3.5 ft/s) is desirable to resuspend settled solids. A lower minimum velocity of 0.5 m/s (1.6 ft/s) can be tolerated if a twice-daily velocity of 1.1 m/s (3.5 ft/s) is attained.

Friction Coefficient

Use the correct friction coefficients for pipes. Excessively rough friction factors, although conservative with respect to the carrying capacity of the pipe, are dangerous for the selection of motors and pumps. When determining the carrying capacity, a conservative practice is to use $C = 120$ for lined or plastic pipe (or $C = 100$ for unlined pipe), which conforms to Ten-State Standards; then redraw the system curve for $C = 145$. Make sure that the system can operate at both conditions or at any intermediate condition.

25-15. References

1. Davis, C. V. and K. E. Sorensen, *Handbook of Applied Hydraulics,* 3rd ed., McGraw-Hill, New York (1969), pp. 40–16.
2. ASCE Manual No. 37, *Design and Construction of Sanitary and Storm Sewers,* American Society of Civil Engineers, New York (1970).
3. Pankratz, T. M. *Screening Equipment Handbook for Industrial and Municipal Water and Wastewater Treatment*, 2nd. ed., Technomic Publishing Co., Inc., 851 New Holland Ave., Box 3535, Lancaster, PA 17604 (1995).
4. Maintenance Protection Systems, 43317 Moody Dixon Road, Prairieville, LA 70769, or Thortex America, 420 Babylon Road, Horsham, PA 19044, or Thortex America West, 600 S.E. Maritime Ave., Ste. 250, Vancouver, WA 98661, or Mountain West Thortex, 6969 N. 16th., Coeur d'Alene, ID 83814.
5. Monroe, P. C. "Pump baseplate installation and grouting," *Proceedings of the 5th International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, TX, (1988), pp. 117–128.
6. Hrivnak, S. J., "In-plant perspective: Eastman Kodak Company," *Pumps and Systems* (January 2000).
7. Piotrowski, J., "Understanding and using shaft-to-shaft alignment measurement systems," *Pumps and Systems* (April 1999).
8. *Nuclear Reactors and Earthquakes*, Technical Information Document 7024, Lockheed Aircraft Corp. (1963), Chapter 6 and Appendix F.
9. "Rectangular Concrete Tanks," *Information Sheet* ISOO3.03D, Portland Cement Association, Skokie, IL (1969 Rev. 1981).
10. Moody, W. T., "Moments and reactions for rectangular plates," *Engineering Monograph No. 27*, U.S. Bureau of Reclamation, Denver, CO (1960 Rev. 1963).
11. Housner, G. W., *Earthquake Pressures on Fluid Containers*, California Institute of Technology, Pasadena, CA (1954).
12. Housner, G. W., "The dynamic behavior of water tanks," Bulletin, Seismic Society of America, 53(2): (1963).
13. ACI, *Guide to the Design of Anchor Bolts and Other Steel Embedments*, AB-81, American Concrete Institute, Farmington Hills, MI (1981).
14. Redner, J., R. Hsi, and E. Esfandi, "Evaluating coatings for concrete wastewater facilities: An update," *Journal of Protective Coatings and Linings,* 11(12): 50–161 (December 1994).
15. Redner, J., R. Hsi, and E. Esfandi, "Evaluating coatings for concrete exposed to sulfide generation in wastewater treatment facilities, "*Journal of Protective Coatings and Linings,* 8(11): 48–56 (November 1991).
16. Tnemec Co., Inc., 6800 Corporate Drive, Kansas City, MO, 800-863-6321, www.tnemec.com.
17. Nixon, R. and R. Briand. "A novel analytical approach for evaluating protective coatings performance in wastewater environment," presented at *WEFTEC 2003, 76th Annual Conference & Exposition*, Water Environment Federation, Los Angeles, California (October 2003).
18. Corrosion Probe, Inc., 12 Industrial Park Road, Centerbrook, CT 06409.
19. Crittenden, J. C., R. R. Trussell, D. W. Hand, K. J. Howe, and G. Tchobanoglous, *Water Treatment, Principles and Design*, 2nd ed., Montgomery Watson Harza, John Wiley & Sons, Inc., Hoboken, NJ (2005).
20. Merrill, D. T., R. L. Sanks, and C. Spring, *Corrosion Control by Deposition of* $CaCO_3$ *Films. A Handbook of Practical Application and Instruction*, American Water Works Association, 6666 Quincy Ave., Denver, CO 80235 (1978).

Chapter 26

Pumping Station Design Examples

GARR M. JONES
GARY S. DODSON
THEODORE B. WHITON

CONTRIBUTORS
Roger Cronin
Philip A. Huff
Paul C. Leach
Ralph E. Marquiss
M. Steve Merrill
Earle C. Smith
Patricia A. Trager
William Wheeler

Chapter 12 contains details for selecting pumping equipment and installing it in properly designed wet wells with piping arrangements that allow for easy access in a minimum of space. The principles for the layout of the basic elements of wastewater and water pumping stations respectively are given by example in Chapters 17 and 18. These principles should be followed with an overall objective of keeping the installation as simple as possible. The guiding rule should be to protect the equipment and installation and, as much as practicable, protect against the prospect of catastrophic failure. Care should be exercised to follow the requirements of appropriate code documents and industry standards.

This chapter contains examples of both actual and hypothetical designs that reflect improvements in technology that have occurred since publication of the first edition of *Pumping Station Design* in 1989. The examples of pumping station design include:

- Section 26-1. Duplex submersible pumps for domestic wastewater in a hopper-bottom sump. Capacity: 25.9 L/s (410 gal/min). Improvement of original design.
- Section 26-2. Three dry-pit, V/S pumps for domestic wastewater in a trench-type sump. Capacity: 219 L/s (5 Mgal/d). Improvement of original design.
- Section 26-3. Three four-stage, two-speed vertical turbines for raw water in a trench-type sump. Capacity: 920 L/s (21 Mgal/d). Designed in 1993, and began service in 1995.

Dimensions and values in the plans and text for each section are, for simplicity, written in either SI units or in U.S. customary units but not usually in both. Readers should have no difficulty in transforming one set of units into the other.

26-1. Redesigned Clyde Wastewater Pumping Station

The Clyde Wastewater Pumping Station in Contra Costa County, California, was rebuilt in 1991 to

feature a self-cleaning sump. It is cleaned by pumping the water level down while vigorously mixing the contents with water from the force main. In the as-built plans, shown in Figure 17-22, eccentric plug valves in the valve vault can be regulated to take water from either the force main or from either of the two pumps. The water is discharged under considerable pressure at the surface of the lowered water level in the sump while a pump discharges the mixed liquid to the force main. The system works well for removing both scum and sludge and leaves the wet well remarkably clean.

In this section, the station is described as it might be designed in 2005 with the technology developed since the original plans were drawn. These changes consist of (1) steeper slopes to allow sludge to slide down to the pump intakes so that sludge is removed with every motor start, and (2) a sloping approach pipe for introducing the inflow without a cascade and for supplying added storage to reduce the size of the wet well. In other respects, the design approach closely follows that of the existing Clyde pumping station except that fewer valves are used in the valve vault.

The actual design was carried out in U.S. customary units, so those are the units used in this example. The original sewer design studies, surveys, and discussions with operation and maintenance staff established the following general requirements for this wastewater lift station.

- *General:* submersible pumps were preferred because of overall low cost, low maintenance, simplicity in operation, and minimizing visual impact on the neighborhood.
- *Flow rates:* Present average dry-weather flow: 30 gal/min.
  Present peak wet-weather flow: 236 gal/min.
  Future peak wet-weather flow: 410 gal/min (equals the capacity of one pump).
- *Ground elevation:* 13.2 ft. Pumping station site is relatively flat.
- *Force main:* An existing 8-in. cement-lined ductile iron pipe 2750 ft long was available.
  Invert elevation: 6.6 ft at the pumping station and 20.6 ft at the discharge.
  Slope: constant.
- *Reliability:* ability to pump future peak wet-weather flow with either of the two pumps out of service.
  Hook-up for portable engine-generator due to lack of space for permanent engine-generator.
  High wet well power-failure and intrusion alarm hooked up to an auto-dialer.
- *Location:* on shoulder of narrow residential street.
  Considerations include space, visibility, odors, noise, and security.

Station Siting

Station siting is established by the low point in the tributary area as well as access, availability of property, proximity to residents (i.e., farther is better), and the cost of piping to and from the site. The low point in the tributary area usually dictates the general location. Access is important because operation and maintenance staff must be able to visit the facility at any hour of the day and under adverse conditions. Access by public roads (paved, if possible) without the need to traverse private property or move parked automobiles is required. It is also preferable to provide room for maintaining the station without obstructing traffic or endangering workers.

Property and easement acquisition begins immediately after selecting the preferred site and before design on the pump station begins. Many projects have been delayed and/or designs changed because the site acquisition process did not begin soon enough. Such delays and changes will result in significant costs to the owner of the facility.

Hydraulic Design

Hydraulic design includes sizing the force main and developing the system curves, which are then used to select the number and size of the pumps. The rest of the facility is designed around the pumps. The force main invert elevation at the pumping station should, if possible, be set to allow for a constantly rising slope. High spots (knees) in a wastewater force main are to be avoided if at all possible because knees require air release valves, and it is wise to avoid them if possible (see Section 5–7). Force main installation costs increase with depth, so it is best to keep the pipe as shallow as possible. The discharge end of the force main is susceptible to hydrogen sulfide corrosion and should be protected by using corrosion-resistant piping (PVC, or HDPE) where exposed to air or else be submerged to prevent corrosion. Corrosion-resistant piping, as shown in Figure 26-1, should begin 10 ft before the point where the static water level contacts the soffit of the force main.

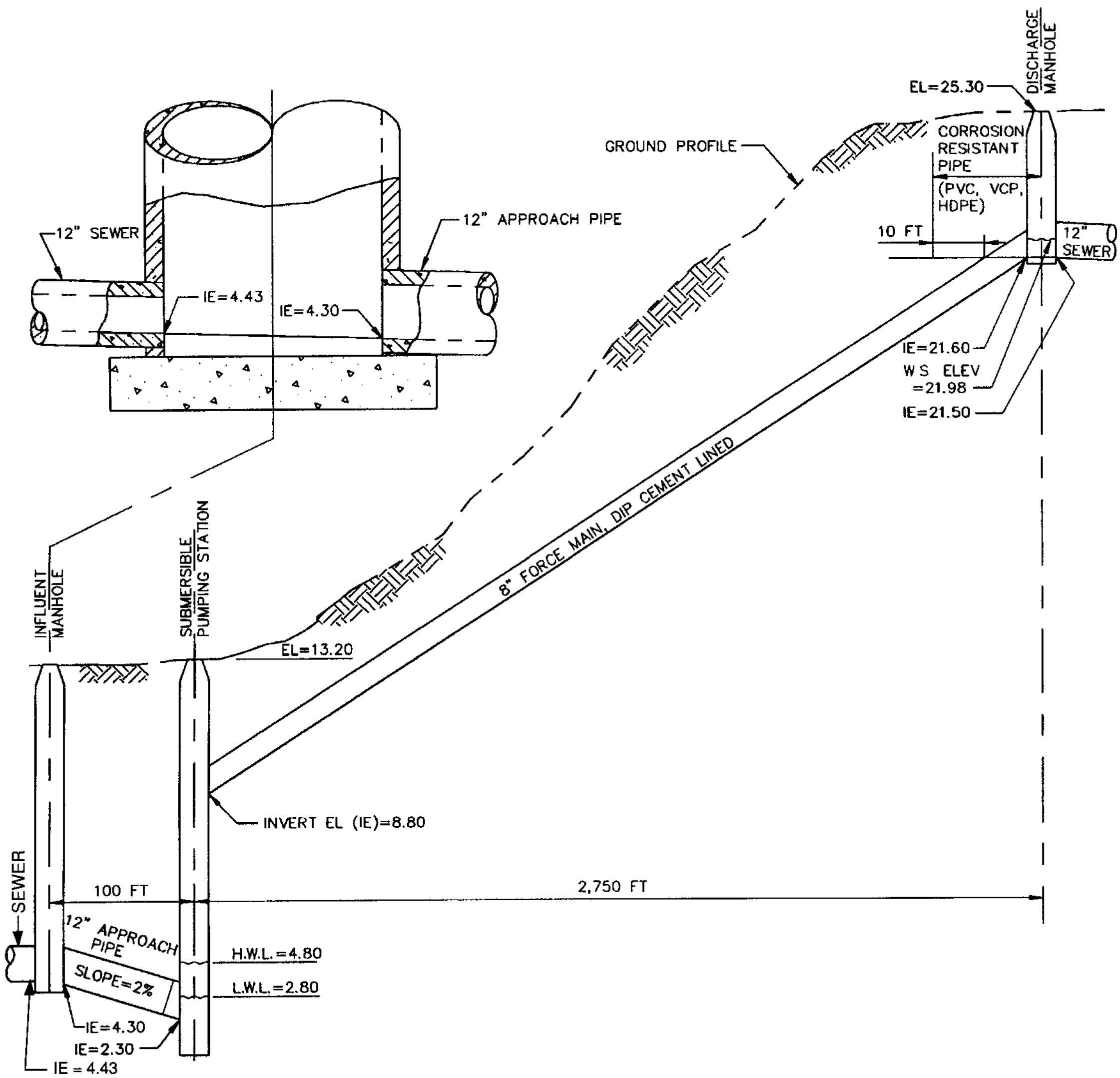


Figure 26-1. Piping profile.

Force mains should be sized to provide a minimum fluid velocity of 2.5 ft/s at present flows and a maximum fluid velocity of 6 to 8 ft/s at future peak wet weather flows. The minimum fluid velocity ensures that most solids will be moved through the force main. The maximum fluid velocity is set to minimize headloss and reduce surge pressures in long force mains. A 6-in. force main would meet these criteria for the stated flow conditions. However, an 8-in. mortar-lined ductile iron pipe was already in place and was therefore used. The fluid velocity in the force main for this pumping station with its one duty pump is 2.5 ft/s (see Table B-2 for the cross-sectional area).

System curves are developed to define the design operating point and extreme operating conditions for the pumps. Computations for these conditions at the future peak wet weather flowrate are given in Table 26-1. Note that K-values are not absolutes. Different engineers may elect to use different values. Those in Table 26-1 differ somewhat from those in Table B-6. Minimum losses are given for a pipe roughness corresponding to $C = 145$, and the maximum is for $C = 120$. The results are shown graphically in Figure 26-2. Point A is the normal flow and head condition, and Point B is the extreme flow and head condition at which the pump may operate.

Table 26-1. Total Dynamic Headlosses for $Q = 410$ gal/min (0.91 ft^3/s)

Description		Headlosses, ft	
		$C = 145$ Minimum	$C = 120$ Maximum
Station losses (4-in. DIP, $v = 10.3$ ft/s)	K-value		
Entrance	0.50	0.82	0.82
90° bends, 2 at 0.25	0.50	0.82	0.82
45° bend	0.20	0.33	0.33
Ball check valve	1.35	2.22	2.22
Eccentric plug valve	0.50	0.82	0.82
Tee, line flow	0.30	—	0.49
90° bend 4- to 8-in. expanding	0.50	0.82	0.82
Tee, branch flow, $v = 2.49$ ft/s	0.75	0.07	0.07
Force main losses			
2750 ft 8–9 in. DIP, lined ($v = 2.49$)		6.93	9.83
Minor losses, valves, discharge. $\sum K = 2.5$		0.24	0.24
Static lift		8.78[a]	19.18
Total design head		21.85 (≈22)	35.64 (≈36)

[a]Wet well is assumed to be filled to ground level.

Not all submersible pump manufacturers include the entrance and discharge elbow losses in their pump curves. The specified design point should be clear on what has been included for losses. The minor losses are not very important for pumping stations with long force mains, but they could be significant for a lift station with low head requirements.

Pump Selection

Operation and maintenance personnel prefer equipment with which they are familiar. They do not like "experimental" applications. Pump selection starts by soliciting input from the people who operate them. Identical pumps are used for multiple pump

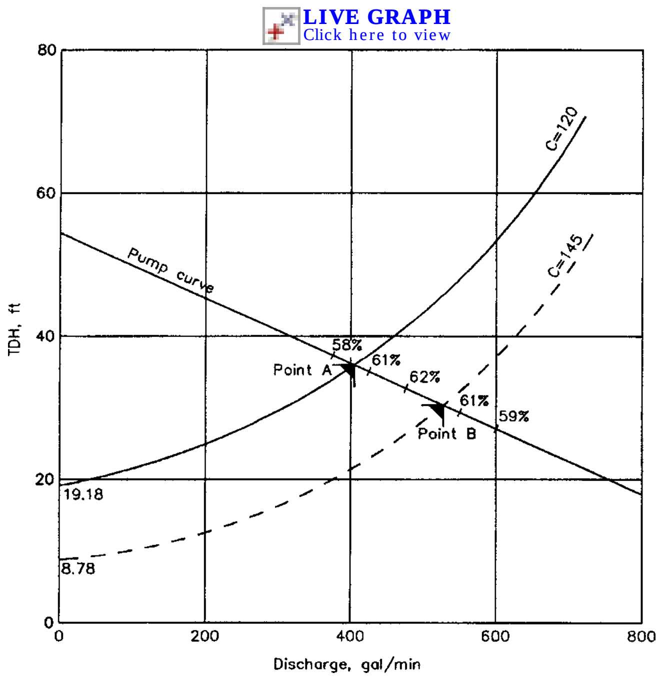


Figure 26-2. Pump and system head-capacity curves.

applications whenever possible to simplify maintenance and provide interchangeability of parts. It is advantageous to have service and parts available from a nearby source.

The selected pump curve and impeller diameter should meet the design point near the best efficiency point (BEP). The selected pump should also operate at the low head and high head extremes without cavitating or vibrating. Pump curves with steep slopes are better than those with flat slopes because there is less variation in capacity with varying head conditions. (Flat spots or dips in the pump curve are therefore undesirable.) Wherever possible, choose impellers of intermediate size so that a larger impeller can be substituted for larger future flows. The motor is sized for the worst possible operating point (which is often the low head extreme with one pump operating). For multiple pumps, the best pump efficiency should be at normal operating conditions and not at the ultimate peak flows. However, the emphasis should be on finding pumps that can operate within the POR at the most frequent pumping condition, within the AOR for all pumping conditions, and without vibration or cavitation at *all* anticipated service conditions.

Wet Well

The wet well is designed to (1) have adequate space for the pumps, (2) facilitate cleaning, (3) contain sufficient storage volume, (4) limit pump starts, and (5) minimize installation costs. For a small duplex submersible pumping station, the most economical wet well is often a reinforced concrete pipe 1.8 or 2.4 m (6 or 8 ft) in diameter standing on a cast concrete bottom.

To improve solids removal, the pumps are confined by close-fitting, nearly conical, smooth walls sloping at 60 degrees or more. Sludge and grit slide down the walls to the pump suction. If the walls clear the pump volutes by no more than 100 mm (4 in.), the flat floor area is small and within the influence of the suction currents of both pumps. Consequently, the sludge is so confined that it cannot accumulate under the inactive pump for longer than one pump cycle if the pumps are alternated. The sloping walls may be constructed by placing a custom form in the wet well and injecting concrete behind the form. Fiberglass-reinforced plastic or stainless steel (or any smooth, durable material) can be used as the form and can then remain in place as a liner to provide a smooth, corrosion-resistant surface to facilitate cleaning. Holes are required in the form for the pump discharge elbows, which are largely contained within the walls. If a disposable form is used, the concrete should be covered with a protective liner such as PVC. The vertical walls above the “cone” should also be protected with a PVC liner.

Active Volume

The active or working volume of wet well and approach pipe must be adequate to limit the frequency of pump starts to a safe value. Conical bottoms reduce the volume available in the wet well, but the addition of an approach pipe laid on a 2% gradient, as described in Example 12-2, supplies additional volume.

The normal LWL is set above the invert of the approach (inlet) pipe by 60% of its diameter and the invert is set at or somewhat above the top of the pump volute. Except during cleaning, (1) the motor is always at least half-submerged, and (2) there is no free fall into the wet well. Consult the manufacturer about the minimum submergence for the motor and the maximum frequency of pump starts. Starting frequency for small to medium-sized submersible pumps may be expected to vary between 10 and 20 starts/hour depending on the resting time between starts. Using an alternator in stations with multiple pumps to switch lead pumps after each pump cycle reduces the starting frequency and prolongs the life of the motor. However, one pump may be out of service, so duplex stations must be operable without alternation. Nevertheless, when both pumps are serviceable, alternation could be used to extend the life of the starters and motors.

Inlet pipes are traditionally oriented in a plane normal to the plane of the pumps, as shown by the dashed lines in Figure 12-31. Such a configuration tends to promote rotation that could increase to swirling in pump intakes. A better configuration is to orient them in the plane of the pumps so that the pump motor acts as a baffle for the incoming current as shown in Figure 26-3a and b by the solid lines in Figure 12-31. Baffles are desirable to break up incoming currents and reduce turbulence. Furthermore, as the current is split by the pump motor, there is no tendency for rotation to occur.

Calculate the required active volume from Equation 12-8. If the allowable starting frequency for one pump is 10 cycles/hour, each cycle takes 6 min and the required volume is

$$V = \frac{TQ}{4}$$
$$= \frac{(6\,\text{min})(410\,\text{gal/min})}{4} = 615\,\text{gal} = 61.8\,\text{ft}^3$$

where T = time in minutes and Q = flow in gpm.

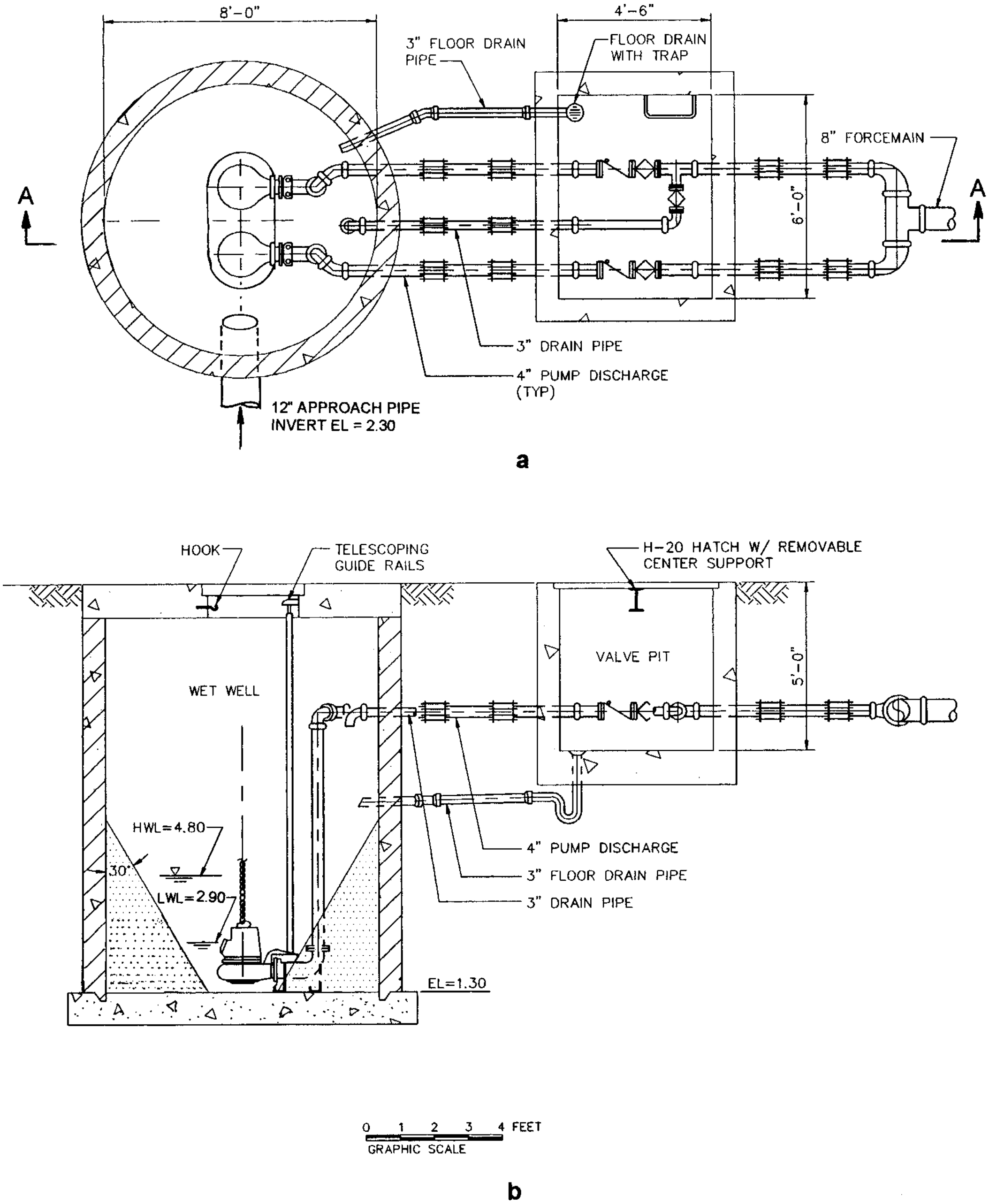


Figure 26-3. Layout of redesigned Clyde pumping station. (a) Plan view; (b) Section A-A.

The wet well and approach pipe are laid out so that the working volume is 615 gal. The existing wet well has limited volume, but an approach pipe 100 ft long by 12 in. in diameter, laid on a 2% slope, increases the working volume to 870 gal. The invert elevation at the wet well is 2.30 ft. The LWL is set at 2.80 ft to force the hydraulic jump to occur in the pipe and not in the wet well. The HWL is set at 4.80 ft. At the upstream manhole, the invert elevation of the approach pipe is 4.30 ft. On a rising grade of 2%, the invert on the upstream side of a 48-in. manhole would be 0.08 ft higher, but there is some form and friction loss in the transition from half-filled sewer to quarter-filled approach pipe, so an increase

of 0.05 ft brings the sewer invert to an elevation of 4.43, as shown in Figure 26-1.

Standby Power

The site cannot accommodate the installation of a permanent engine-generator for standby power. The operator has chosen to store a portable engine-generator at a site about a mile from this facility. A power failure and high-water alarm indicate potential overflow at the station. During dry-weather flow, more than one hour is available to transport and connect the portable engine-generator. However, during wet weather periods, there may be less than 10 min available for that task. Wastewater pumping stations should normally be provided with permanent, on-site standby power to reduce the exposure to wastewater overflows.

Station Piping

Station piping within the wet well is limited to the two pump discharge lines and a force main drain line. Piping within the wet well is as simple as possible with few fittings and no valves. Flange bolts within the wet well are 316 stainless steel, as are the pipe supports and hardware. The piping design within the wet well minimizes items that corrode, that require regular maintenance, or that may catch floating debris. Valves and fittings are contained in a separate valve pit next to the wet well.

The valve pit contains the pump isolation and check valves on the two pump discharge lines and their connection to the force main. A valved force main drain line that discharges back into the wet well is also included. It can be used to agitate the contents of the wet well vigorously so that scum, mixed into the contents, is ejected with the wastewater. Such mixing eliminates the need to pump the water level down to the pump volute to develop vortices for engulfing the scum—an operation that subjects the pump to vibration, stress, and wear of the mechanical seals. Valve stems and nuts are extended nearly to grade to permit operation without the need to enter the vault.

An alternative is the use of a pump that ejects part of its discharge into the wet well during the first minute or two after the pump is switched on. The water ejected through a flush valve mixes the solids so that some are discharged to the force main during each pump cycle. The advantage is that the wet well is kept continuously clean automatically. The disadvantages are a small loss of efficiency, the added mechanical device on the pump, and a single manufacturer.

Controls and Alarms

The pumps are set up in a lead/lag arrangement with automatic alternation after each pump cycle to balance run times and minimize starts per hour. HAND/OFF/AUTO operator selector switches are provided for each pump at the control panel located with a view of the wet well. Pumps are operated by the wet well water level indicated by a pressure transducer/transmitter hanging in a PVC pipe in the wet well. Keypad controllers are generally preferred because they are easy to operate, do not have pins (easy to lose), and can be programmed with a security code. The high-water-level alarm consists of a float switch connected to an auto-dialer that pages the on-call operator. A stainless-steel chain supported by the hatch frame is connected to the float switch so the operator can periodically test the switch. A nylon cord would serve the same purpose at less cost.

Operation and Maintenance

Double-leaf, spring-loaded aluminum hatches rated for an H-20 loading are installed over the wet well for access to the pumps. Similar hatches are installed over the valve vault. Safety chains are provided to create a barrier when the hatch is open. Hatches are equipped with padlocks for protection against vandalism.

Telescoping stainless-steel tubes are used for guide rails for installing the pumps. Except during times needed, they are lifted out of the water and hence do not collect debris. Lifting chains to the pumps are omitted because it is easier to lift the pump by its lifting bail. Lifting by a chain requires a high boom. The owner uses a truck-mounted boom and winch for lifting pumps but, alternatively, a fixed crane or hoist could be installed on site. Winches are equipped with ratchets for use in both directions.

The site is equipped with overhead lights with electric power supplied from a nearby power service pole. Lights are also provided within the outer door of the control panel. A weather guard extending 18 in. in front of the panel is installed at the top front of the control panel. A hook-up with a manual transfer switch makes it easy to use the owner's trailer-mounted engine-generator.

Wash-down water is obtained on-site from the potable water supply. The equipment consists of an

air-gap tank, water pump, and hose bibb contained in a locked steel cabinet. The wash-water pump, equipped with a preset timer for automatic shutoff, also has an automatic recirculation line with a pressure regulating valve. Steel traffic bollards are placed around the water and electrical control cabinets. There is no fencing around the site.

Final Check

After completing the pump selection and piping layouts, the system hydraulics are checked again to see that the pump selected is the best fit, and that the motor and electrical gear are sized adequately. The drawings and specifications are reviewed by the owner, and the operators are walked through the design system-by-system. Final revisions are made before bidding the project.

Critique

Compare Figures 26-3, 17-21, and 17-22. The choice of pumping station configuration for each site should be based on sound judgment in which first cost is balanced against cleanliness and desired control of odors and the ease and cost of maintenance.

Some operators prefer that the pump discharge lines be cross-connected with each other upstream from the check valves (with a valve on each branch) and connected to the force main drain line, as in Figure 17-22. Although such piping increases the number of valves, the size of the valve vault, and the project cost, it allows the operator to agitate the wet well contents with one pump before pumping to the force main with the other. It also allows the operator to backflow one pump with the other to remove clogs without removing the pump.

The plans in Figure 26-3 indicate a cone-shaped hopper bottom. To keep the water surface area as small as possible so that scum will be readily drawn into the pump intakes at pump-down, the sides of the sump should hug the pump volutes with no more than 4 in. of clearance. These pumps are too far apart. A clearance of 4 in. is enough. Interference is no consideration. The pumps are not operated simultaneously.

26-2. Redesigned Kirkland Wastewater Pumping Station

Designed originally in 1965, the Kirkland Pumping Station is shown in U.S. customary units in Figures 17-13 and 17-14, so the same units are used in this section. The station has been in continuous operation since 1967. The following example is a revised version of the existing station, modified to reflect: (1) wet well design developments that facilitate the self-cleaning features described in Chapter 12; (2) the best in current technology; and (3) more recent and stringent reliability standards. The station consists of three 2.5 Mgal/d pumps operating against a total dynamic head of 189 feet.

The top of the influent sewer is no more than 6 ft below finished grade. For such a shallow site, horizontal pumps were selected for both the 1965 and the current designs. Because they are less prone to vibration, horizontal pumps are preferred when they can be justified by little increase in structure cost. Instead of the eddy-current couplings used in the original station, the revised design has 125-hp adjustable-frequency drives. A 300-kW standby generator protects against power outages.

The force main terminates at an interceptor sewer 3150 ft from the pumping station site at an elevation 123 ft above the soffit of the influent sewer. Calculations were performed by using *PUMPGRAF*© [1], a computer spreadsheet program configured specifically for pumping station design work.

Individual Hydraulic Losses

Calculations for individual hydraulic losses from the pump inlet to the connection with the discharge manifold were performed first and are shown in Table 26-2. The pump inlet bell diameter in the wet well was selected on the basis of a conservative limiting fluid velocity of 4.5 ft/s even though ANSI/HI 9.8-2000 allows 5.5 ft/s. Maximum velocities in pump connecting piping were considered acceptable for a variable-speed station, where higher losses are only realized when the equipment operates at full speed. Upon completion of these calculations, they were automatically loaded into the program for the calculation of station system losses.

Station System Losses

Station system headlosses in the force main (including static lift) were calculated from the manifold to the point of discharge and are shown in Table 26-3. Instead of the asbestos cement pipe used in the original project, HDPE was the material selected for the force main. Losses were initially calculated for a Hazen-Williams C of 140 and then recalculated for

Table 26-2. Pump Intake and Discharge Headlosses

2.5	= Starting Flow	– mgd	Title:	Revised Kirkland Pumping Station
119.98	= Elev @ Start	– Feet		Brown and Caldwell
140	= Default C-Value	>25	New C:	140 HiGrf LoGrf PmpCorr
14	= Default Diam	–Inch	New D:	14 53.34 = Target Headloss
304.8	= Default Pipe Length	–Feet		9.185 = Total Headloss – Feet
Date:	06-July 96	ParaEq =		3169.6 = EqivLnth (Std=def) – Feet

Flow, mgd	Item Description of Friction Loss	Diameter, in.	K or Cval	Fixed Loss	Length, ft	Item No.	Loss, ft	Hyd Grad, ft Pmp → Dch	Vel, fps	V-Hd, ft
2.5	Bell Mouth Entrance	14	0.05			1	0.01	119.97	3.6	0.2
2.5	90° Elbow	10	0.3			2	0.23	119.74	7.1	0.78
2.5	Plug Valve	10	0.6			3	0.47	119.27	7.1	0.78
2.5	Reducer	10	0.01			4	0.01	119.26	7.1	0.78
2.5	PUMP	14	0.01			5	0.00	119.26	3.6	0.2
2.5	Increaser	6	1			6	6.03	113.23	19	6.03
2.5	90° Elbow	10	0.6			7	0.47	112.76	7.1	0.78
2.5	Check Valve	14	3.5			8	0.71	112.05	3.6	0.2
2.5	Plug Valve	10	0.6			9	0.47	111.58	7.1	0.78
2.5	Branch Flow Tee	10	1			10	0.78	110.80	7.1	0.78
2.5						11				

Table 26-3. Force Main Headlosses for $C = 140$

5	= Starting Flow –	mgd		Revised Kirkland Pumping Station
119.98	= Elev @ Start –	Feet	Title:	Brown and Caldwell
120	= Default C-Value >	25	New C:	120 HiGrf PmpCorr
14	= Default Diam –	Inch	New D:	14 = Target Headloss
1000	= Default Pipe Length –	Feet		169.49 = Total Headloss – Feet
Date:	06-July-96	ParaEq =		3341.6 = EqivLnth (Std=def) – Feet

Flow, mgd	Item Description of Friction Loss	Diam, in.	K or Cval	Fixed Loss	Length, ft	Item No.	Loss, ft	Hyd Grad, ft Pmp → Dch	Vel, fps	V-Hd, ft
5	Fixed Static head			122		1	123.00	289.48		
5	90° Elbow	14	0.6			2	0.49	288.99	7.2	0.8
5	Straight Pipe	14	120		25	3	0.35	288.64	7.2	0.8
5	90° Elbow	14	0.3			4	0.24	288.40	7.2	0.8
5	Straight Pipe	14	120		140	5	1.95	286.45	7.2	0.8
5	45° Elbow	14	0.25			6	0.20	286.25	7.2	0.8
5	Straight Pipe	14	120		1040	7	14.47	271.77	7.2	0.8
5	90° Elbow	14	0.3			8	0.24	271.53	7.2	0.8
5	22° Elbow	14	0.1			9	0.08	271.45	7.2	0.8
5	Straight Pipe	14	120		975	10	13.57	257.88	7.2	0.8
5	90° Elbow	14	0.3			11	0.24	257.64	7.2	0.8
5	Straight Pipe	14	120		995	12	13.85	243.79	7.2	0.8
5	Outlet Loss	14	1			13	0.81	242.98	7.2	0.8
5						14				

a *C* of 120. Individual losses for the pump inlet and discharge piping in Table 26-2 were not included in these calculations.

Pump Selection

Station system losses were then transferred through the program to the pump selection program (see Table 26-4), and a pump was selected from a previously entered library of pump manufacturers' catalog information. Plots of the selected performance curves against station system curves are shown in Figures 26-4 and 26-5. The pump performance curves plotted on the figure have been adjusted for individual pump inlet and discharge piping losses of 9.2 ft at 2.5 Mgal/d. These values must be added to the information on the plot to arrive at the correct rating for the pump. In this example, the pumps are to be rated at 2.5 Mgal/d at a total head of 180 ft. The pump selection is considered acceptable because the intersection between the pump performance curve and the expected range of operating conditions lies well within the AOR. Note that the intersection between the manufacturer's curve and the station system curves lies to the right of the pump's best efficiency point. As speed is reduced in variable-speed operation, the point of intersection of the curves passes through the pump's zone of best efficiency.

After an acceptable pump selection was found, the program was used to evaluate pump performance at variable speed, and these data are shown in Table 26-5 and in Figure 26-5. As indicated, the minimum operating speed is approximately 1300 rev/min, which corresponds to a flow rate of 0.6 Mgal/d.

Table 26-4. Pump Selection for Two Pumps Operating

ENTER PLOT DATA RIGHT: Title --> Revised Kirkland Pumping Station
ENTER PUMP DATA BELOW: Brown and Caldwell

Name——— Fairbanks Morse
Nonclog, 4" 5414/5424
Crv/Imp—T4D1B, 3" solids

Impeller Rng	12.0	to	15.5
Speed Rng	1150	to	1785

15.5 = Impeller for Curve Points
1785 = Speed for Curve Points

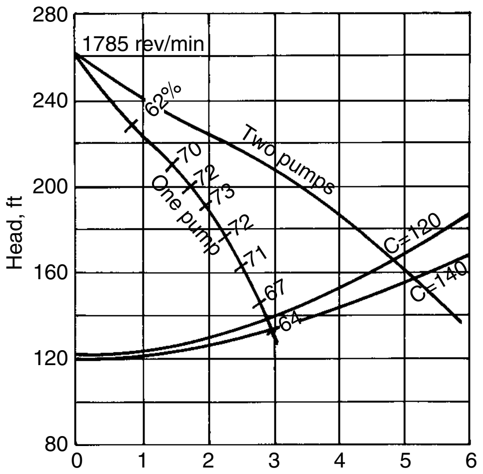
Point/ Eff/NPSH	Head, ft	Flow, gpm	Head/Stg
	290.00	0.00	
	267.00	400.00	
62.00	248.00	800.00	
70.00	232.00	1200.00	
71.50	222.00	1400.00	
72.80	212.00	1600.00	
72.30	200.00	1800.00	
70.50	188.00	2000.00	
67.00	170.00	2200.00	
64.00	158.00	2300.00	

List Up To 12 Points In Any Order.
Put 1st Pump Curve Labels First
Column (Optional).
Delete or Add -->
Curve Labels -->
To Suit -->

Imported Data->Hi-Grf Lo-Grf
Pump Curve Calculations
2 = Number Pumps
For Fixed Speed Pumps,
Omit 2nd, 3rd & 4th Speeds
Data For Plot
14.70 = Impeller for Plot
1785.00 = Speed for Plot
= Optional 2nd Speed
= Optional 3rd Speed
= Optional 4th Speed

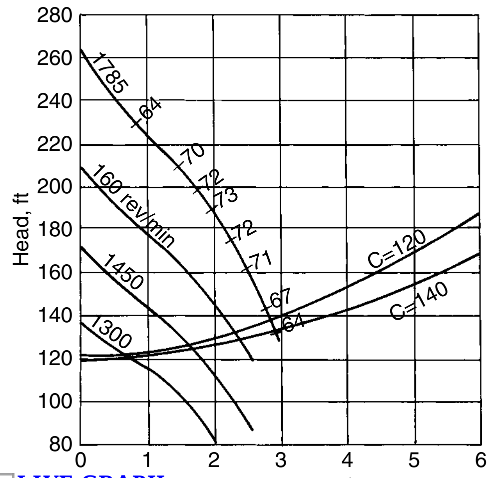
Pump Curve Correction, 0 if not used
9.18 feet Pmp Pipg Loss
2.50 mgd Flow Used

Overriding System Head Curve Data
0.00 mgd =Design Flow
100.00 = C-Value for Friction Loss
11.01 feet =Frtn Loss @ DgnFlo
50.00 feet =High Static Head
50.00 feet =Low Static Head
120.00 =Low C-Value
140.00 =High C-Value
1.40 =Peak Factor Sys Head
C = 120.00 Hi-SysHd Curve Label
C = 140.00 Low-SysHd Curve Label
1785.00 1st Pump Curve Label
2nd Pump Curve Label
3rd Pump Curve Label
4th Pump Curve Label



LIVE GRAPH
Click here to view

Figure 26-4. Pump and system curves.



LIVE GRAPH
Click here to view

Figure 26-5. Single pump at variable speed operation.

Table 26-5. Operation of a Single Pump at Variable Speed

ENTER PLOT DATA RIGHT: Title --> Revised Kirkland Pumping Station
ENTER PUMP DATA BELOW: Brown and Caldwell Consultants

Name——— Fairbanks Morse
Nonclog, 4" 5414/5424
Crv/Imp—T4D1B, 3" solids

Impeller Rng 12.0 to 15.5
Speed Rng 1150 to 1785
15.5 = Impeller for Curve Points
1785 = Speed for Curve Points
NmbrStgs = 1.00

Point/ Eff/NPSH	Head, ft	Flow, gpm	Head/Stg
	290.00	0.00	
	267.00	400.00	
62.00	248.00	800.00	
70.00	232.00	1200.00	
71.50	222.00	1400.00	
72.80	212.00	1600.00	
72.30	200.00	1800.00	
70.50	188.00	2000.00	
67.00	170.00	2200.00	
64.00	158.00	2300.00	

List Up To 12 Points In Any Order.
Put 1st Pump Curve Labels First
Column (Optional).

Delete or Add -->
Curve Labels -->
To Suit -->

Imported Data->Hi-Grf Lo-Grf
Pump Curve Calculations
1 = Number Pumps
For Fixed Speed Pumps,
Omit 2nd, 3rd & 4th Speeds
Data For Plot
14.70 = Impeller for Plot
1785.00 = Speed for Plot
1600.00 = Optional 2nd Speed
1450.00 = Optional 3rd Speed
1300.00 = Optional 4th Speed

Pump Curve Correction, 0 if not used
9.18 feet Pmp Pipg Loss
2.50 mgd Flow Used

Overriding System Head Curve Data
0.00 mgd =Design Flow
100.00 = C-Value for Friction Loss
11.01 feet =Frtn Loss @ DgnFlo
50.00 feet =High Static Head
50.00 feet =Low Static Head
120.00 =Low C-Value
140.00 =High C-Value
1.40 =Peak Factor Sys Head
4.50 =Curvature factor
C = 120.00 Hi-SysHd Curve Label
C = 140.00 Low-SysHd Curve Label
1785.00 1st Pump Curve Label
1600.00 2nd Pump Curve Label
1450.00 3rd Pump Curve Label
1300.00 4th Pump Curve Label

Hydraulic Profile

Next, a plot (Figure 26-6) of the force main and hydraulic profiles was constructed for a visual portrayal of hydraulic conditions in the system at both minimum and maximum system capacity. From an inspection of the plot, it was found that the hydraulic profile remains well above the force main profile from the pumping station to the point of discharge under all operating conditions. In addition, there appeared to be no high points or knees that would be the location of column separation in a hydraulic transient condition. Finally, the pipeline profile does not touch the mirror image line, so column separation does not appear to be a problem (see Section 7-1). For a 21-in. force main constructed of HDPE, the wave propagation velocity is about one-third of that for ductile iron or asbestos cement (refer to Chapter 7). Based on inspection of the force main/hydraulic profile plots and the low wave propagation velocity, a formal transient analysis might not be necessary. However, under the rules given in Section 6-8, a transient analysis would be required because (1) the TDH exceeds 50 ft and the flow rate exceeds 500 gal/min, and (2) the pipeline has a long, steep gradient followed by a long, shallow gradient.

Geometry

The station capacity is 5 Mgal/d or 7.75 ft^3/s. From the principles presented in Chapter 12, the cross-sectional area of the wet well above the trench should be sufficient to limit the average forward (plug flow) velocity above the trench to 1 ft/s for any inflow. At the maximum inflow of 5.0 Mgal/d, a cross-sectional area of 7.75 ft^2 is required, and at half the flow with the inlet pipe half full, the required cross-sectional area above the trench must be no less than 3.88 ft^2. In this design, the second criterion is the more critical and results in locating the top of the trench 4.0 in. below the invert of the influent pipe. If the sides are allowed to slope at 45 degrees, the wet well above the inlet pipe can be made 5 ft wide. Note that installing the sluice gate requires a flat surface at least 12 in. wider than the diameter of the inlet pipe.

Pump Intakes

The dimensions of a ductile iron bend and flare were judged to be unfavorable, so a special fitting consisting of a short-radius steel 90-degree bend welded between a 14 × 10 reducer (with an OD of 14 in. or 1.17 ft) and a steel flange was selected. The trench was made 2.33 ft wide—equivalent to two bell diameters.

The maximum intake fluid velocity for a single pump at a flow rate of 2.9 Mgal/d (see Figures 26-4 and 26-5) is 4.2 ft/s. The minimum center-to-center spacing of pump intakes should be at least 2.5 *D* or 2.92 ft.

Intake Submergence

Submergence, governed by Equation 12-1, was calculated to be 3.0 ft at runout conditions for a single

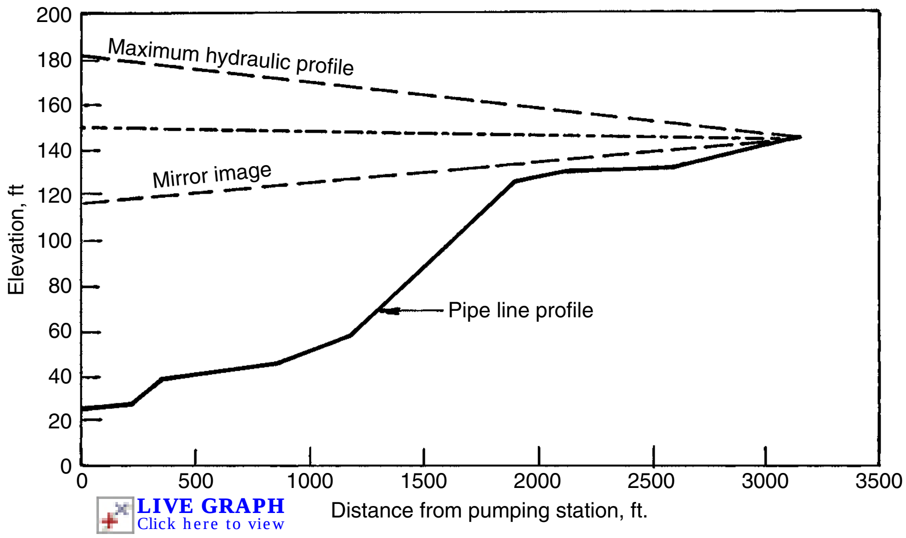


Figure 26-6. Plot of hydraulic and force main profiles.

pump. (Details of a somewhat similar calculation are given in Example 12-1, Part C.) The LWL would be set somewhat above the invert of the influent pipe. For example, minimum flow would be expected to be about one-tenth of maximum flow or 0.5 Mgal/d. By using Figure B-4 or Table B-8, minimum flow depth in the influent sewer can be determined and the LWL set at that same elevation. For practicality, however, if the 4 ma signal from a 4 to 20 ma range is set at the invert of the influent pipe, the LWL should be about right. The trench depth was based on submergence with respect to (1) wet well level even with normal flow depth in the influent sewer, (2) LWL at runout for a single pump, and (3) a floor clearance of D/2 for upstream pumps. The trench must be at least 2.5 D deep, but it might have to be deeper to satisfy submergence requirements.

The last pump (farthest from the influent sewer) has its inlet located D/4 above the floor. Find the sequent depth during pump-down, however, and make sure the inlet is about D/2 below the water surface to avoid losing prime too quickly. For other wet wells, it might be necessary to drop the floor at the last pump. For this short trench, a dropped floor is unnecessary as proven by model studies.

Plans

The plans, shown in Figures 26-7 through 26-9, illustrate the conceptual design for the revised station. Because the station must be located on a very small site, the pump inlets were located at the minimum permissible separation so as not to encroach on the space needed for the engine-generator. The pump inlet piping for Pumps 2 and 3 were skewed to permit the pumps to be located with a clear separation of 3.5 ft between bases.

The wet well has been revised to include an ogee ramp and a motor-operated sluice gate to protect the wet well in emergencies. Instead of locating a grated walkway over the pump inlet channel as in the original design, a walkway alongside the channel makes access far more convenient for housekeeping chores (such as hosing grease off the walls).

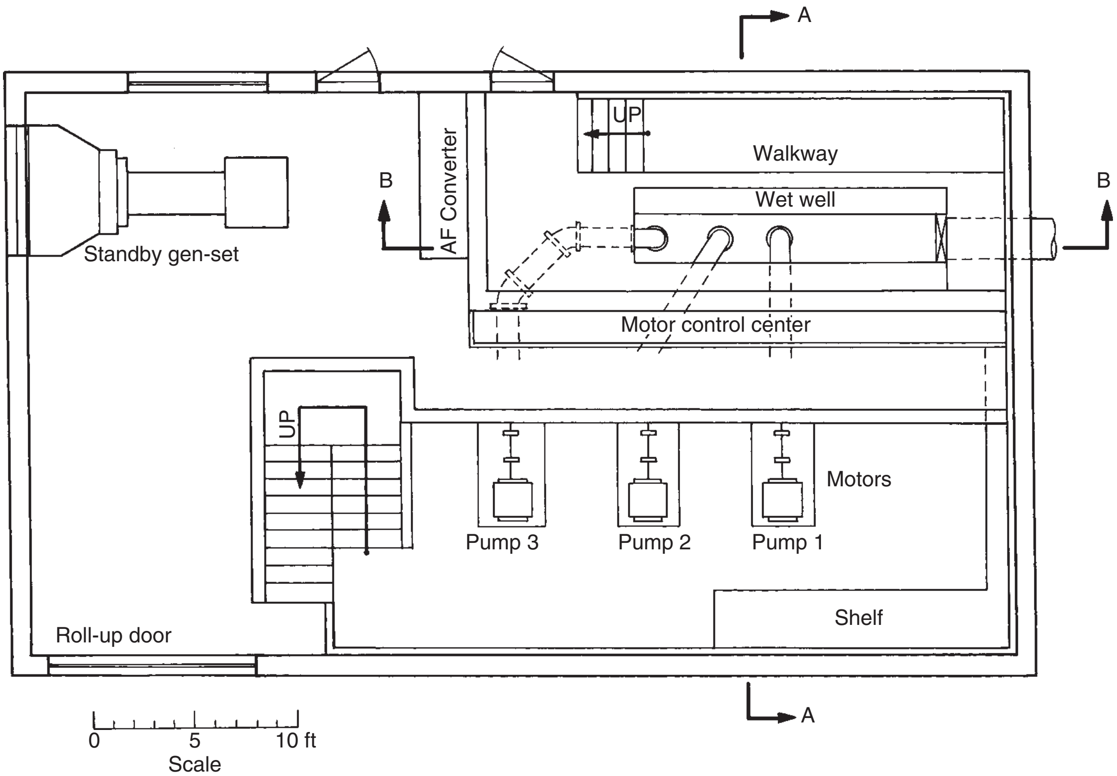


Figure 26-7. Ground floor plan. Revised Kirkland Pumping Station.

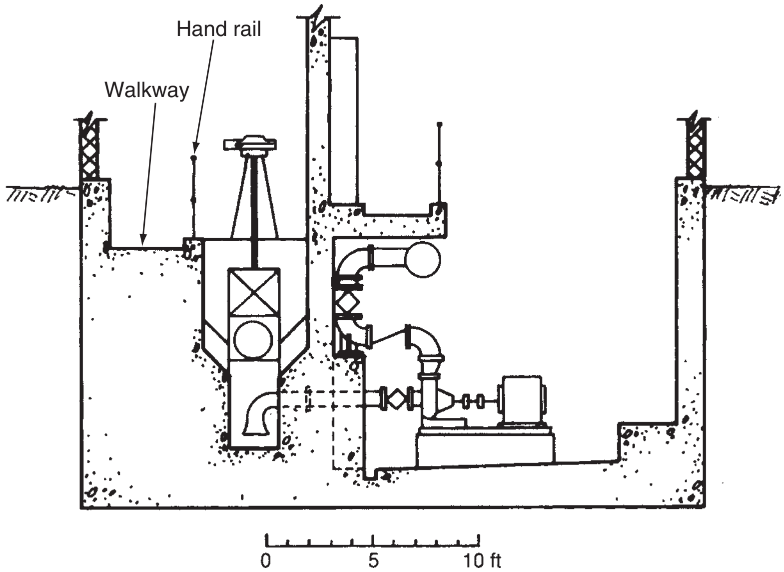


Figure 26-8. Section A-A. Revised Kirkland Pumping Station.

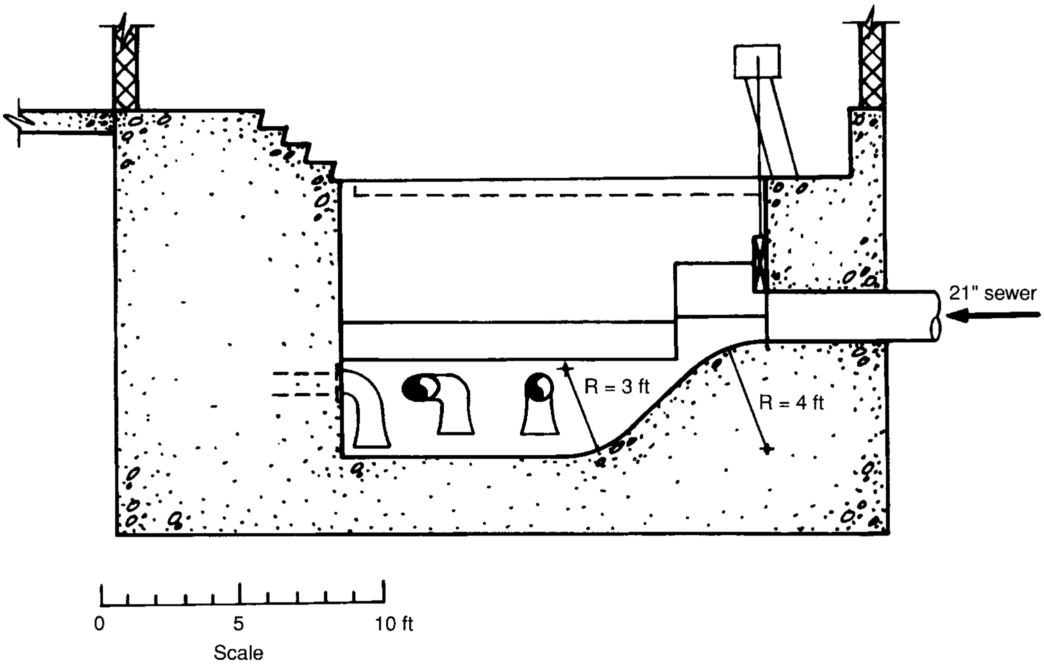


Figure 26-9. Section B-B. Revised Kirkland Pumping Station.

Instead of a spiral stair (permissible under then-prevailing codes), a conventional stairway for access from grade level to the pump room is used in the revised design. All station auxiliaries (water pumps, hydropneumatic tank, air compressors) are located on an L-shaped bench along the south and east walls. To conserve space at grade level, cabinets for electrical equipment (instead of being located in a separate room with a controlled climate) are to be furnished with an internal cooling system.

Critique

The suction bells, spaced at 2.5 *D* on centers, could perhaps be spaced at 2.0 *D* on centers (as shown for unconfined wet wells in an older edition of the Hydraulic Institute's Standards [2]) and thereby save 14 in. of wet well length. But the greater spacing is more conservative and still keeps the wet well very short. Whether to keep this spacing and splay or bend the suction pipes, or to use rectilinear piping (as in the original shown in Figure 17-13) and a much longer wet well, is a matter of engineering judgment. One disadvantage of the plans shown here is that if the suction piping for Pump No. 3 ever clogs, it would be more difficult to rod out debris than it would be for a straight pipe. The need for rodding, however, is rare because of the protection afforded by setting the pump intakes at relatively small floor clearances. The pump intake piping will be fitted with flush-out connections to permit clearing the pipe with high-pressure water. Mitered bends of a radius long enough to pass rodding equipment could be used as an alternative. Another disadvantage is that because of the elbow connection, the distance from the back wall to the center of the pump intake is 1.0 *D* instead of the preferred 0.75 *D*. The latter disadvantage can be overcome by adding vertical fillets at the corners of the back wall to inhibit vortex formation in the stagnant water downstream of the intake.

A trench width of 2.33 ft is too narrow for installing flow splitters, cones, or fillets. Omit them, as they were omitted in the original pumping station. Although side wall and floor vortices are known to exist, they have caused no problems.

26-3. Jameson Canyon Raw Water Pumping Station

The Jameson Canyon Pumping Station was constructed to replace an existing installation at the Cache Slough Reservoir, which receives water from the Sacramento River system and delivers it to the City of Vallejo, California, Fleming Hill Water Treatment Plant. In addition to the new pumping station, the project included construction of a parallel pipeline from the reservoir to the treatment plant on an alignment through Gordon Valley. The original station, which has been retained and modified to function as a reserve facility, was found to be troublesome because of capacity-limiting inlet conditions and other defects. In addition to other problems, the station was sometimes troubled by accumulations of silt in the reservoir.

Vertical turbine pumps were selected for the new station because it could be constructed without the cost of a pump room below ground level. Three pumps were installed initially, with a position for a fourth. Each pump has a nominal capacity of 416 L/s (9.5 Mgal/d).

The new pumping station can receive water from three sources, all with differing energy gradients. To limit costs and provide acceptable sump levels for operation with any source, a modulating butterfly valve (installed upstream from the station but downstream from the connection to the three sources) operates to maintain a fixed level in the pump sump. The siltation problem noted in the operation of the existing station dictated a sump configuration that could avoid accumulations of the silt that hampered the operation of the existing station. The plan and cross-sectional views of the finished design are shown in Figures 26-10 and 26-11.

An evaluation of operational considerations showed that two-speed pumps afforded considerable savings in operation and maintenance costs over variable-speed drives, which would have no significant advantages for this facility. The sump was designed with sufficient volume to accept a two- to three-minute fluctuation in water delivery rate without upsetting the pump control system. Power-actuated 400-mm (16-in.) ball valves on the pump discharges were specified to control surges in the station sump on pump start-up and shut-down.

An overflow consisting of a weir, trough, and three 600-mm (24-in.) pipes (Figures 26-10 and 26-11) from the sump to the reservoir is for relieving surges and backflow to the sump on power failure.

After a thorough economic analysis of the needs and options for service during a power failure, an on-site emergency generator was eliminated in favor of reconditioning a 597-kW (800-hp) pump with a combination motor and engine drive installed in the existing station. Piping from the new station to the existing station a short distance away included power-actuated valves that can be operated from a remote control center.

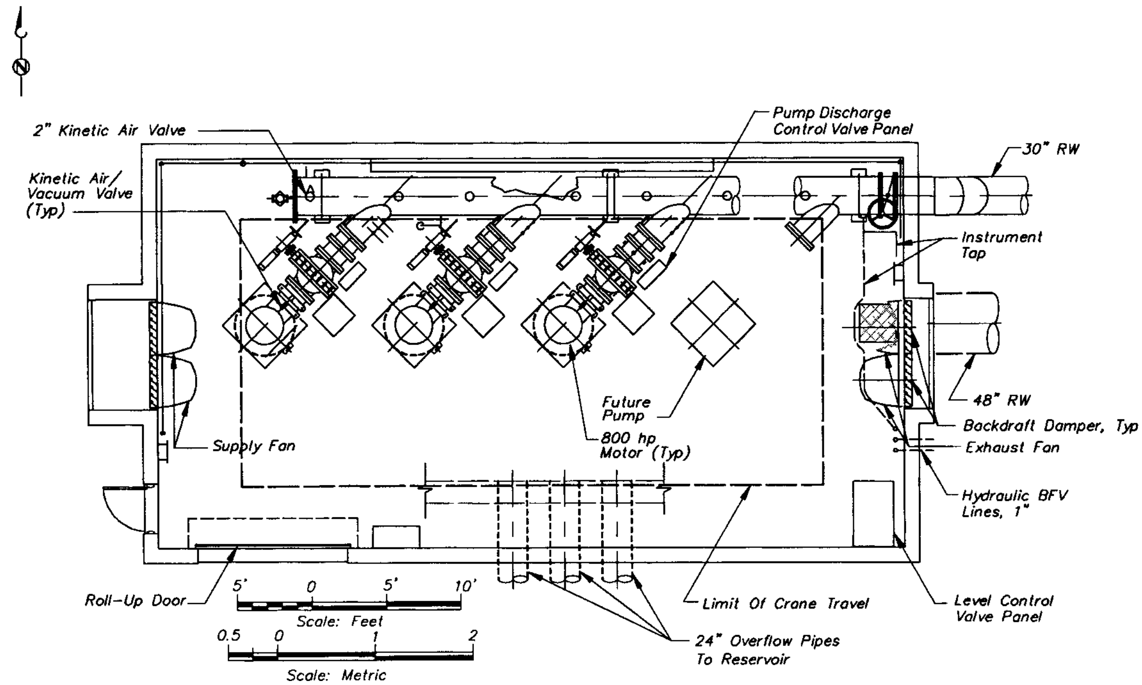


Figure 26-10. Plan of Jameson Canyon Pumping Station. Courtesy of Brown and Caldwell.

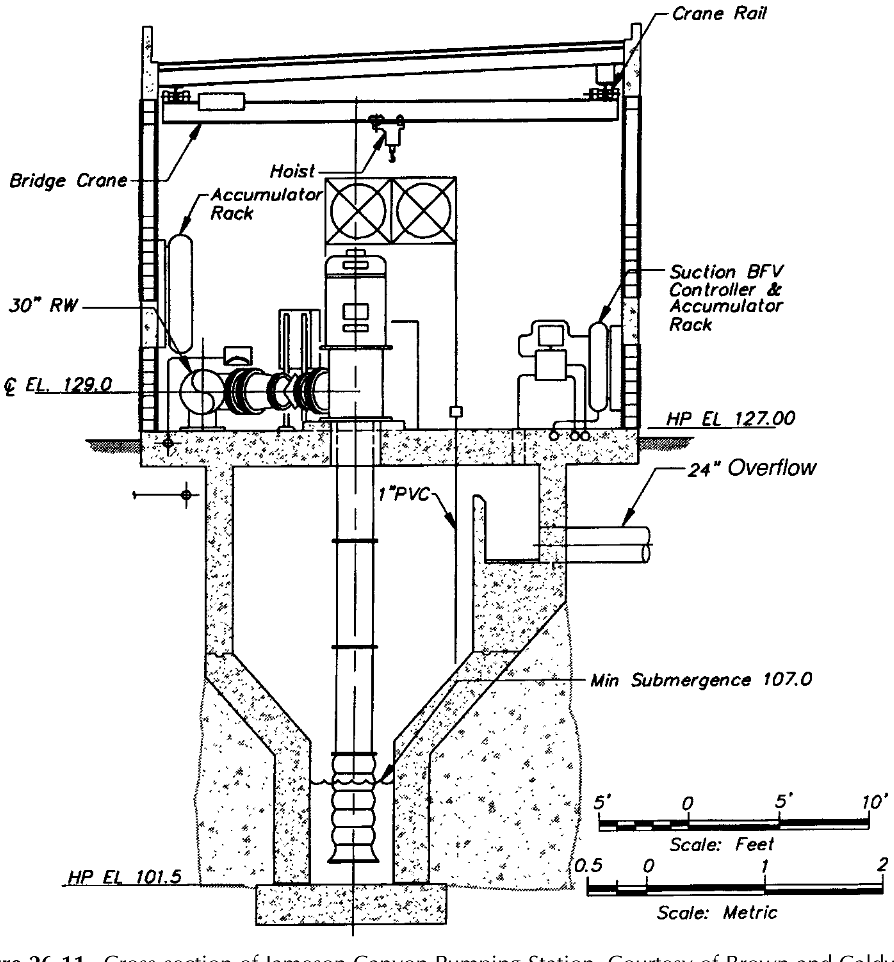


Figure 26-11. Cross-section of Jameson Canyon Pumping Station. Courtesy of Brown and Caldwell.

Using the Stoner Associates (now Advantica, Inc.) *LIQT-PC* [3] program, a transient analysis of pumping operations revealed that a high point in the transmission main approximately 5500 m (3.4 miles) from the pumping station effectively limited station discharge head at lower flows. Depending upon the operational mode at the water treatment plant (throttling or free discharge), the hydraulic profile was dominated by the high point for flows less than 657 L/s (15 Mgal/d) for throttling or 920 L/s (21 Mgal/d) for free discharge (see Figure 26-12).

The high point in the transmission main also would be the site of a vapor cavity of damaging proportions on power failure at maximum flowrates. The transient analysis showed that the stop-and-check ball valves at each pump must be controlled to close in 8 min on power failure to keep the pressures in the transmission main under control (see Figure 26-12). During power failure, water will bleed back through the pumps as the ball valves close. The pumps had to be specified to accept this reverse flow and spin backward without damage. Time delays are provided to prevent restart until the pump stops turning. A large number of air-vacuum valves were installed in the transmission main.

Critique

The decision to omit the ogee spillway for removing sediment is a matter of balancing the extra cost of a longer wet well against the possible deterioration of water quality caused by possible organics deposited with the solids. The likelihood of problem sediments was judged to be remote for Jameson Canyon, but for other raw water pumping stations, the inclusion of an ogee spillway should be considered. If the water is clean, some cost could be saved by substituting a flat bench for the sloping side walls.

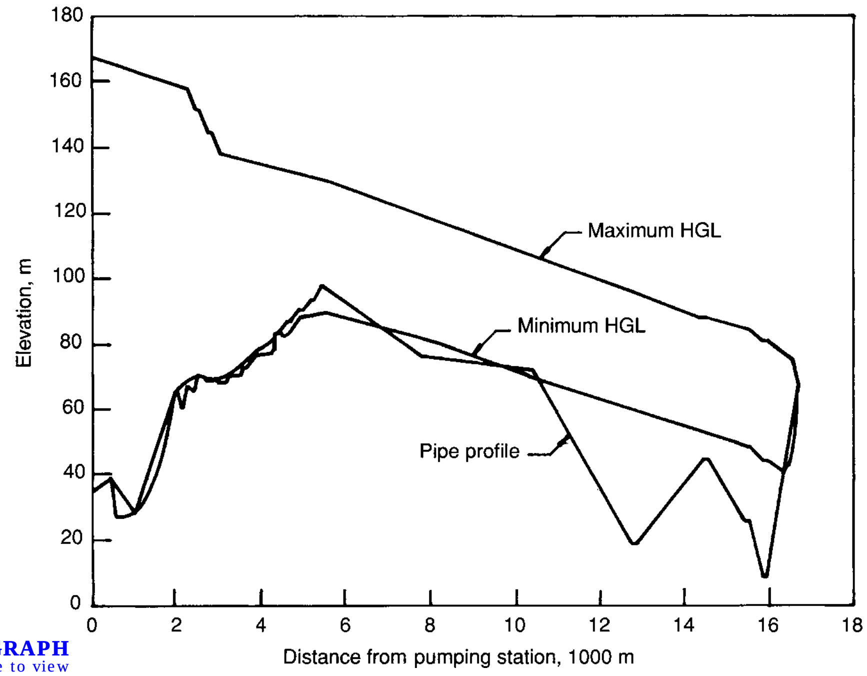


Figure 26-12. Minimum and maximum hydraulic grade lines and pipeline profile. From a Stoner Associates (now Advantica, Inc.) *LIQT-PC* transient analysis.

26-4. References

1. Wheeler, W., *PUMPGRAF*©. Contact William Wheeler at 683 Limekiln Road, Doylestown, PA 18906-2335 for this program.
2. *Hydraulic Institute Standards for Centrifugal, Rotary & Reciprocating Pumps,* 14th ed., The Hydraulic Institute, Parsippany, NJ (1983).
3. Stoner Associates. Now Advantica, Inc., 5177 Richmond Ave., Ste 900, Houston, TX 77056. www/advantica.biz.

Chapter 27

Avoiding Blunders

ROBERT L. SANKS

CONTRIBUTORS

Robert H. Brotherton
Fredric C. Burton
Patrick J. Creegan
Johannes deWaal
James C. Dowell
Fred A. Fairbanks
Philip A. Huff
William R. Kirkpatrick
Frank Klein
R. Russell Langteau
Jerry G. Lilly
Ralph E. Marquiss
Charles D. Morris
Raynold "Ray" Nickel
Constantine N. Papadakis
Carl W. Reh
William H. Richardson
Marvin Dan Schmidt
Arnold R. Sdano
Earle C. Smith
Charles E. Sweeney
LeRoy R. Taylor
Michael G. Thalhamer
David Walrath
Gary Z. Watters
Frederick R. L. Wise, Jr.

Mistakes in pumping station design are sometimes disastrous and always costly and troublesome. They can be minimized by (1) knowing and frequently reviewing the most common blunders so as to avoid them; (2) making periodic design reviews (see Appendix E) at various stages of design—in-house if experienced personnel are available but, if not, by engaging outside, experienced consultants; (3) acquiring extensive knowledge about the manufactured products and pumping station design (by attending equipment shows, examining many stations, and interviewing managers and operators); (4) using a good deal of common sense and care; (5) properly coordinating the work of the different disciplines and professionals; (6) making knowledgeable inspections during construction; and (7) giving warnings against hazardous operation in the O&M manual.

27-1. General

Some of the most common blunders and some ways to avoid them are given in this chapter. The list of blunders is abridged, however, so read Chapter 24 (especially) as well as Appendices E and F for more completeness.

One of the most common blunders—one that fits no particular category—is failing to maintain adequate written records of all significant conversations, letters, decisions, and calculations that pertain to the project design. The records must be neat, indexed, complete, understandable, and dated so that any engineer can interpret them. Careful recordkeeping is desirable for others who may wish to discover the secrets of a successful design and a necessity when personnel changes are made or when there is a subsequent arbitration or a lawsuit. The records should be kept neatly bound in, for example, one or more three-ring binders.

Another of the most common blunders—one that seems to appear in most or at least many pumping stations—is inadequate room for heads, hands, feet, wrenches, and rumps. Much of the maintenance and repair work requires space for a team of workers—not just one individual. An absolute minimum of clear space free from all protuberances is 0.92 m

(36 in.), preferably on three sides of the machine, and 1.1 m (42 in.) is better.

Many blunders seem so obvious. It does not even take an engineer to avoid them. It just takes thoughtfulness and a sense of responsibility. It helps to cultivate an attitude of asking, "How can this piece of equipment be maintained, disassembled, fixed, and reassembled, or replaced?"

Standard specifications and codes are identified only by their common abbreviations.

27-2. Site

Existing utilities and obstructions not shown on the plans. Obtain utility company information for the specific site. Prepare a good site plan and field check the design.

Location in floodplain without adequate protection. Interview old-timers about high-water marks, obtain government flood-hazard maps, and, if critical, compute flood elevations.

Inadequate street access and parking. Plan for access and enough parking space at the site (closed to public parking) for maintenance vehicles, including, if appropriate, a crane.

Nonconformity with setback and other planning and zoning ordinances. Review the regulations before selecting the site.

Site plan scale inadequate to show all details of site work. Normally, choose a scale of 1 in. = 50 ft or larger.

Wells located too close to sanitary hazards or property lines. The site plan should show the existing facilities. Follow the state sanitary codes.

Elevation of finished work incorrect. Insist on ties to at least two benchmarks. Field check the contractor's work prior to placing concrete for critical items.

Ground floor too low or floors without slope. Flooding or drainage problems are created and may violate sanitary codes. Have a good site survey and site plan, establish benchmarks, slope the floors to drains, and follow the state sanitary codes.

27-3. Environmental

Inadequate silencers on engines. Specify an acceptable noise level, say, 55 dB, at the lot line. Use a residential-quality silencer.

Fans too noisy. Use larger fans at lower speed.

Noisy machinery located within, say, ½ km (⅓ mi) of inhabited areas. The alternatives are: (1) relocating the pumping station; (2) using storage (except for wastewater) instead of standby diesel power; (3) obtaining independent, supplementary electrical power as a standby power source; (4) using submersible pumps; or (5) planning to soundproof the station at the outset of the design. Take ambient noise-level readings before construction for protection in possible lawsuits.

Sewer inlet with a freefall to the water surface. A freefall of wastewater into a wet well might (1) enhance odors, and (2) promote foaming and air entrainment. Consider Examples 12-1 and 12-2 for preventing ingestion of air into pumps, and, for both controlling air ingestion and reducing odors and release of toxic, corrosive gases, consider an approach pipe as shown in Figure 12-28.

Odor production within about ½ km (⅓ mi) of residential areas. The alternatives are (1) relocating the pumping station, (2) pretreating the wastewater, (3) planning for frequent housekeeping by adding hose valves (bibbs) for washing out odor-producing deposits, (4) sealing the wet well with manhole covers (see Section 23-1 for details and safety precautions for entry), or (5) adding odor-control facilities on the wet well exhaust.

Odor control. Near residential areas, all vented air may require odor control (see Section 23-2), which may well be the most expensive part of O&M. *Always* investigate present and future needs for odor-control facilities if the station is (or will be) near inhabited areas, and design it so that odor control can be added. When using odor-control systems such as carbon sorption, consider thermostatically controlled heating to prevent the freezing of moisture in the exhaust.

27-4. Safety

It is poor economy to skimp on items that have a strong potential for life-threatening or hazardous situations.

Ventilation

Inadequate ventilation for control rooms containing AFD equipment. Be sure to include the effect of the heat loads from AFD equipment in designing and sizing the ventilating and air-conditioning equipment for electrical control rooms containing adjustable frequency drive equipment. The mechanical and electrical engineers need to coordinate on ventilation issues. IGBT equipment does not produce as much heat load as the older types of AFD drives used many

years ago, but the issue still needs attention to avoid embarrassing redesign and rework effort.

Inadequate ventilation system for wet well. For wastewater wet wells that must be entered frequently, 12 air changes per hour (AC/h) is a minimum to meet most codes. Nevertheless, 12 AC/h can be hazardous, and an exchange rate of 20 AC/h is a more reasonable minimum. The California State Department of Health recommends 25 and the Arizona Department of Health requires 30 AC/h. Intermittent ventilation, which is allowed by some codes, including Ten-State Standards [1], is considered here to be entirely inadequate where human life is at stake. The scavenging of wet wells is imperfect at best, and the energy cost saving is not worth the risks. Consider omitting powered ventilation for wet wells by ensuring that there is no need for personnel to enter routinely (but note that a vent is required).

Wet well fans. Power air into the wet well at the top with high-energy diffusers, and power the exhaust air out at a low elevation using slightly more exhaust volume than supply volume to create a small negative pressure (see Section 23-2). (Opening or closing doors defeats the purpose of ventilation *by a single fan* whether the air is powered in or out.) Design for Class I, Division 2 (NEC Code), and use corrosion-resistant materials (such as fiberglass) and explosion-proof construction in the exhaust system. Wet well exhaust is very damp and corrosive. Protect screens from freezing.

Inadequate ventilation for dry well. Ventilation in dry wells of either water or wastewater pumping stations is required to limit the humidity and accumulation of toxic or explosive gases. (Explosions have occurred in water pumping stations.) Six air changes per hour for continuous ventilation or 30 complete air changes per hour for intermittent ventilation is a practical minimum. Intermittent ventilation may be hazardous because it might not prevent the accumulation of explosive or toxic gases.

Other Considerations

Inadequate headroom. Design for 2.1-m (7-ft) clearance under pipe flanges, hand wheels, and other protuberances. But if such a clearance cannot be met, pad the protuberance and hang a row of light chains around it as a warning. However, locate oil or water traps low enough so they can be serviced easily.

Obstructions in walkways. Lay out dry well and other floors to provide easy access to maintenance points (such as packing glands, oil and grease points, and pressure gauges) so that workers need not climb over piping or other obstructions. A good rule is to require a 1.07-m (42-in.) clearance (horizontally) between pumps, motors, valves, etc., and any obstruction.

Ladders and circular stairs. Avoid ladders if possible, because (1) emergency egress is impeded, (2) personnel can carry tools only with difficulty, and (3) disabled workers cannot be carried at all. Observe OSHA rules for landings (required at maximum intervals of 3 m or 10 ft), ladder cages, nonskid treads, and toe plates. Make ladders easy and safe to enter.

Circular stairs are only marginally better. Straight stairways with landings conforming to local building codes are justifiable, considering safety, convenience, and daily use for 20 years or more. If a compromise must be made, consider ship stairs for minimum run length (one horizontal to two vertical). Check local building codes for double-access requirements and for stairs to augment elevators.

Guards. Install guards around dangerous rotating machinery, belt drives, and other moving machinery.

Rescue. Provide a means for moving a helpless worker to the building exit. Consider the need for gas masks and self-contained breathing apparatus (always in pairs) at wastewater pumping stations.

Codes. Become familiar with design standards such as those published by the Hydraulic Institute (ANSI/HI series) and with the safety regulations of such codes as OSHA, NFPA, and NEC.

27-5. Hydraulics

Sumps and wet wells inadequate or mismatched with pumps. Review Chapter 12. If the design differs from ANSI/HI 9.8, Chapter 12, or a design known to be successful, consider model tests. Model tests are required (if ANSI/HI 9.8 is followed) under certain circumstances. See Section 12-4, Prosser [2], Dicmas [3], and especially ANSI/HI 9.8-1998 and ANSI/HI 1.1-1.5-2000.

Pipe roughness overestimated. If the actual friction headloss is less than the estimate, the actual discharge will be greater than predicted. The greater discharge increases the horsepower requirement and, thus, overloads the pump and motor. Design for the code-specified roughness (e.g., in the Ten-State Standards [1], $C = 120$ for lined and plastic pipe or $C = 100$ for unlined pipe) or design for the roughness expected after, say, 20 years of service. Then redraw the station curves for new pipe (e.g., $C = 145$), find the new operating point, and specify a pumping unit that can operate at AOR for all of the pumping

conditions between the two extremes and at POR for the most common pumping conditions. For final calculations, use the actual, true inside pipe diameter for headloss and flow calculations.

Before accepting the completed product and making the final payment to the contractor, run tests with calibrated pressure gauges (and preferably with calibrated flowmeters) to ensure pumps and impellers are suitable for minimum, maximum, and average flowrates.

Hydraulic resonance. The presence of hydraulic resonance within a pumping system can result in very high-pressure pulsations, high vibration, loud noises, and premature failure of the pump and/or piping equipment. The best way to avoid hydraulic resonance is to calculate the expected resonance frequency while the piping system is being designed and make sure that it is well away from the vane-pass frequency of the selected pump. Corrective measures commonly employed are:

1. Change the size and stiffness of the piping system to change the resonance of the piping system.
2. Block out the operating speed range where the phenomenon is experienced.
3. Change the vane-pass frequency by changing the number of pump impeller vanes.
4. Change the vane-pass frequency by changing the pump operating speed.
5. Add an accumulator to the piping system to change the resonance of the piping system.

Prerotation in wet wells. The vortexing that results from prerotation entrains air, reduces flow, and leads to vibration and noise in pumps. See Sections 12-1 through 12-8 and 27-6 for advice about wet wells.

Suction inlets too small for rated flows. Cavitation and vibration will occur and the pump will not discharge the designed capacity. The "average" velocity at the bell mouth of the pump intake should not exceed 1.7 m/s (5.5 ft/s), and the authors believe the velocity (particularly in trench-type pumping stations) should not exceed 1.1 to 1.2 m/s (3.5 to 4 ft/s). The velocity in a pump inlet pipe should not exceed 2.4 m/s (8 ft/s), but see also ANSI/HI 9.8-1998.

Pump volutes inadequately submerged. Always place the low water cutoff above the pump volute or, more conservatively, above the stuffing box.

Suction inlets inadequately submerged. The required submergence is based on empiricism, and opinions vary. Probably the best rule is given by Equation 12-1, taken from ANSI/HI 9.8-1998.

Gratings and other devices above the inlet can reduce the required submergence, but unless they are smooth and slope steeply, they collect debris in wastewater wet wells. See Figure 12-13. Model studies are recommended.

Vertical pipes connected to invert of manifolds. The solids in raw water and wastewater can collect in the riser and, given time, plug it. Connect the riser to the side of the manifold with a short length of horizontal pipe.

Float switches untethered in turbulent wet wells. Untethered float switches tangle, so attach them to a weighted chain (which permits easy removal) or to a vertical pipe. Also consider other sensors described in Section 20-3.

Pipe profile incompatible with hydraulic profile. Plot both profiles on the same drawing for comparison. Avoid high spots in pipe profile—especially in wastewater service where the height of the pipe should never be above the mirror image of the hydraulic profile (see Figure 7-1). In water service, the pipe profile should never be above the hydraulic profile by more than 7 m (23 ft), and then control equipment (e.g., air-vacuum valves) must be installed at high points. Use the services of experienced personnel or consultants for the design of air release valves.

Water hammer. Prevent column separation in the force main at all costs. A complete transient analysis should always be made for all systems exceeding, say, TDH $=$ 12 m (40 ft), $Q = 30$ L/s (500 gal/min), and $v = 1.5$ m/s (5 ft/s). Beware of water hammer with an expulsion of air at the top of wells. Make certain that thrust blocks and anchors at bends and elbows can resist both static head and surges.

Wastewater systems are especially vulnerable to water hammer because of buildup of grease and foam in control devices such as air valves. See Section 5-7. Pilot-operated surge-control valves for wastewater are unreliable due to the potential clogging of pilot lines, so avoid them in wastewater pumping stations. Avoid water hammer in wastewater force mains by controlling the pipeline profile and/or by adding flywheels to drivers. Consider hydropneumatic tanks for wastewater only as a last resort.

Resonance. Avoid it in design. If discovered after construction, ameliorate it. See Sections 22-10 and "Hydraulic Resonance" in Section 25-5.

Air chambers close to swing check valves. Never put air chambers close to swing check valves because the short water column can accelerate very quickly and cause valve slam with exceedingly high pressures that can explode pipe or break valve casings.

Cleaning and supercritical velocities. Water issuing under sluice gates, flowing freely down approach pipes, or flowing down inclined ramps flows at supercritical velocity. In this state it cannot be turned,

redirected, or deflected with ease, as can be done at subcritical velocities. Striking any obstruction (e.g., bolt head, nose of a flow splitter, etc.) causes an eruption. Be very cautious. Insist on model testing if there are any doubts whatever.

Backflow Prevention

Cross-connection installed. Use reduced-pressure-principle backflow preventers or, better, completely separate potable water from the station (seal water, washdown) water supply with a positive air gap.

Applicable plumbing code not checked. Under some codes, a reduced-pressure backflow preventer is not an acceptable substitute for an air gap.

Backflow preventer not equipped with drain. Under some conditions, the backflow preventer can release a large stream of water that requires a high-capacity drain that can inundate the dry well, so exhaust the vent pipe outside the building above flood level.

Backflow preventer installed at too low an elevation. Always install backflow preventers above any possible flood level.

Potable water. Water downstream from a backflow preventer is subject to contamination and is not potable. Pipe only potable water to sinks.

Potable water faucets with hose threads. Hoses are possible sources of cross-connection and contamination. Do not install hose valves (bibbs) on potable water pipelines.

Hose valves (bibbs) without warning signs. Place warning signs indicating that the water must not be drunk.

27-6. Wet Wells

The geometry of the wet well and its associated piping and entrance conditions profoundly affect pump performance. Hydraulic conditions detrimental to pump performance include:

1. Asymmetrical velocities in the approach to the pump intake
2. Nonuniform or asymmetrical velocity distribution in the pump intake
3. Swirling or prerotation of flow at the impeller
4. Surface and subsurface vortices.

Asymmetrical flow to a pump intake is likely to produce swirling and/or asymmetrical velocity distribution in the intake throat. Asymmetrical velocities in the intake throat create a higher load on one side of the impeller, bend the shaft, and put extra load on bearings and couplings resulting in rough operation, vibration, and loss of head and capacity. These effects worsen as pumps get larger and as the specific speed increases.

Swirling changes the angle of attack on the impeller vanes, reduces head and capacity, and decreases efficiency. An angle of 5 degrees from axial is the maximum allowed in ANSI/HI 9.8.5.6-1998. Swirling currents near a pump intake can degenerate into vortices. The pressure in the core of a vortex is reduced and air (or other gases) coming out of solution can cause noisy operation and vibration. When a vortex is severe, it results in cavitation that quickly erodes metals.

Strong surface vortices entrain air bubbles that collapse as they pass from low to high pressure zones across the impeller vane, thus creating noise, uneven torque, erosion, and undue stress on shafts, bearings, wearing rings, and couplings. A small amount of air can cause a large decrease of capacity, head, and efficiency. Milder surface vortices tend to become smaller but more intense as they enter the intake.

Strong subsurface vortices, formed as liquid separates from walls or floors, are often present and difficult or impossible to detect in the field. Normally, the bubbles in subsurface vortices consist of air that comes out of solution due to the decreased pressure in the core. Subsurface vortices are just as damaging as surface vortices—sometimes more so. If vapor bubbles can form, they cause rapid erosion. In fact, the rate of erosion can be reduced (although performance is also reduced) by introducing air deliberately to form large bubbles that do not completely collapse (see Section 10-4).

Designers should try to visualize the flow patterns likely to develop in a sump. Although mental pictures are only approximate, they can nevertheless serve to identify potential problems. Bear in mind several phenomena that govern the flow of water and solids including:

1. Low to moderate fluid velocities past pump intakes tend to wash swirls downstream and prevent them from organizing into vortices.
2. Water emerging as a jet from a pipe into a relatively quiescent pool tends to continue to flow in a straight jet that spreads only about one unit per eight units of travel. The center of the jet tends to retain its original velocity for a considerable distance, and eddies are generated along the edges of the jet.
3. When a jet of water is forced to turn, swirls are created and some energy is lost. Swirls tend to

persist and may develop into vortices at pump intakes.

4. Vortices easily form in quiescent water both at the water surface and under water. Mild currents tend to wash swirls downstream and to prevent them from organizing into vortices.
5. In quiescent test pits, pumps can lower the water level to the mouth of the suction bell, but during pump-down in a trench where currents are confined by adjacent walls and floors, end pumps tend to lose prime at submergences less than about 0.3 to 0.5 *D* due to adjacent, large, air-breathing vortices. (At Kirkland Pump Station, prime could not be maintained for more than a minute or so at a submergence of less than 0.55 *D*.) Submersible wet pit pumps without nozzles, for example, cannot reduce the water level below the top of the volute.
6. Moving deposited solids, even slowly, requires fluid velocities no less than about (1) 0.2 m/s (0.7 ft/s) for light organic sludge, (2) 0.3 m/s (1 ft/s) for heavier sludge, and (3) 0.6 m/s (2 ft/s) for grit. Much greater fluid velocities are needed to scour solids with reasonable celerity.
7. A free fall (even if considerably less than 0.3 m or 1 ft) into a pool generates masses of air bubbles that are driven deep, easily captured by pump intake currents, and drawn into pumps, thereby reducing capacity, head, and efficiency. The presence of air in pumped fluid can be a major source of damage to bearings and shaft seals. Air bubbles can be kept out of intakes by moving the pump intake a sufficient distance from the waterfall (see Falvey [4]) or by preventing the waterfall from penetrating deeply into the pool (e.g., by a horizontal baffle to produce horizontal currents).

Failures

The phenomena above are useful for explaining the shortcomings of the wet wells described below and for predicting the behavior of any wet well.

Portland Airport Fire Pumping Station

Diesel engine-driven pumps (for fire suppression in two huge airplane hangars), placed in the wet well of Figure 27-1, vibrated excessively and failed to meet their required flow rate. The wood baffle and steel WF beams did not exist in the original design. Without them, the inlet jets would obviously strike the back wall unabated, and rebound in circulating currents that would cause serious swirling. The fire marshal refused to approve the system. A wooden baffle wall was recommended to confine the pump intakes in a "trench" to separate the destructive currents from the pumps. The wall (actually made entirely of steel) was installed (over the objections of the pump manufacturer, by the way), after which the pumps did deliver their rated capacity and the project was approved. Nevertheless, the sump is deficient in other respects, such as excessive floor clearance under pump intakes and poor inlet design, to name two.

Lancaster Reclamation Plant Lift Station

In the cross-section shown in Figure 11-12, the baffle beside the sluice gate was intended to cause sufficient fluid velocity along the sloping floor to move grit to the pump. Under the best circumstances, however, the flow to the pump would occur through a section under the baffle about 0.46 m (1.5 ft) high by perhaps 1 m (3 ft) wide for a velocity of about 0.3 m/s (1 ft/s)—insufficient to move grit. And so it proved. Grit stacks up to the top of the baffle. Submersible pumps with bypasses to stir the contents, a backwash pipe from the force main as in Figure 17-22, or a mechanical mixer are possible remedies.

Sixty-third Avenue Pumping Station

The wet well is an annular ring of 15.4 m (50 ft) OD and 8 m (26 ft) ID with a floor sloping nearly 7 degrees toward the inner wall. Four horizontal pumps are crowded into the central dry pit with their intakes evenly distributed around the annulus.

As the fluid velocity around the annulus is very low, the wet well acts as a sedimentation basin. There were no provisions for cleaning it, and on a visit in 1992, about 3 yd^3 of grit and sludge were piled in a bank over 2 ft high. The stench within the pumping station was unbearable. Sludge must be removed manually.

Durham Pumping Station

The wet well, originally designed to discharge a total flow rate of 2400 L/s (54 Mgal/d), is shown in Figure 27-2 as a two-cell rectangular basin with divider baffles to isolate either cell for cleaning. The pumps were noisy. Impeller noses quickly eroded, so impellers had to be replaced frequently. When a capacity of 6100 L/s (140 Mgal/d) was required, a model study proved that the short cascade (less than 0.3 m or 1 ft in the prototype) from the distribution box into the wet well caused great masses of air bubbles

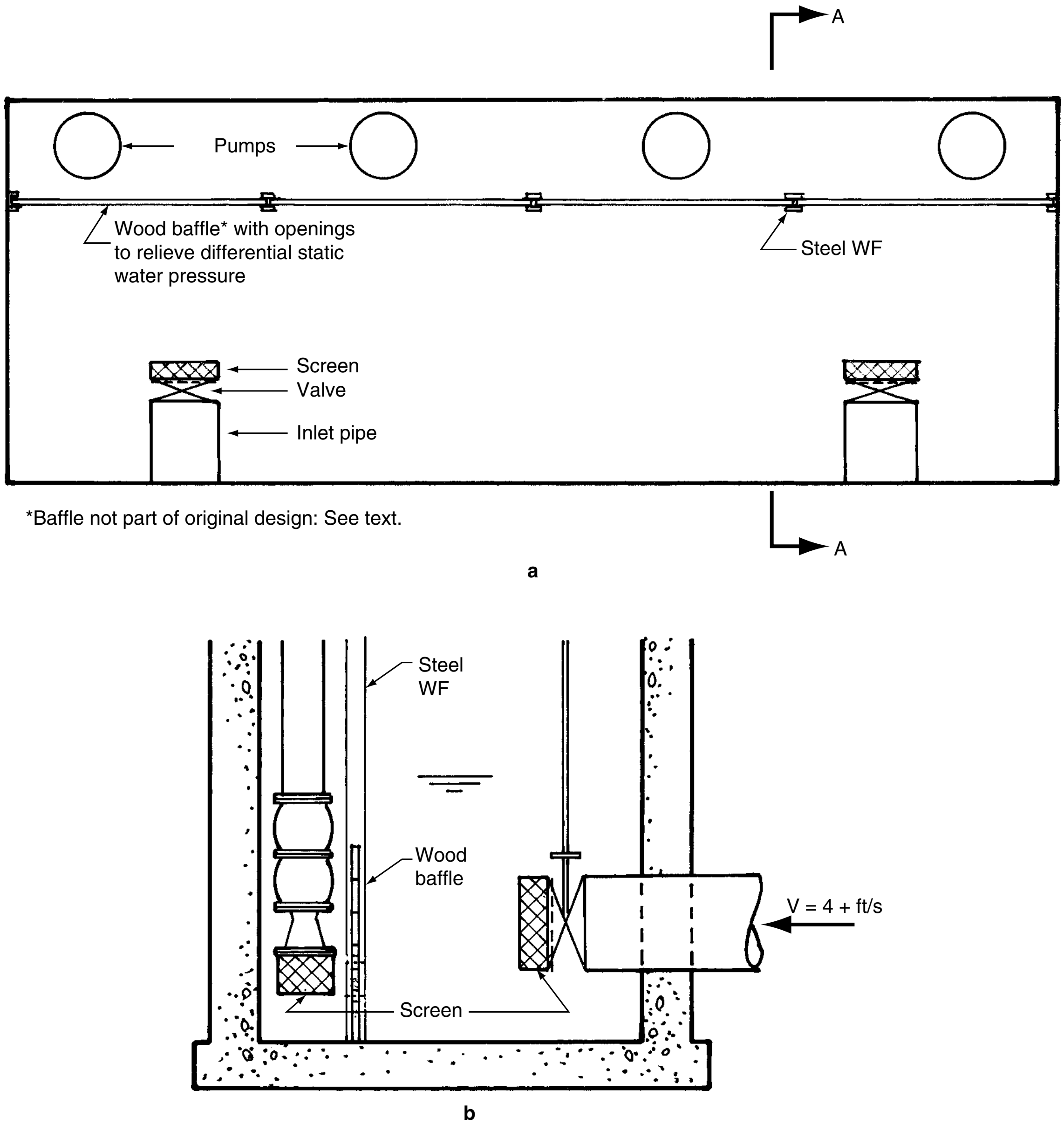


Figure 27-1. Sump for fire-suppression pumps at Portland, Oregon, airport. (a) Plan; (b) Section A-A.

to be driven to the floor where the pump intake currents captured them. See Phenomenon 7 above. There were severe surface vortices. By installing the retrofitted inlet box shown by the dashed lines in the figure and keeping the water level well above its floor, water is discharged horizontally from the inlet box so that bubbles, also move horizontally and float to the surface. No bubbles entered any pump intake after the inlet box was installed—not even at the highest flow rate. Pumps ran smoothly. The wet well still traps sludge and scum, however, and it is difficult to clean.

Oak Street Pumping Station

The wet well was designed in 1972 for a capacity of 55 Mgal/d. The layout involved a wet well and dry well inside a caisson, somewhat like that of Figure 9.8.4C in ANSI/HI 9.8-1998. The vortex breaker was a group of horizontal aluminum "planks" separated by 75 mm

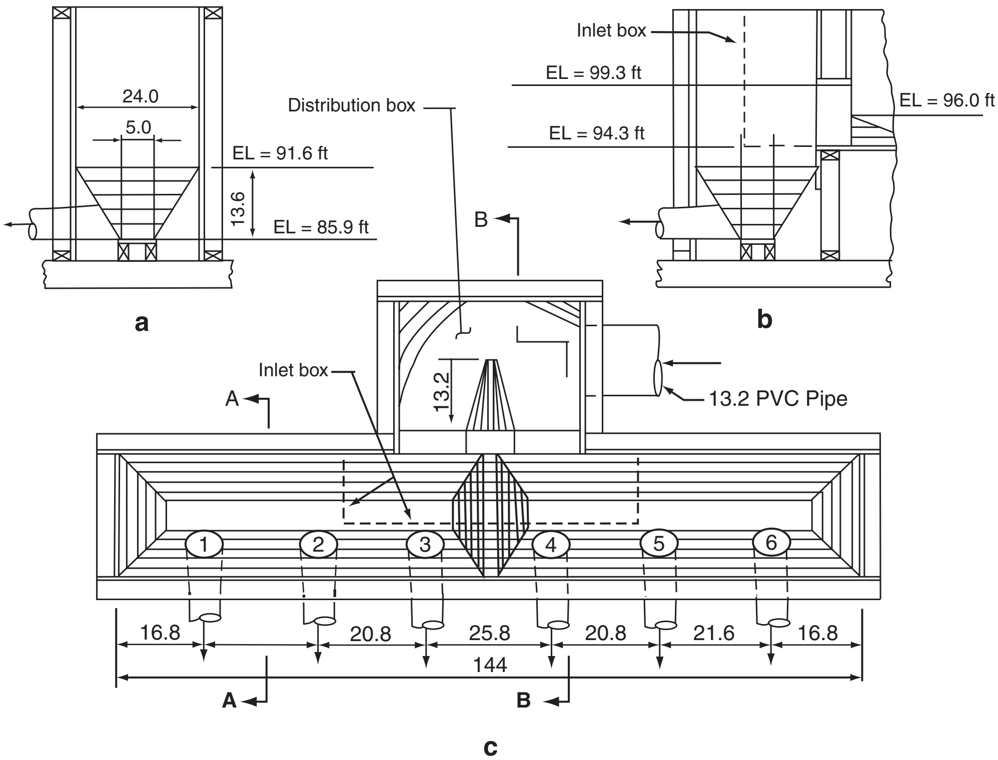


Figure 27-2. Hydraulic model of Durham raw wastewater pump station. Dimensions are model inches. Elevations are prototype feet. Model prototype ratio is 1:5. (a) Section A-A; (b) Section B-B; (c) plan. Courtesy of ENSR International.

(3 in.). The small, short, shallow wet well resulted in: (1) violent turbulence, (2) almost unbearable pump noise, and (3) grossly inadequate pump capacity. Five pumps together could pump no more than 75% of the design capacity of four pumps. The low performance was most noticeable in the outer two pumps, which also needed frequent rebuilding. Debris on the vortex breaker (Figure 27-3) had to be removed manually—a hazardous, filthy task, fortunately an infrequent activity due to the turbulence.

When a higher capacity was desired, a model test confirmed the violent turbulence, and the wet well was found to be completely inadequate without significant improvements. One solution was a group of vertical pipes in the path of the inlet to reduce entrance velocity and turbulence, but the ultimate station capacity would be limited to 60 Mgal/d, and cleaning would be a problem. Another solution was the conversion of the entire caisson into a wet well containing submersible or column pumps, but cleaning the wet well would remain a problem. The solution adopted was a trench-type wet well designed for a present firm capacity of 65 Mgal/d but expandable to 88 Mgal/d. The adjacent caisson was converted to a larger dry pit by demolishing the interior wall to make room for three large and two small, vertical nonclog pumps.

The size of the wet well must be large enough to suppress excessive turbulence, but unfortunately at this writing, *no guidelines exist for the relation between inertia of the influent and size and shape of the wet well.* Designers should be cautious and conservative in allocating sufficient volume to suppress excessive currents from inlet conduits.

General

Aping the geometry of traditional designs is no guarantee of success. Triplett, Fletcher, and Grace [5] emphasized the problems concerning some established pump sump design criteria with their statement in 1988 that *"approximately 50% of all sumps for flood-control pumping stations designed and constructed by the Corps [of Army Engineers] needed some sort of post-construction modification to improve*

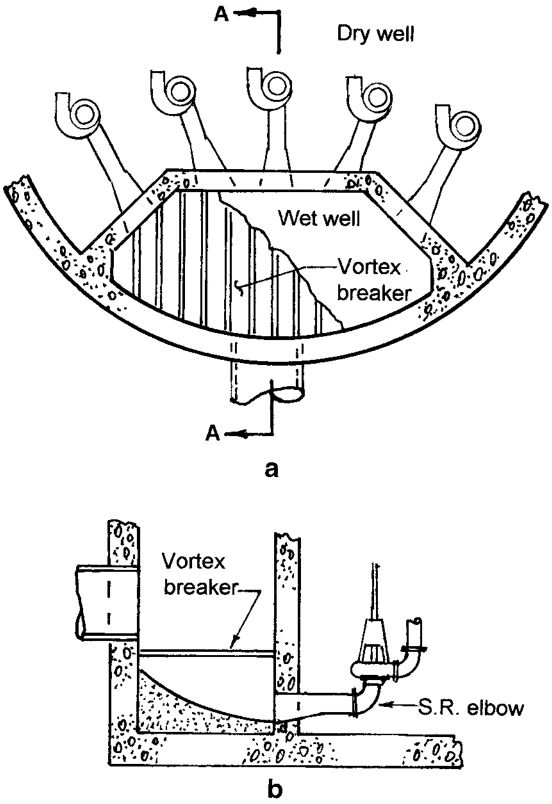


Figure 27-3. Schematic diagram of Oak Street Pumping Station. (a) Plan; (b) Section A-A.

the flow conditions to the pumps. These modifications were costly and in most cases did not correct the problem but improved the conditions only slightly." Criticisms such as this should jar engineers out of any complacency toward wet well design (see also Karassik et al. [6]).

Flow Splitters and Fillets for Trench-Type Wet Wells

Flow splitters prevent floor vortices in normal pumping operations. During pump-down for cleaning, (1) they improve the hydraulic radius, (2) Froude numbers do not deteriorate with wet well length so quickly, and (3) sequent depths are greater, so it may be unnecessary to drop the floor at the last pump. They ought to be installed in wastewater trenches wider than 1.2 m (4 ft) and, without question, in trenches 25% wider. They are costly, mostly because of the labor of installing them. Use stainless steel plate, as the cost of material is only a small fraction of installed cost. In spite of the cost, always extend flow splitters to the top of the ramp because cleaning is unsatisfactory if flow splitters terminate at the toe of the ramp—a serious blunder.

In V/S pumping stations, flow entering the sump at pump-down is only slightly supercritical, so the nose at the top of the ramp can be short and relatively blunt. In C/S pumping stations with steep approach pipes, however, the entering fluid velocity at the top of the ramp is well above critical. A blunt nose breaks the flow into a spray and a satisfactory hydraulic jump never forms. Consequently, the nose must be very gradually tapered from zero at the top of the ramp to full size at somewhere near the mid-height of the ramp.

Fillets, always advantageous, should also be installed in trenches wider than 1.2 m (4 ft). They prevent sidewall vortices during normal pumping operations. Like flow splitters, they improve the hydraulic radius at pump-down during cleaning. If made of gunite, reinforced, screeded, and troweled smooth, they are reasonably inexpensive. Terminate them with a blunt nose at the top of the ramp for both V/S and C/S pumps, because the entering flow from a round pipe does not impact the nose.

Cleaning and Pump-Down Problems in Trench-Type Wet Wells

Supercritical flow. Any interference with water moving at high speed causes a "rooster tail" that saps energy and grossly interferes with downstream flow. If a sluice gate is to be used during cleaning, taper the noses of the flow splitter and the fillets.

Hydraulic jump incomplete. If the hydraulic jump stops short of last pump, water is probably circulating behind the pump or underneath it. Block openings behind the pump and install a fore and aft vane beneath the bell, as shown in Figure 12-20.

Cleaning time excessive. If cleaning time exceeds three or four minutes, select a lower rate of inflow and/or operate two or more pumps to reduce the depth of water quickly to about $\frac{2}{3}\,D$ above upstream bells, then continue to use the last pump to achieve complete pump-down.

Grease on walls. Use a commercial sewer washer or, better, pressurize 2.5 to 4.4 L/s (40 to 70 gal/min) of clear water from an air break tank to a pressure of about 690 kPa (100 lb/in.^{2}) with a regenerative turbine or a multistage centrifugal pump (see Chapter 11). Use a 9- to 13-mm ($\frac{3}{8}$- to $\frac{1}{2}$-in.) nozzle. If needed to reach the grease, attach the nozzle at the end of a 25-mm (1-in.) steel tube.

27-7. Pumps

Specifying pumps. See Chapter 16, Section 27-12, and Appendix C.

Improper selection of the type of pumps and pumping station. A design concept that may be entirely suitable for one agency or application may be entirely unsuitable for another, even if the stations happen to serve the same hydraulic conditions. For example, the use of a submersible wastewater pump design may be appropriate for an agency that already has a significant number of such facilities and has the attitude, experience, personnel, and equipment to maintain them. Such a pumping station may be quite unsuitable for an agency that has no experience with such a facility and lacks the equipment and personnel to service and maintain it. A dry well may be the best option for the second agency in spite of the additional first-time construction cost. In summary: *All aspects of the facility*—initial construction cost, power costs, ease of operation and maintenance, availability of equipment and skilled maintenance personnel, and so on—must be considered in evaluating the various types of stations and selecting the most appropriate.

Large submersible pumps. Large [more than 75-kW (100-hp)] pumps may have very high operation and maintenance (O&M) costs if the pumping station is not tailored for ease in handling big pumps. Ease in handling large submersible pumps is particularly important for wet well installations due to the long, heavy power cables that (because they cannot be handled manually) make large submersible pumps entirely different from smaller ones in their requirements for trouble-free removal. There are many horror stories of power cables cut on sharp edges or flexed beyond their limit; pumps jammed on guide rails that are thereby bent; the need for massive mobile cranes to pull pumps located an excessive distance from a parking area; long discharge pipes that vibrate excessively; pumps dropped by inexperienced crane crews; and even of a discharge elbow pulled out with the pump. In addition to these scenarios of woes, large pumps (especially those of moderate quality) tend to break down more often than do smaller pumps. Many utilities seem to agree that large submersibles should be confined to dry pits where they can be serviced in place. In at least one large utility, however, wet well installations of submersible pumps are favored regardless of size (see the subsection "Submersible Pumps" in Section 25-6).

Cable attachment details omitted. Lack of foresight in planning for the removal of submersible wet pit pumps (especially those over 75 kW or 100 hp) can make removal difficult and expensive and may even require entry in confined hazardous spaces (with all the associated personnel and safety requirements). Discuss the removal of the pump with the manufacturer and the utility, decide what method is to be used, and tailor the facilities to make the job as easy as possible. Detail the method in the O&M manual, and describe the cable attachments in the specifications and show them on the plans. See the subsection "Submersible Pumps" in Section 25-6. It is attention to details such as these that makes the difference between easy and difficult maintenance and between good and bad engineering.

Unreasonably conservative estimates of flow rates and pipe roughness. Fear of insufficient capacity for future flows often leads to the most frequent and worst blunder of all—choosing pumps and pipes too large for the actual flows to be encountered. The results are (1) pumps that operate at "runout" (too far to the right of the BEP), exceed their capacity, vibrate, and cavitate; and (2) motors that are overloaded and therefore overheat. Pumps and motors in such conditions have short lives, and parts must frequently be replaced.

Avoid this blunder by choosing the pumps and impellers for operation within the AOR at low (as well as high) flows, for smooth pipe as well as aged pipe, and for flooded wet wells (with lowered static lift) as well as for low water levels. In lift stations with multiple pumps, for example, characteristically only one pump operates most of the time—and it does so at substantially reduced total head. So the pump should operate in its POR (near its BEP) for that condition. Also select pumps and motors to operate safely over the full range of conditions to be encountered. Low flows are, if anything, more critical than high ones. See Figure 10-27 and 10-30. If future flows will be considerably larger than present ones, try the following in the order listed: (1) install impellers with smaller diameters now and replace them with larger ones when needed; (2) leave space and hook-ups for another, future pump; or (3) plan on replacing present pumps and motors with larger ones when needed. Piping must be satisfactory for both the present (with fluid velocities high enough to transport solids) and the future (with friction headloss within bounds). Hazen-Williams C values for lined steel, ductile iron, and plastic pipe should be in the range of 130 to 150 for normal operation (see Sections 3-2 and 3-3). Future flows might require a second, parallel force main.

During start-up, always test both suction and discharge pressure at each pump for the widest range of conditions encountered. If necessary, change or trim impellers to suit the true system requirements and to reduce noise and vibration (an embarrassment to the designer). Test motors for current draw to be sure they

are not overloaded at any condition. Make sure the ventilation system can adequately cool the motors.

Variable-speed pumps. Never allow variable-speed units to operate at speeds or flowrates lower than recommended by the pump manufacturer. If flows are too low, add a smaller unit or program the controller to operate on a start-and-stop, fill-and-draw basis. If a pump operates consistently at low speed, turn it to high speed once or twice per day to clear it of rags and heavy sludge.

Net positive suction head (NPSH). Always make sure NPSHA is at least 1.35 times NPSHR, and also never less than 1.5 m (5 ft) more than NPSHR (see Section 10-4 and Figure 10-13).

Pump efficiency poor. Reliability and ease of maintenance are of prime importance, but search diligently for the best pump because an increase of only 2 or 3% in pump efficiency returns a startlingly high present-worth cost saving.

Pump and impeller selection poor. Avoid the largest impeller in a family of impellers because of vibration and reduced bearing life. Try to use a diameter that is no more than 90% of the largest diameter. Select a pump for good efficiency above the expected head range so that the heads "bracket" the highest efficiency of the pump.

For V/S units, specify (or select) pumps so that the operating point lies to the right of the best efficiency point (BEP). Thus, the pump selected will be the smallest applicable and will operate through the zone of best efficiency at the various normal (reduced) speeds. But specify (or select) pumps with the operating point at or near the BEP for C/S pumps. If a greater capacity is anticipated, specify the future operating conditions.

Relationship between impeller, speed, flow, head, and power incorrect. When calculating flows (which vary as revolutions per minute) for reduced speeds, also calculate the head (which varies as revolutions per minute squared) to find a new point on the pump H-Q curve. Draw the new pump H-Q curve through several such points to an intersection with the station H-Q curve to find the true flow and head at the reduced speed. To avoid mistakes in applying the affinity laws, calculate the power from head, flow, and pump efficiency—not from the relationship between power and revolutions per minute cubed. Note that manufacturers of V/S pump drives often assume—erroneously—that power varies as revolutions per minute cubed, and, hence, their curves of driver efficiency are inaccurate (see Section 10-3, Example 10-3, and, especially, Figure 15-4 and Section 15-3).

Excessive radial loads. Avoid applications where pumps operate for extended periods near the shut-off head. Radial forces developed in the casing and acting on the impeller at low speeds (see Figure 10-18) cause rapid bearing wear and damage the wearing rings, shaft sleeves, and packing. If such operation is necessary, specify custom-made pumps with heavy-duty shafts and bearings.

Cavitation damage. Ascertain the cause and correct it if practical, or replace the impeller with one made of a more cavitation-resistant material. See Table 10-3.

Machinery vibration. Test pumps for vibration during factory pump testing, but be aware that the foundation or supports in the field are different and these affect vibration. Support vibrating machinery (such as engines) either on (1) solid, well-grouted foundations as explained in Section 12-10, or (2) on spring isolators, and isolate piping from vibrating machines with vibration isolators (see Chapter 22 for a discussion).

Noisy pump. Pump noise can indicate cavitation, resonance within a pumping system, or excessive turbulence at the pump intake. An oscillogram from a piezometric pressure cell with a high-frequency response can usually determine whether the cause is resonance. Avoid resonance by calculating the expected resonance frequency while designing the piping system and ensure that the frequency is far from the vane-pass frequency. Corrective measures include:

1. Block out the operating speed range where resonance occurs
2. Change the vane-pass frequency by changing the number of pump impeller vanes
3. Change the vane-pass frequency by changing the pump operating speed
4. Add an accumulator to the piping system to change the resonance of the piping system
5. Change the size and stiffness of the piping system to change the resonance of the piping system.

Inadequate or unclear pump performance specifications. Manufacturers differ in their presentation of published pump performance data. Hence, it is important to define the conditions of performance clearly so that competing products can be compared on the same basis. One basis for comparison is to specify energy, head-capacity, and efficiency criteria for the complete pump, for example: (1) flange-to-flange for dry well pumps, (2) pool-to-discharge flange for wet well pumps, or (3) pool-to-discharge elbow outlet flange for submersible pumps.

Single-vane impellers. A single-vane impeller is asymmetrical, and no amount of machining can balance it dynamically in water. Some manufacturers

can and do trim these impellers; others do not. As alternatives to trimming, (1) consider V/S drives unless an unaltered impeller can be found to match the flow conditions, or (2) use a symmetrical impeller, which can be trimmed and balanced more easily.

High pump speeds. Avoid high speeds (especially for wastewater pumps) because wear is a function of the second or third power of speed. Use the slowest practical speed (such as 705 to 1150 rev/min) to reduce maintenance costs.

Impellers improperly chosen for deep well turbine pumps. Because shaft stretch becomes important for deep pump settings greater than about 30 m (100 ft), specify closed impellers and consult the pump manufacturer or a specialist.

Oil-lubricated deep well pumps. In general, oil lubrication should not be used in pumps for potable water.

Prelubrication system omitted in lineshaft deep well turbine pumps. A system for supplying water to the bearings during starts is required. Consult the pump manufacturer for assistance in the design.

Light-duty pumps. Many pumps with attractive prices are too lightly constructed for constant or severe service. Require rigid construction. For example, shaft deflection at the packing gland should not exceed 0.05 mm (0.002 in.), nor should deflection at wearing rings (outboard side of impeller) exceed about 0.20 mm (0.008 in.). A custom-designed pump may be needed to meet these requirements. Even sump pumps should not be light-duty machines.

Mechanical seals on pump shafts with a long overhang from bearing to impeller. If pumps are variable-speed or operate frequently at low flow (or near the shut-off head), high radial loads that bend the shafts and cause excessive wear on mechanical seals are produced. Most seals cannot withstand such shaft distortions. Pumps must be engineered for such applications. Specifying a maximum shaft deflection of 0.20 mm (0.008 in.) at the impeller and a B-10 bearing life of 30,000 h at the worst combination of operating conditions and 100,000 h at the best operating conditions promotes longer life, low maintenance, and life-cycle economy.

Bearings and seals or packing difficult to replace. Packing and bearings are the most frequently replaced items on pumps. Make seal replacement easy by ensuring easy access to the seal (such as avoiding close-coupled pumps and motors) and, for bearing replacement, provide a convenient means for lifting and storing motors and pump parts.

Bearing life not specified. Specify bearing life and insist that the manufacturer furnish certified bearing life calculations for the design point, pump speed, and impeller selected for both the worst and the most frequent (or best) combination of specified continuous operation conditions. The worst combination of operating conditions is particularly critical for V/S pumps. Be especially careful with high radial load conditions when operating near shut-off, especially if the head exceeds about 15 m (50 ft).

Positive-displacement pumps unprotected. A clogged line can break a positive-displacement pump unless it is protected by (1) a bypass line with a pressure-relief valve such as a simple air-loaded pinch valve, or (2) a pressure-actuated switch to stop the motor.

Poor or nonexistent pressure gauge installations. Every station should either have gauges or outlets for gauges on the pressure side (and also on the suction side, if practical) of a pump or pipe (see Figure 20-6). By using quick-connects with double shutoffs, one gauge can serve several locations. Ease of frequent calibration and inconsequential wear are two advantages of portable gauges.

Wastewater pumps inadequate for passing solids. Specify that the pump must be certified by the manufacturer to pass a particular size of sphere. Furthermore, test the pump at the site before installation. The Ten-State Standards [1] requires a sphere diameter of at least 75 mm (3 in.). Many 100-mm (4-in.) or even 150-mm (6-in.) pumps cannot pass a 75-mm (3-in.) sphere.

Cleanouts not specified. Some pumps are built without suction elbows or suction piping containing hand holes for removing rags or other debris from non-clog pumps, and such pumps cannot be cleaned without dismantling. Always specify hand holes for unclogging wastewater pumps and consider how to drain the pump and its piping conveniently (such as a drainpipe to the sump).

Use of main pumps for scum control. The buildup of scum in wet wells can be controlled with main pumps by manually bypassing the low-level pump control and by operating one or more pumps at full speed until the pumps break suction. The short-term vortexing sweeps the scum into the pumps. Equip the manual bypass feature with a spring-loaded return-to-automatic switch to restore the water level to the standard LWL and to avoid the hazard of operator error (see also Section 27-16).

Variable-speed drives in low friction head systems. Some authorities caution against the use of V/S drives for flat or nearly flat system curves or where the pump head-capacity curve is close to the slope of the system curve. The rationale, apparently, is that the pumps will be too difficult to control and the system will hunt. The authors believe to the contrary and suspect that the advice comes from experiences where it was not possible to tune the variable-speed

equipment to adjust for the dynamics of the physical system. In general, variable-speed equipment can react to error signals far more rapidly that the system requires. Essential features of any variable-speed control system include the following:

- Independent adjustment of minimum speed and maximum speed settings. The minimum speed setting should be established at the speed that represents development of system static head at zero flow.
- Independent adjustment of acceleration/deceleration rates. The rate of change of acceleration/deceleration must be linear with respect to time.

With these features, the authors' experience is that any variable-speed system can be tamed to respond properly to system dynamics.

Variable speed is appropriate for (1) positive-displacement pumps; (2) centrifugal pumps in systems where an instantaneously variable demand will occur, such as a tankless water system; (3) situations where relatively precise water level regulation is required; and (4) situations where sudden changes of flow rate due to start-up of a C/S pump cannot be tolerated by downstream processes. But be careful, and heed the above paragraph because V/S design in low friction head systems requires a careful tuning of the pump control system where a small change in speed produces a large change in flow rate.

Use of pressure gauges at start-up. Impellers may not be compatible with the system hydraulics because of errors or even blunders in determining TDH. To avoid this source of vibration and noise, always use calibrated pressure gauges on both suction pipe and discharge pipe for testing the pump. Compare with manufacturer's curve and trim or change impellers, if necessary, to keep the pump operating point definitely within the AOR and, for usual operation, within the POR.

27-8. Valves

Check Valves Improperly Selected

Be careful to match the valve in type and size to the system and to the flow.

Swing checks. Oversized valves operate improperly, but reducers and smaller valves can be used for low flows to obtain sufficient velocity to open the flapper fully. An external arm or a position indicator is desirable to show the position of the flapper.

Slam. Check valves may slam and cause disastrous water hammer. Avoid slam by using valves that close either very quickly (so that no reverse flow can occur) or very slowly (so that the water column comes gradually to rest). Quick-closing valves are preferable. Slow-closing valves tend to promote water hammer unless the closing mechanism (such as an external dashpot) is adjusted carefully and matched to the system. Dashpots are vulnerable to tampering. A water hammer analysis and an expert to adjust the closing mechanism are required (review the subsection entitled "Slam" in Section 5-4).

Quick closure. The quickest closure is provided by center-guided lift (or "silent") check valves followed by double-disc (or "double-leaf" or "double-door") check valves, but these types are for use only in clear water service. Closing characteristics of swing check valves with counterweighted versus spring-loaded lever arms is controversial (see "Slam" in Section 5-4). Carefully select check valves for the specific system.

Slow closure. Reverse flow will occur and the pump will run backward unless restrained. Impellers must be keyed, not screwed to the shaft. The wet well must be able to accept the reverse flow. Rags in wastewater flowing backward may jam swing checks, so use ball, lubricated plug, or cone valves for slow closure in wastewater service. One requirement is a stored-energy closure system that operates the valve quickly through the first stage of closure, then slowly to shut-off so as to minimize water hammer. Swing and slanting disc check valves are satisfactory for water service if equipped with dashpots that can be regulated to meet system requirements.

Swing check valves without cover plates. In wastewater service, the lack of a cover plate means that the entire valve must be removed to extract debris, so require removable cover plates.

Other Aspects of Valves

Improper location. Location is often critical for any type of valve. Never allow swing checks or gate valves in a vertical pipe carrying raw wastewater or grit-laden raw water. When the flow stops, grit settles and clogs the valve (or the valve bonnet of a gate valve), and the valve will eventually fail. If valves *are* placed on a vertical pipe carrying dirty water, use only eccentric plug, ball, cone, or pinch valves. Never install butterfly valves near (within two pipe diameters of) elbows or pumps because the curved streamlines may cause so much torque on the vane that the valve actuator will not function. Never install valves in a wet well. Never install swing check valves close to a hydropneumatic surge tank, because the short water column will accelerate too quickly and cause slam.

Air- and vacuum-release valves in wastewater service. Try to avoid using air-release and vacuum-relief valves for wastewater service. Relocate the force main or the pumping station. Dig deep trenches or tunnel. If the use of such valves is unavoidable, consider that the common types are high-maintenance items, that they frequently clog, and that they are unreliable in wastewater service. Specify them in pairs to provide a backup, and specify wastewater types with stainless-steel trim and flushing connections. In the O&M manual, require not less than monthly cleaning. Removal of all internal parts and shop cleaning is more effective than field flushing.

Vent-O-Mat® air-vacuum valves in field trials have remained fully operational several times longer than the older, common types in wastewater service. Refer to Figure 7-2 for proper installation.

Throttling. Except for slow starts and stops to control water hammer, seek methods other than throttling to control flow (such as variable speed, spillage, a bypass, or a smaller impeller) because throttling wastes energy. Gate valves should not be throttled for extended periods because they are quickly ruined by vibration. Low fluid velocity through a throttling valve results in poor control, so if throttling is unavoidable, consider using a reducer, a smaller valve, and an expander.

Nearly all butterfly valves are unsuitable for severe throttling service (as in pump-control valves). AWWA C504 specifications alone are insufficient to ensure the adequacy of valve seats. Some (but not all) replaceable body seats are satisfactory. Some manufacturers of pump-control systems who either use top-of-the-line makes or buy customized valves of special design may be able to offer advice.

Air-release valves in transmission and force mains. Air-release valves can cause water hammer (see Section 7-4) unless appropriately sized.

Vacuum-relief valve danger. Be extremely wary of using vacuum-relief valves in a pipeline if the pump control valve is of the slow-closing type. If the water column reverses its direction and the fluid velocity increases while the check valve is still mostly open (and unable to prevent high fluid velocities), an explosive force can be produced that is sufficient to throw slabs of concrete into the air. Hydraulic analyses and consultation with the vacuum-relief valve manufacturers are necessary to determine whether such a problem can occur.

Air in piping. Preferably, arrange piping so that air is automatically expelled as the wet well or sump fills. Air entrapped between the pump inlet and a check valve can air bind the pump, so install manual or automatic air-release valves (or both in wastewater service) at all possible points of air entrapment, such as the volutes of horizontal pumps, pressure gauges, suction piping, and both upstream and downstream from the check valve. A manual air release should have, say, a 12- or 19-mm ($\frac{1}{2}$- or $\frac{3}{4}$-in.) gate valve (for water) or a ball valve (for water or wastewater) followed by an air break and drainpipe to the floor sump.

Natural rubber seats. Because natural rubber in wastewater valves is attacked by petroleum products, specify a resistant synthetic elastomer (e.g., Buna-N or polyurethane).

Thin elastomer seats. Elastomers may be cut by grit in slurry service, so thick layers should be specified. Urethane is preferable to either natural or synthetic rubber.

Zinc and aluminum bronze. Avoid zinc and untempered aluminum bronze in valves for waters that cause dezincification (see AWWA C508-93, 2.2.2.4 and C504, Foreword III.7 and main text 2.2.11.2).

Valve handles. Chain wheels in wastewater wet wells are an explosion hazard. Hand wheels should be placed within easy reach and away from walls or other knuckle-cracking obstructions. Use either hand wheels or chain wheels with worm gears and plastic chains instead of levers on plug valves 150 mm (6 in.) or larger.

Valve stems protruding into walkways. Protruding valve stems constitute a safety hazard in walkways, so relocate them.

Power-actuated valves without manual actuators. The valve cannot be moved if the power actuator does not work. Always specify an auxiliary manual closure.

Power actuators not designed for continuous throttling service. Valves designed for throttling should have actuators that are specified accordingly; otherwise, maintenance is excessive.

Hand wheel with nonstandard direction. Valves should either open when the hand wheel is turned counterclockwise or the hand wheel should be permanently marked with the direction for opening.

Valve permits cavitation. Cavitation causes excessive noise, wear, or premature failure of valves. Select and install valves so that cavitation does not occur. Valves must operate within the limits recommended by the manufacturer.

Valve fails in the wrong manner. Design power-actuated valves and appurtenant piping so that the valves act in a safe manner when power fails (as it will).

Valve cannot be exercised without raw wastewater discharge. Redesign the system so that valves can be exercised with no resulting problems.

Improper support. Improperly supported valves or pipe can be distorted, which causes binding. Use an adequate number of supports.

Water valves in wastewater service. Use only valves designed for wastewater in wastewater service. Rags can wrap around shafts or catch on protuberances.

Freezing. Valves must be protected from freezing. If the fluid is warm enough, insulating the valve housing may be sufficient unless there is a possibility of zero flow.

Plug valve installed backward. Eccentric plug valves seat in only one direction. Analyze the requirements carefully and indicate the seating direction. In wastewater service, install eccentric plug valves with the shaft horizontal so that the plug is in the upper portion of the valve in its open position.

Cleaning tools will not pass. Select valves that allow passage of a cleaning pig. Otherwise, the piping may have to be disassembled for cleaning—a serious blunder.

Isolating valves added to pressure relief or other control valves. It must be possible to repair a valve by isolating it. Always design the system so that safety is not compromised by the forgetfulness of a careless worker (in reopening a valve, for example).

Control piping of incorrect diameter. It is important to use the correct diameter with respect to the size and speed requirements of valve actuators.

27-9. Mechanical

Cranes and hoists. Cranes and hoists must have access to the equipment with vertical falls to meet OSHA requirements. Piping and other obstructions above pumps or heavy valves must be located so as not to interfere. Design the facilities to permit lifting all heavy equipment straight up and depositing it directly on a truck bed.

Lifting eyes. Lifting eyes are an abomination, but if (for some reason) they must be used, at least design them so that stresses (in steel) do not exceed 69,000 kPa (10,000 lb/in.^{2}) at the smallest cross-section. Never use cast iron or malleable iron for any fittings to hold lifting eyes (see Figure 25-2).

Obstruction in walkways. Avoid them. For example, seal water can be carried in stainless steel piping buried in floors. Overhead piping and other objects are not obstructions if the headroom is at least 2.1 m (7 ft).

Unharnessed sleeve couplings. Sleeve (Dresser®) couplings, flanged adapters, bell and spigot joints, and mechanical joints can blow apart at any time unless they are longitudinally restrained. The restraint can be provided by burial in soil, confinement between rigidly restrained piping or equipment, or harnessing with enough bolts to resist the longitudinal forces due to water pressure (plus water hammer) in the joint.

Long shafts. Either avoid the use of long shafts or else plan for them carefully. Provide sufficient headroom for disassembly. *Always* balance long shafts dynamically, and *always* investigate torsional vibration, which is not apparent to the senses and is therefore more dangerous. Provide access for lubrication and consider framing the concrete structure for both the support of and access to any intermediate steady bearings. Also confer with the shafting manufacturer when considering the use of tubing large enough to allow for the omission of intermediate bearings.

Pipe flanges of different pressure class than pump, valve, or fitting flanges. Coordinate the plans and specifications to ensure flanges will fit.

Piping system inflexibility. Flexibility must be sufficient for the replacement of valves and fittings. Consider flange adapters or grooved-end (Victaulic®) or sleeve (Dresser®) couplings at strategic locations.

Sludge lines clogging. Sludge or concentrated raw wastewater tends to clog when at rest or when pumped at fluid velocities that are too low. Design for fluid velocities above the critical velocity, which is about 1 m/s (3 ft/s) for 2% sludge, 1.5 m/s (5 ft/s) for 4% sludge, and 2.4 m/s (8 ft/s) for 10% sludge. Refer to Chapter 19 for the calculation of headloss. Design sludge systems for easy cleaning—for example, by installing crosses instead of elbows to make rodding easier.

Water meter inaccurate or readout and capacity improper. Water meters are inaccurate unless the entrance pipe is straight for 15 to 20 pipe diameters for "head" meters and 2 to 5 pipe diameters for magnetic meters. Downstream flow disturbances also affect meter accuracy. Do not "fudge" on the manufacturer's recommendations for the required straight inlet and outlet distances. Choose capacity and readout units for present (not future) flow variations. Calibrate in situ.

Inadequate well head design. Many problems can occur, such as improper well casing ventilation, depth gauge, tap for raw water samples, clearance of top casing above the floor, and pressure gauge for pump testing. Review sanitary codes and engage a well pump specialist.

Flywheels used for water hammer control. Flywheels impose very severe service for bearings. Use oversize bearings (100,000-hour, L-10 life) and flexible couplings to the flywheel shaft.

27-10. Electrical

Control system too complex for contractor or operators. Assess and do not exceed the level of sophistication of local contractors and the client's personnel. Keep it simple.

Corrosion in control panels. Locate control panels in a clean, dry environment. If they must be in a damp environment, use NEMA 4 or 12 panels and purge them with dry air. NEMA 4 panels require thermostatic heaters in most of the United States because of nighttime moisture condensation in the cabinet. Use moisture-absorbing cartridges for corrosion control in all panels.

Excessive starts per hour. Starting motors and motor starters too frequently reduces their life. Contact the manufacturers—not the sales representatives—of both the motor and the starter regarding the number of allowable starts per hour and obtain guarantees in writing. Consider (1) "soft-starting" with reduced-voltage, solid-state starters for many starts per hour, or (2) V/S drives.

Big motors. Big motors may not start on the first try. Then the operators must wait for hours for a second try. To avoid the delay, specify enough thermal capacity for at least two starts. An operator, who should be standing by, should stop the motor if it fails to reach full speed in 10 seconds. Correct the cause of the malfunction before attempting a restart.

Junction boxes in wet wells. Avoid placing junction boxes in wet wells or at any location where water could enter the box. Boxes in moisture-laden atmospheres accumulate water that can travel along conductors and cause serious faults. In a wet well, a worker must enter a hazardous, confined space to disconnect, for example, a submersible pump. One means to keep junction boxes out of wet wells is to run the power cables through conduits to the control cabinet, as shown in Figure 17-21. If there is no alternative, specify watertight motor and device terminations. Keep moisture from entering by filling the junction box with potting compound and, for submersible motors, also specify non-wicking cable.

Inadequate lighting of work areas. Review lighting needs with the operating personnel and the electrical engineer. Operating panels should be lighted with a minimum of 500 luxes (50 ft · cd), but 200 luxes (20 ft · cd) is adequate for the pump room. Portable work lights are cheaper than fixed lights and are actually better. Check the hazard rating. Install guards on lamps and ensure easy access for replacement. Add a few incandescent lamps to provide instantaneous light to augment lighting that requires a long warm-up time before reaching full brilliance.

Inadequate or dangerous outlet receptacles. Always specify GFCI receptacles and always test them. Plan enough receptacles for power tools, electric hoists, and lights.

Phase not protected. Use breakers, not fuses, on motor-control and other three-phase circuits. Specify a three-phase monitoring instrument on the incoming power line to protect against undervoltage or a loss of phase.

Improperly sized circuit breakers. Specify properly sized circuit breakers to protect against (1) short circuits, (2) overloading over a short time interval, and (3) nuisance tripping. All electrical equipment must have adequate withstand ratings.

Improper grounding. All electrical (and mechanical) equipment must be properly grounded (see "Grounding" in Section 8-5).

Power failures. Power failures will occur, so design the system to prevent column separation and other disasters. Add back-up in the form of stored energy to operate slow-closing valves.

Power supply inadequate. Check with the power company before beginning the design.

Positive-displacement pumps with high static heads or long force mains. These sometimes require high-slip (8 to 13%) "D" motors to accelerate properly the tons of liquid in a force main. Check the pump and operating characteristics with the motor manufacturer.

Transfer switches without time delay in neutral. The field voltages on motors driving loads with a large WR^2 do not collapse rapidly. If the motor is transferred quickly from one power source to another (say, to an emergency generator or vice versa) and if the two sources are out of sync, mechanical forces that are large enough to twist shafts, break couplings, and tear windings out of motors are produced. Specify a transfer switch with an adjustable time delay in neutral. Usually 20 to 30 cycles is a sufficient delay.

Wet well electrical equipment not explosion-proof. Always classify wastewater wet wells as Class I, Group D, Division 1 or 2 in accordance with the NEC. Electrical equipment should be explosion-proof. Use nonsparking tools and safe maintenance practices when working in or around wet wells.

27-11. Structural-Architectural

Anchor bolts incorrectly located or with inadequate projection. Anchor bolts out of place is an all-too-frequent problem that can be solved by (1) using sleeves to permit minor adjustments of bolts, and (2) fastening bolts to a plywood template before casting the concrete to ensure exact placement. Admon-

ish the field inspector to check and recheck to ensure correct placement. An alternative is to place anchors in accurately located drilled holes with epoxy (or a similar quick-setting, two-component glue) in breakable capsules, but some experienced engineers will not use them, because the strength of the anchor depends so much on the quality of the field work. In corrosive environments, use extra-heavy bolts, or use stainless steel or nonferrous material.

Inadequate detailing. Show details of such elements as water stops, roof flashing and drainage, louvers, weather stripping, clearance around equipment, and provisions to guard against freezing. Detail roof and wall penetrations for adequate clearance between engine exhaust pipes and combustible materials. Ensure that the work of structural, mechanical, electrical, instrumentation, and architectural disciplines is closely coordinated.

Differential settlement between structure and exterior piping. Be sure that the exterior pipeline has enough flexibility in its joints adjacent to the wall or support both the structure and the pipe on piles. Always use wall sleeves. Because of the high probability of differential settlement or frost heave, interrupt plastic electrical conduit with metal conduit through exterior walls.

Foundation problems. Obtain the services of a qualified geotechnical engineer and adequately investigate the site. Include field borings or test pits (see "Subsurface Investigations" in Section 25-2 and "Geotechnical Considerations" in Section 25-10).

Inadequate hatches. Make hatches large enough (at least 700 mm × 700 mm or 28 in. × 28 in.) for a worker with a belt of tools to enter. In wastewater wet wells, allow at least 700 mm × 900 mm (28 in. × 36 in.) for a self-contained breathing apparatus in a backpack. Consider aluminum checkerplate with stainless-steel hardware, positive-latch openers, and inside grab rails for easy access to the ladder or stairs. Check OSHA standards for hatches and their locations.

Level floors and, consequently, puddles. Slope floors about 1% to a floor drain (2% is better). Concrete floors can be cast level if a wearing slab of variable thickness is applied later, but there is considerable difficulty in obtaining a good bond.

Inadequate monorails or cranes. Preferably, arrange the station to provide monorails (for small stations) or cranes (for medium-sized to large stations) above all heavy equipment, and make it easy to set heavy items on a truck in a doorway. Consider motorizing all motions for cranes and hoists with capacities greater than 680 kg (3/4 ton). Use slow speeds because pumping station operators are rarely skilled in the operation of hoisting equipment. Make access hatches light (for easy lifting) and large enough to remove equipment without dismantling it. At the very least, small stations should have lifting eyes for portable hoists (see Figure 25-2).

Inadequate clearances. Allow enough room for the removal (and replacement) of every piece of equipment without disassembly or with minimum disassembly. Avoid cramped working areas by allowing at least 1.1 m (42 in.) around all equipment (including appurtenant piping) for heads, hands, feet, rumps, wrenches, and lights for maintenance repairs and operations. Carefully draw plans for complicated equipment such as engines or generators together with all attached auxiliary or appurtenant equipment to ensure that there is about 3 m (10 ft) of clearance for ease in dismantling for repair.

Inadequate access. Install platforms and stairs to reach areas that require frequent maintenance. Platforms should be strong enough to support dismantled machinery parts.

Inadequate strength, stiffness, and watertightness. Avoid using the minimum design requirements in the ACI 318 Code. Heavier construction is required, so use ACI 350 code.

Inadequate ventilation for motor cooling. Provide for close coordination between mechanical and electrical disciplines. Consider air conditioning, especially if noise must be suppressed.

Vibration. Avoid resonance frequencies by supporting pipes at frequent intervals and using deep beams and thick floor slabs. Isolate diesel engines by setting them on spring supports (see Chapter 22).

Wet pit atmosphere seeping into pump room. Completely sequester wet pits from the pump room so that neither gas nor wastewater (even if the wet well overflows) can enter the pump room. Seal wires within interconnecting conduits placed at the highest practical elevation. Sump pumps should always discharge above the highest possible water level to a sewer *outside* the pumping station.

27-12. Specifications

Pumps. Specify: (1) the pump operating point at maximum station flow rate; (2) the least head for a single pump operating with the smoothest expected force main (runout); and (3) peak efficiency. It is virtually impossible for a pump manufacturer to meet several required efficiency points or to meet a specified efficiency at a specified operating point. Realize that the designer does not know where the pump will operate most of the time anyway, so it is unreasonable to specify where the peak efficiency should be located.

Equipment specifications not to code. Equipment specified by a single named manufacturer or by sole-source procurement must meet federal and state requirements.

Pump testing. Require witnessed factory tests of all large and important pumps, but use judgment before requiring witnessed tests for small pumps. Require the field operational tests described in Section 16-6.

Incomplete or conflicting specifications. The project engineer must ensure that specifications for all materials, workmanship, and submittals are complete and must coordinate them with the plans (see Chapter 17 and Appendix E).

Specifications not incorporating unit responsibility. Specifications should require that a factory-certified representative of the manufacturer be responsible for the complete coordination of multiple-component equipment packages (such as pumps, V/S driving equipment, drivers, and associated controls) when furnished by a single manufacturer. Otherwise, you must be prepared to coordinate all the parts and assume responsibility for such diverse aspects as motor size, motor heating characteristics, control system characteristics, pump torque requirements versus motor torque output at variable speed, coupling sizes, coupling bore sizes, shaft length, torsional analysis, shaft guard design, and a host of other concerns over which—not being the manufacturer—you have little or no control.

27-13. Economics

Failure to select an appropriate pump. Thoroughly investigate a pump's efficiency, ease of maintenance, energy costs, and compatibility with other pumps in the system.

Reliance on first cost for bid selection. A penalty/reward specification can be added in a bid clause to include the present worth of anticipated energy use (see Chapter 29). Pump specifications can be written for low maintenance (see Chapters 11, 12, and 16).

Obsession with economy. How well the station works and how easily and conveniently it can be maintained is vital. Two or three points of efficiency or a 4 to 5% savings in construction cost is trivial in comparison with a design that ensures a maximum of convenience and a minimum of maintenance and repairs.

27-14. The Future and Remodeling

Future expansion not considered. Design the station to allow for an increased discharge by (1) specifying pumps that can accommodate larger impellers, (2) adding another intake with a blind flange, or (3) choosing oversized piping (especially on the suction side) so that pumps and motors can be replaced with larger ones. Fasten a lifting eye on large blind flanges because they are otherwise difficult to handle.

Develop a strategy for adding or changing pumps and piping so that the station need not be shut down for the change. A valve with a blind flange is one way, but valves not exercised will freeze and, thus, be worse than useless. The strategy must be safe and sure and should be written into both the specifications and the O&M manual.

Reliance on record drawings and specifications. Never trust record drawings. Always make an accurate field survey (or engage an expert to do it) to determine the *actual* static head, TDH, flowrate, pipe roughness coefficient, possibility of obstructions in the force main, wire horsepower, exact locations of pipe and flanges, and any other measurements that may affect the new plans, before beginning the design.

27-15. Find the Blunders

Schematic diagrams of two pumping stations, each containing at least 10 blunders, are shown in Figures 27-4 and 27-5. Many of these blunders are discussed in this chapter, but others are found elsewhere. Sharpen your wits by finding them before turning to the answers in Appendix D.

27-16. Design Reviews

A design review strategy should be imposed prior to beginning the design. Ideally, review teams should be made up of the personnel listed in Table 27-1.

For a typical large project, reviews might well take place on the following schedule:

- Predesign review
- Intermediate reviews (at 10, 50, and 90% completion, plus a field check)
- Regulatory agency review
- Post-construction review.

For smaller projects, the reviews might be made less often (or at different times), but project leaders should realize that reviews by those not directly associated with the detailed design are vital if blunders, errors, and problems of inadequate design are to be minimized. The cost of such reviews is small compared with the cost of field correction of mistakes, the value of the firm's reputation, and malpractice insurance premiums.

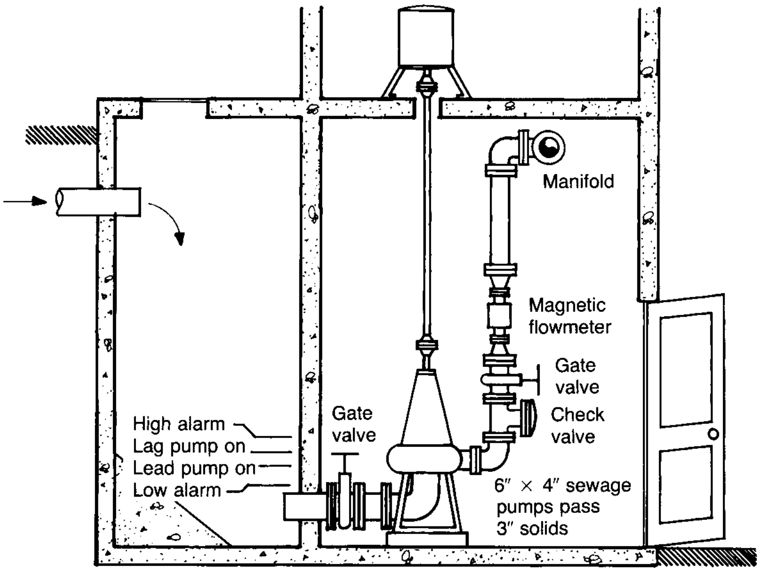


Figure 27-4. A composite of several wastewater pumping stations. Find at least 10 blunders before looking at the answers in Appendix D.

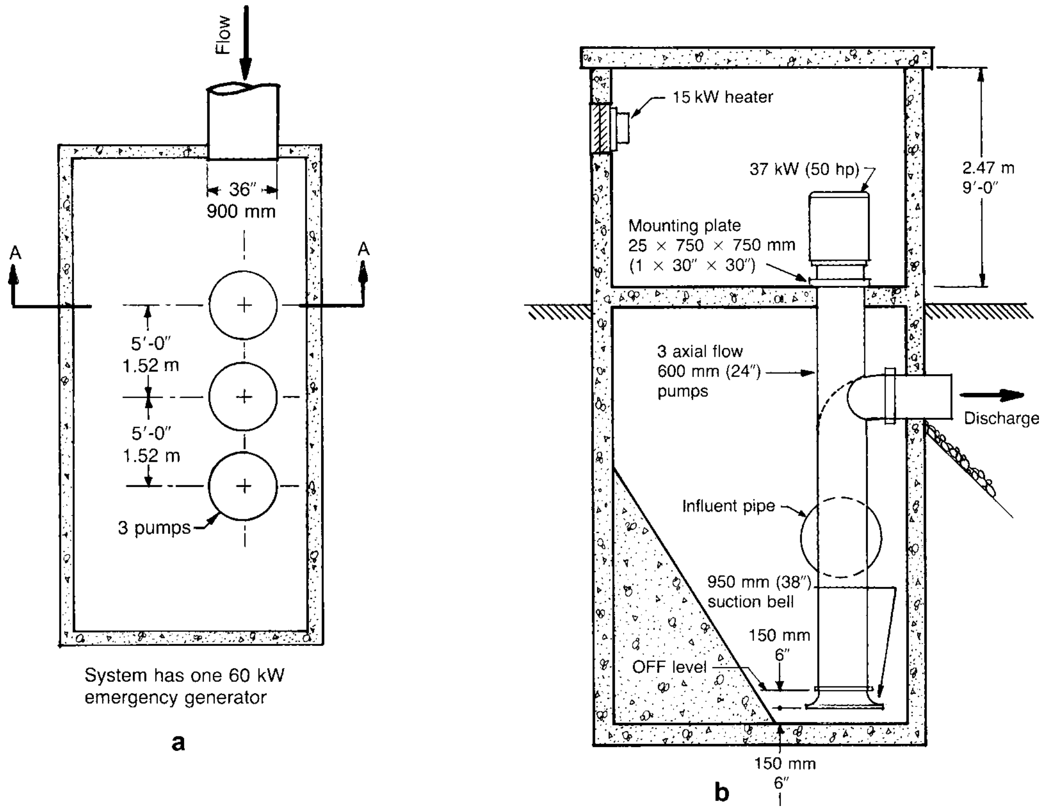


Figure 27-5. An actual low head wastewater pumping station. The station has one 60-kW standby generator. (a) Plan; (b) Section A-A. Find 10 blunders before looking at the answers in Appendix D.

Table 27-1. Personnel on the Review Team

Small pumping stations	Large pumping stations
Project leader	Project leader
Engineer from each discipline[a]	Lead engineer from each discipline[a]
Owner's representative	Project and/or technical consultants with extensive experience in each major discipline involved
At 50% completion of detailed design, an engineer with broad and extensive experience (inhouse if available, or an independent consultant if not)	Owner's representative
	Owner's operating personnel
	Regulatory agency personnel

[a]See the listing of disciplines in the foreword of Chapter 25.

Appendix E contains checklists for the following disciplines:

- Civil design
- Structural/architectural design
- Electrical design
- Instrumentation and control
- Cross-connection control, especially for wastewater plants
- Mechanical design.

27-17. Operations

O&M manual not supplied. Always supply an O&M manual. If the owner objects to paying for it, do your best to overcome the objection by explaining it is a savings—not an expense. No pumping station should be without an O&M manual in which inherent pitfalls and dangers are identified clearly, and how to get the best from the equipment is stated in simple language. State the design assumptions and considerations, especially with respect to meeting future flows (by changing impellers or adding pumps). Put schematics on one page for the operator's use. Avoid engineering jargon in the O&M manual and use simple, clear language with (if necessary) sketches for clarity. Furnish several copies of the O&M manual and at least one copy of record plans and specifications. Furnish a weekly/monthly inspection form developed in cooperation with the maintenance supervisor. During the design, write portions of the O&M manual to clarify your thinking and to produce a better, more easily maintained pumping station.

Cleaning wastewater wet wells. Clean wet wells on a periodic basis—at least weekly and more often, if necessary. Waiting overwhelms pumps, generates odors, and increases cleaning time. It should be possible to clean a well-designed pumping station in a few (five or less) minutes if cleaning is frequent. Prevent excessive grease buildup on the walls by hosing at, for example, 2.5–4.4 L/s at 620 kPa ($40-70\,\text{gal/min}$ at $90\,\text{lb/in.}^2$).

Cleaning storm water wet wells. Try to clean them at the tail end of storms when the inflow of solids is at a minimum and there is still sufficient flow for the task.

Operating pumps in flooded station. Operating pumps when submerged is hard on seals and bearings and may ruin the motor. The O&M manual should warn operators to dewater first.

Entry into confined spaces. In the O&M manual, state the safety precautions for entry into wastewater wet wells and other hazardous confined spaces. The precautions include pretesting for explosive gases before removing manhole covers, using portable forced-air ventilation, testing for toxic gases and oxygen depletion before entry, using a safety harness hooked to a winch, and using trained support personnel.

Pumping scum with main pumps. Give proper directions in the O&M manual for the removal of scum with main pumps; operate all pumps to draw water surface down to suction inlets quickly where the low submergence causes vortexing and draws scum into pumps. As soon as air is drawn into the pumps (which is signaled by an increase in vibration), stop the pumps and make sure that the switch is returned to automatic and that the pumps are reprimed. Operation of submersible motors with water level below the motor for more than a few moments may overheat and damage the motor. Verify the allowable time for out-of-water operation with the manufacturer. If it is necessary to remove scum more frequently than once a week, investigate the source of the scum and grease and have it controlled or use other methods to pump scum from the wet well.

Warnings buried in the O&M manual. Prominently display a summary of hazards and warnings at the beginning of the manual. Give safe operating procedures for emergencies such as chlorine leaks, fire, flooding, power outages, and injury to workers.

Disassembly not given in the O&M manual. Furnish a strategy for the disassembly of piping and the removal and replacement of machinery.

27-18. References

1. Ten-State Standards, *Recommended Standards for Sewage Works,* Great Lakes-Upper Mississippi River Board of Sanitary Engineers, Health Education Service, Inc., Albany, NY (updated periodically).

2. Prosser, M. J., "The hydraulic design of pump sumps and intakes," British Hydromechanics Research Association (BHRA)/Construction Industry Research and Information Association, Cranfield, Bedford, UK (July 1977).
3. Dicmas, J. L., *Vertical Turbine, Mixed Flow, & Propeller Pumps,* McGraw-Hill, New York (1987).
4. Falvey, H. T., *Air-Water Flows in Hydraulic Structures*, U.S. Dept. of the Interior, Bureau of Reclamation, Engineering Monograph 41, Superintendent of Documents, Washington, DC (1980).
5. Triplett, G. R., B. P. Fletcher, and J. L. Grace, "Pumping station inflow-discharge hydraulics, generalized pump sump research study," Technical Report HL88-2, Department of the Army, Waterways Experiment Station, Corps of Engineers, Vicksburg, MS (February 1988).
6. Karrasik, I. J., J. P. Messina, P. Cooper, and C. C. Heald, *Pump Handbook*, 3rd ed., McGraw-Hill, New York (2001).

Chapter 28

Contract Documents

JOHN E. CONNELL
THOMAS M. FLEGAL

CONTRIBUTORS
David L. Eisenhauer
Earle C. Smith

This chapter contains a brief description of what is included in construction project documents, how they should be prepared, and a recommended format to be used. The information given here in conjunction with the references should be sufficient for guidance and assistance in the development of clear, concise contract documents.

28-1. General

Construction documents must communicate clearly a complex array of technical and legal information defining both the work to be accomplished and the responsibilities of the involved parties. For a traditional design-bid-build project delivery method, that information is presented in the following documents:

- Specifications
  - Bidding Requirements
  - Contracting Requirements
  - Technical Specifications
- Drawings
- Addenda and Modifications

Drawings depict the work graphically and the specifications provide a written description. Addenda and modifications are official changes to both.

The specifications address (1) the bidding process by which the cost of the work is decided; (2) the legal relationship and contractual obligations of the owner, who is paying to construct a facility, and the contractor, who has agreed to construct it; (3) the status and responsibilities of the engineer who designed the project and the construction observer who will act on behalf of the owner during construction; (4) the products to be incorporated in the work and the activities of the various parties engaged in the construction process.

The specifications may be outlined as follows:

I. *Bidding Requirements*. Address prospective contractors, subcontractors, and suppliers.

   a. Invitation to Bid (Advertisement for Bids: Bid Solicitation)
   b. Instructions to Bidders
   c. Bid Form
   d. Bid Bond

II. *Contracting Requirements*. Address the contractor and the owner.

   a. Agreement
   b. Payment Bond
   c. Performance Bond
   d. Notice of Award
   e. Notice to Proceed
   f. Change Order Form
   g. General Conditions
   h. Supplementary Conditions
   i. Addenda

III. *Technical specifications*. Address the contractor directly and address the suppliers and manufacturers through the contractor.

Specification Language

The specifications are a complex and extensive set of instructions that direct the contractor. The writing style must reflect this objective by using clear, positive statements. The indicative mood is most effective in giving this direction to the contractor. Language that only suggests or implies an action is likely to cause misunderstandings or misinterpretations of the true intent of the statement. Engineers should be precise in specifying requirements for materials of construction and workmanship. Phrases such as the following should not be used:

- *Suitable*
- *Stainless steel* (without stating the AISI grade)
- *Steel angle* (without stating the ASTM designation)
- *As required*
- *As necessary*
- *Appropriate*
- *In a workmanlike manner*
- *Of the best quality*
- *Heavy duty*
- *Industrial grade*
- *Commercial grade*
- *Including but not necessarily limited to*.

For accurately describing what is to be provided in the contract documents, the importance of careful reading before referencing current standards cannot be overemphasized.

Format

Choose a format for the specifications that is clear, logical, and consistent from section to section. The format for some projects is established by the owner. Federal and state agencies often require their format be used in writing specifications. Many consulting firms have their own specification format, which uses a standard approach and is set up on a word processing system. There are also professional societies, such as the Construction Specification Institute (CSI) [1], that support a specific style and format (exemplified in Appendix C) or system for writing specifications. These societies have large libraries of standard specifications for bidding and contract requirements and for technical specifications.

28-2. Bidding and Contracting Requirements

The documents listed in Section 28-1 under "Bidding Requirements" and "Contracting Requirements" often are referred to in combination as the "front-end" documents. Although the contract documents include the entire set of drawings and specifications, these documents are sometimes referred to as the contract documents because of their legal as opposed to technical content. They are used to establish the contract price and set out the basis for the owner–contractor relationship in a binding contract agreement. They must be coordinated among themselves and with the rest of the contract documents so that they do not conflict, and they must be properly referenced.

Special articles and paragraphs often must be added to the bidding and contracting documents for projects funded by various governmental agencies such as the U.S. Environmental Protection Agency (EPA) to satisfy agency requirements. Become familiar with agency requirements and include the additional documentation when required.

Standard bidding documents, agreements, general conditions, and other documents are published by various professional organizations and interest groups. Perhaps the most useful to engineers are the documents prepared by the Engineers' Joint Contract Documents Committee (EJCDC) [2] and issued and published jointly by the National Society of Professional Engineers (NSPE), the American Consulting Engineers Council (ACEC), and the American Society of Civil Engineers (ASCE). The use of such documents is beneficial because they have been approved and endorsed by the Associated General Contractors of America and the CSI and also because contractors, engineers, and owners have become familiar with their contents. Owners such as state agencies, federal agencies, cities, or industries may require use of their own set of standard documents. Standard forms and documents recommended by the American Institute of Architects (AIA) [3] are used for most private commercial and residential building construction. Engineering firms often develop their own standard documents. There should always be a review by an attorney to ensure that the content is appropriate for the contract.

Invitation to Bid

The Invitation to Bid (also known as the "Advertisement for Bids" or "Bid Solicitation") should be concise and must contain:

- The owner's name and address
- A brief statement of work
- The time, place, and method of placing bids
- The locations where bid documents may be obtained and information on plan deposit can be found
- The amount and type of bid surety
- Special qualifications that may be required of bidders.

This document is usually published in construction or trade journals in the region where the project is to be constructed. Government projects usually require a legal advertisement in a local paper, and the advertisement for such bids must be written to comply with these requirements.

Instructions to Bidders

Instructions to Bidders is a summary of the bidding procedure and the requirements of the bid. It is more detailed than the advertisement and contains information on preparing and submitting the bid, on bid bonds, on performance and payment bonds, and on the methods of evaluating responsive bids, as well as a schedule and procedure for awarding the contract and proceeding with the work. Available information on existing conditions (such as buildings and soil information) and special requirements (such as license requirements or other regulations affecting the contractor) are often mentioned in the Instructions to Bidders. Care must be taken in preparing the bidders instructions package and all other documents so that they do not conflict with other parts of the specifications. To reduce the chance of a conflict, do not repeat items that are addressed in detail elsewhere in the documents. Examples of Instructions to Bidders are given in EJCDC and AIA documents [2, 3].

Bid Form

The bid form contains the agreed compensation for the work described. All prices, whether for a lump sum or a unit-price contract, are given in words and figures. If there is a discrepancy, the amount in words governs. Receipt of all addenda must be acknowledged on the bid form.

A bid bond is normally attached to the bid form. Other attachments often include bidders' prequalification data and a list of subcontractors. Examples of such attachments are included in the EJCDC and AIA documents [2, 3].

Bid Bond

The bid bond may be issued by a bonding company; the alternative of presenting a certified check in the amount of the bid bond is sometimes allowed. The purpose of this bond is to protect the owner if an apparent low bidder defaults and refuses to enter into a contract agreement. The bid bond is usually 10% of the bid price. The bonds for the three lowest bidders are held until a contract is awarded.

Agreement

This document is the legal instrument used as the formal contract. The contracting parties, the time for completion, and the amount of compensation are identified or established in the agreement, which also incorporates all of the contract documents by reference and attachment. If the owner elects to specify damages for failure to complete the work on time, those provisions are included here. The completion time must be reasonable for the work, and the damages must also have a reasonable basis. Unreasonable damages will likely not be awarded if challenged in court by the contractor. Reasonable damages may include additional engineering fees, administrative costs, loss of revenue to be generated by the project, and fines or penalties assessed by another authority (such as fines for a violation of the National Pollution Discharge Elimination Permit). (See the EJCDC and AIA documents [2, 3].)

Performance and Payment Bond

These bonds are provided by the contractor from a surety company in the amount of 100% of the contract price for each bond. These bonds protect the owner if the contractor fails to complete the work properly or if the contractor fails to pay those who worked on the project. Examples are listed in the EJCDC documents [2].

Notice of Award

The selected bidder (usually the low bidder) is notified in writing that his bid was accepted, and the contractor is given a specified time to present the necessary bonds, insurance, and executed contract agreement to the owner. An example of the notice of award is given in the EJCDC documents [2].

Notice to Proceed

Upon review and acceptance of the contractor's bonds, insurance, and agreement, the owner executes the agreement and provides written notice for the contractor to begin work within a specified time and to complete the work within the agreed completion time (see the example in the EJCDC documents [2]).

Change Order Form

Changes in the work, completion time, and contract price all require a "change order" to the contract. The specific form required to be used for this purpose is sometimes included as part of the formal contract documents. It is typical to add attachments describing any changes in detail. An example of a change of order form is contained in the EJCDC documents [2].

General Conditions

The general conditions are the focal point of the contractual documents. Documents and terms are defined. The authority of the owner and the relationship of the engineer and construction observer are outlined, as well as the duties and obligations of the contractor. ("Construction observer" or "resident project representative" are the preferred terms because the words "inspector" or "supervisor" may communicate an unintended meaning—and liability—to the courts.) Method of payment, insurance requirements, project completion, guarantee of the work, and methods of resolving differences are examples of the content of the general conditions. The standard general conditions as published by the professional societies are commonly used in specifications (see examples in the EJCDC and AIA documents [2, 3]). All other documents must be consistent with the content of these general conditions.

Supplementary Conditions

Supplementary conditions are used to add, expand, or alter the general conditions. Supplementary conditions are normally written for a project and incorporate requirements that are specific to the project. When the owner requires his or her own supplementary conditions be used in the documents, it may be necessary to have an attorney review both the general conditions and the supplementary conditions to avoid conflicts. When altering a set of standard general conditions, make it clear that it is a change and reference the section to be altered. Avoid intermixing technical specifications with the supplementary conditions. Also avoid the tendency to repeat supplementary conditions from previous project specifications—a common source of conflicting requirements or unnecessary specifications. A guide to the preparation of supplementary conditions is contained in the EJCDC documents [2].

Addenda

Addenda are either written or graphic instruments issued prior to the bid opening to clarify, revise, add to, or delete from the original bidding documents or previous addenda. Like change orders, addenda are often prepared in a special format that later becomes part of the formal contract documents. Addenda must be acknowledged on the bid form.

28-3. Technical Specifications

Technical specifications provide a detailed description of the scope of work, project-specific administrative and procedural requirements, type and quality of materials, performance of equipment and systems, and the level of workmanship expected of the contractor. The specification sections typically are grouped into separate, numbered divisions by type of product and construction specialty, following the format recommended by the CSI. The drawings, which are also part of the contract documents, must be coordinated with the technical specifications. Drawings illustrate construction and provide a graphic means of showing the work to be done. Specifications should supplement, but not repeat, the information shown on the drawings.

The greatest difficulty in writing technical specifications is to decide when a given item is adequately described. Obviously, each and every move required by workers in carrying out the work cannot be described. The underlying presumption is that contractors execute work at a level that is consistent with their particular trade. The specifications are not to instruct carpenters, pipefitters, plumbers, or electricians in their trades but rather to describe the quality and precision of the final product as well as the relationship and interaction of components to a functional finished project. Industry standards of practice, however, are frequently referenced as part of the technical specifications. Building codes, electrical codes, plumbing codes, and so on are typical of such standards of performance.

Pitfalls: Who Should Write and Coordinate the Specifications?

Obviously, one person cannot write all the technical specification required in a pumping station project. Such projects are complex—sometimes *very* complex—and several disciplines are involved, among them civil, structural, architectural, electrical, instrumentation, and so forth. Various specialties, such as acoustics, vibration, model study, and so on, are sometimes needed. All these disciplines or specialties (and more) may be involved in preparing technical specifications for a project and, hence, various parts of the specification must be written by those experienced in such specialties.

A key issue is: Who should coordinate and review the technical specialists' work? Uncoordinated specifications can result in problems ranging from duplicate—and different—specifications for a topic to a subject not covered at all (because the specialists assumed someone else was writing the specification).

Usually, the project manager should be responsible for the review and coordination of the technical specifications. However, one person physically cannot manage both the nontechnical and technical aspects of very large, complex projects. Considerable time is required to examine and coordinate the design drawings and specifications page-by-page. The task requires someone with experience in all aspects of design or at least with enough experience to know where the interdisciplinary problems usually occur and to know how to resolve the problems. Consequently, a technical assistant project manager or chief project engineer may be required to review and coordinate plans and specifications continuously during the project and to perform final review at the end of the design period. Some large organizations have a department of specification writers who become very skilled at writing and preparing specifications.

Problems such as the following are frequently found and must be resolved:

- Geotechnical specialists providing standard or "canned" specifications that do not match the recommendations of the geotechnical report.
- Mechanical and civil (sometimes called "process") specifications for valves, pipe hanger spacings, pipe pressure ratings, pressure test requirements, etc. that are inconsistent.
- Electrical wiring that does not match the requirements of the instruments.
- Specifications for packaged equipment (e.g., pumps, compressors) that are not coordinated with instrumentation and control specifications and P&IDs.
- Materials for corrosion control that are inconsistent (e.g., stainless steel specified by one discipline while galvanized steel is specified in the same area by another discipline).
- Specifications on the drawings causing potential conflict with written specifications.

The inexperienced pumping station designer may think such coordination is simple and easily resolved by a day-long conference of each of the discipline leaders—a naive supposition.

Addressees

Technical specifications must address a wide range of parties. Although the contractor is the primary audience, the technical specifications must also address bidders, subcontractors, suppliers, manufacturers, workers, and the construction observer.

Each technical section is used for:

- Bidding
- Product and material submittals, standard of quality, testing and acceptance
- Product description, materials of construction, and performance
- Product installation, performance testing, start-up, and functional demonstration.

However, the design engineer writes the specifications to the contractor, who will sign the agreement with the owner. Subcontractors are not legal parties to the agreement. Consequently, the design engineer should not write specifications stating things such as "the concrete contractor shall . . . " or "the mechanical contractor shall . . . ." A phrase to be used carefully is "by others." The intent of the engineer, for example, may be to tell the concrete subcontractor that he does not provide the monorail support beams. But, because the contract documents are addressed to the general contractor—the entity with whom the owner has the legal relationship—the actual effect is to tell the general contractor that he does not have to provide monorail support beams at all. The phrase "by others" should only be used to denote equipment or work that is being done under another set of contract documents.

Most projects involve a wide variety of trades, products, and materials that are needed to complete a pumping station. Such complexity requires a systematic approach to writing technical specifications. The systems used may be developed by an engineering

firm, by the owners (such as federal specifications), or by technical or professional societies.

The system or approach should not be confused with specification content. The system is the method that is used to break the complex project into meaningful sections or parts. These parts may then be described in detail, and their relationship to other parts may also be described. An excellent system of logical divisions, which is widely accepted in architectural and engineering projects, has been developed by the CSI [4]. The scheme also includes a more detailed breakdown of individual specification sections into three parts: (1) general, (2) products, and (3) execution. The content of each of these parts also is prescribed.

28-4. Source Material

Numerous sources of information are available for specifying an element of work or a product to be incorporated in the work. In general, these sources are as follows:

- Manufacturers or suppliers of products and materials
- Regulatory requirements (such as building and plumbing codes) or owner-required specifications (such as federal or military specifications)
- Professional or trade organizations that have set standards for material composition, performance, and product standards.

As a designer or specification writer, you will encounter numerous representatives of equipment and materials. Most representatives will provide typical specifications for their products. Although such information is very useful for keeping informed on the current competitive market, be cautious when using a representative's standard specifications. The product may meet the regulatory requirements and the trade standards, but it may also contain elements placing it in an unnecessarily favorable bidding position that excludes other acceptable products. Review several such specifications and then edit or rewrite the specifications to ensure two or more sources that would be acceptable for the application. Avoid specifying elements that are not of standard manufacture unless there is a special reason for such a choice. Remember, special items may be difficult to maintain or replace.

Regulatory requirements affecting the work should be referenced in appropriate specification sections. Although the various trades working on a project may be required by law to perform work under a given code, it is best to state that applicable code conditions must be followed in the general portion of the specification. Do not attempt to repeat or paraphrase such codes because that could lead to conflicts and misinterpretations.

There are numerous professional or trade organizations that have developed product and material standards and quality tests. The given trade or industry uses these standards as a means of self-regulation. Many have become national standards that are often recognized by regulatory agencies. The use of such standards is commonplace in technical specifications. Materials may often be specified adequately by simply referencing the appropriate trade standards. A more complex item, such as a valve, may be specified by reference to an appropriate AWWA specification. However, the reference alone may not be adequate because the referenced standard usually has selection options that must also be identified. Thus, such a reference must also identify the options allowed by the referenced standard.

The use of standards is extremely important. Unfortunately, the engineering industry in general and design engineers in particular are all too often increasingly unfamiliar with standards such as ASTM [5] and AWWA [6], and the inexperienced engineer has, at best, a superficial knowledge of standards. Unfortunately, it is also common for the design or specifying engineer or architect to be unfamiliar with these referenced standards and, frequently, not to have read them at all. Many of these referenced standards, if blindly incorporated into a project contract document by reference, will drastically alter the relationship between the design engineer, the owner, and the general contractor. Other referenced standards may require the design engineer to take actions that may be unwanted. Therefore, read every part of a referenced standard carefully and thoughtfully and refer exactly to those portions that apply to the project (see Section 1-4).

Limitations of Published Standards

Published standards provide the specifier with a convenient means for incorporating recognized benchmarks of quality into the detailed requirements for contractor or manufacturer performance. Most published standards are the product of countless hours of effort provided by volunteers interested in improving their industry. Today, after legal decisions have caused a re-examination of the process, most (but not all) standards-setting organizations use balanced committees (memberships representing manufacturers,

users, and consultants or specifiers) to develop consensus documents. Once a document has been developed, it is then published for public comment. Public comments are then considered by the committee and the document is adjusted if necessary before final publication. Standards organizations usually have an oversight process to ensure the documents contain no biased requirements.

These consensus documents *usually* provide the specifier with sound advice on the basic requirements for a product and options for enhanced quality or for alternative features appropriate for special applications. Many standards contain options for quality assurance reporting and for user-nominated requirements for construction options. No standard, however, is perfect. It is unlikely that any standard will be entirely applicable to a given application without some modification. Consequently, it is incumbent upon the specifier to read and understand every standard completely with a watchful eye for any deficiencies such as omissions, poor or weak practice, or inconsistency with other standards. The following shortcomings are a few examples taken from commonly used standards.

- *Omission*. Omission of requirements for surface preparation for coatings, film thickness testing, or frequency of testing.
- *Poor practice*. Allowing threaded joints in Schedule 30 pipe.
- *Inconsistency*. Referencing a specification at odds with the main body of the standard.
- *Weak practice*. Allowing excessive pipe hanger spacing that results in excessive pipe sag.

Many other examples can be cited, but the message is clear. *Referenced standards must be read completely and with care (1) to discover and correct omission, contradictions, and weaknesses, and (2) especially to ensure conformance with the objectives of each project*.

Every specification writer should have access to the *ASTM Standards* [5], the *AWWA Standards* [6], and the AIA *MASTERSPEC* [7] (as well as the other references listed in Section 28-7).

28-5. Methods of Specifying

There are five basic methods of specifying:

- Performance
- Nonrestrictive (also "or Equal")
- Proprietary
- Generic (or Descriptive)
- Reference standards.

Combining two or more of these methods may prove to be the best way to clearly and efficiently state requirements. An example of this approach is given in Appendix C (see also Table 16-1).

Performance Specification

The performance specifications describe the functional result that a product must achieve without identifying any specific material or product. A functional result specification usually places the burden of some design work on the contractor. Acceptance of the product may not be ensured until it has been demonstrated that it can perform the intended function. The cost for such a specification may be high because the contractor must include design costs as well as costs to cover unforeseen problems should the products fail to perform as specified. Performance specifications are more typically used in conjunction with other methods of specifying a product.

Nonrestrictive (or Equal)

The competitive bidding requirements of most governmental agencies have resulted in an approach to specifications in which the writer identifies by name two or more products that meet the requirements of the design. These names are followed by the words "or equal." If a contractor submits a product from another manufacturer, the item is evaluated against those named in the specification. The engineer must then determine whether the item is "equal." Such a determination is somewhat subjective because the items are not expected to be *identical*. Thus, the "or equal" specification is usually combined with a certain amount of performance and product description information (such as a listing of the salient features of each unit with particular regard for maintenance, life, and efficiency) that aids both bidders and engineers in making the "or equal" decision and settling disputes. The evaluation should ensure that all specified performance and quality objectives are being satisfied. This approach has become one of the most common methods of specifying the more complex products.

Federal requirements for naming two products are eased somewhat by Public Law 97-117, which requires naming only one product or equal. However, some EPA regions and some states and local governments still require naming two or more products, so investigate the local requirements.

Proprietary

In proprietary specifications, the product is defined by name. However, only one product is allowed. Obviously, there is no competitive bid in this situation. Proprietary specifications are banned under laws and regulations governing public works projects for most federal, state, and local government agencies.

When a design requires the use of a single source, care must be taken to establish a reasonable price for the product prior to bid. The bid form should be set up in a manner that discloses the value of the proprietary item, thereby allowing the owner to check prices and ensure that the supplier did not take unfair advantage of the noncompetitive situation.

In most instances, an engineer would be hard-pressed to justify an exclusive specification for the equipment commonly found in municipal water and wastewater pumping stations. Exceptions that are sometimes allowed include:

- Instances where the engineer is aware of only one particular product that will meet the project requirements. The specifications should state this fact and should allow an "or equal" in case the contractor finds a product (not known to the engineer) that is thought to meet the requirements.
- Instances where it is more cost-effective for new equipment to match existing equipment.
- Situations where the owner and engineer have agreed that a proprietary process or device is necessary for the purpose of demonstrating or achieving a specific goal.

Generic (or Descriptive)

The generic specification requires the product be described in sufficient detail using, for example, performance, materials, quality control, and tolerances without naming a product. To have a basis for the specification, the specification writer must have identified at least one or more products that will meet the specification. This type of specification is usually lengthy and difficult to write for complex products. It also presents problems in evaluating the product submitted for approval by the contractor.

Reference Standards

The use of reference or industry standards is one of the most common methods of specifying materials or products. As mentioned in Section 28-4, referring to a standard specification may be sufficient for single items. Reference standards for more complex items, such as valves, require more description than a simple reference because there are often options that must be defined to complete the referenced specification. Reference specifications tend to be limited to materials of construction and simple components. Such references, however, are used extensively to describe the quality of more complex items. Thus, references are used in conjunction with the previously described methods.

Be sure to read subsection "Limitations of Published Standards" in Section 28-4 before using reference standards.

28-6. Submittal Requirements

The specifications must maintain a certain level of quality and performance in addition to allowing competition so that the owner can get the best price. The determination of acceptable products and materials is often difficult and controversial. One method of handling products and materials issues is to require the prequalification of manufacturers. Prequalification adds an extra step to the bidding process but provides a very good method of controlling the quality of the products. This approach also eliminates controversy during and after bidding.

The more common acceptance method is through the shop-drawing procedure. Prequalifications do not eliminate the shop-drawing step. Generally, the shop-drawing review is used to ensure that the details of the specifications are being followed. This review may also be used as a basis for rejecting the entire product as submitted by the contractor. Complete rejection of a product at the shop-drawing-review stage can create scheduling and financial problems for a contractor, particularly if the products are a major portion of the contract. The specification should identify products adequately so that contract bids are not based on unacceptable products that must be rejected.

One measure used by some engineers and agencies is to require the contractor or manufacturer to submit copies of relevant specifications as a part of the submittal and to individually indicate by check mark or other means at each specific requirement that the proposed product either meets or deviates from that requirement. This provision forces the individual charged with preparing the submittal to read the specification carefully (often a problem) and reflect on whether the individual attributes of the proposed

product comply. If a deviation from the written specification requirements is proposed, the deviation must be justified in writing to demonstrate that the proposed deviation meets or exceeds the specification requirement. With this requirement in place, a shop drawing reviewer can quickly find the points of deviation and determine acceptability. The authors have found, in over two decades of experience with this provision, that it has resulted in a higher degree of conformity to specification requirements in initial submittals and has reduced both the number of submittal revisions and the time required to reach acceptance. At least one manufacturer has incorporated this technique into its QA/QC program as a means of cutting submittal costs and turnaround time.

In addition to shop-drawing reviews, other quality-control tests and certifications may be required. These controls represent another method of checking product quality. Use care in warranty requirements because unreasonable requirements unnecessarily increase prices.

28-7. References

1. CSI Document MP-PRM2004. *The Project Resource Manual—CSI Manual of Practice (2004),* The Construction Specifications Institute, Alexandria, VA.
   - Module 5. Construction Documents.
2. EJCDC, Engineers' Joint Contract Documents Committee:
   - Document No. C-200, "Suggested Instructions to Bidders for Construction Contracts" (2002).
   - Document No. C-410, "Suggested Bid Form for Construction Contracts" (2002).
   - Document No. C-430, "Bid Bond (Penal Sum Form)" (2002).
   - Document No. C-435, "Bid Bond (Damages Form)" (2002).
   - Document No. C-510, "Notice of Award" (2002).
   - Document No. C-520, "Suggested Form of Agreement Between Owner and Contractor for Construction Contract (Stipulated Price)" (2002).
   - Document No. C-525, "Suggested Form of Agreement Between Owner and Contractor for Construction Contract (Cost Plus)" (2002).
   - Document No. C-550, "Notice to Proceed" (2002).
   - Document No. C-610, "Performance Bond" (2002).
   - Document No. C-615, "Payment Bond" (2002).
   - Document No. C-800, "Guide to the Preparation of Supplementary Conditions" (2002).
   - Document No. C-941, "Change Order" (2002).
3. AIA, The American Institute of Architects, Washington, DC:
   - Document A101, "Standard Form of Agreement between Owner and Contractor—Stipulated Sum" (1997).
   - Document A111, "Standard Form of Agreement between Owner and Contractor—Cost of the Work Plus a Fee, With a Negotiated Guaranteed Maximum Price (GMP)" (2001).
   - Document A114, "Standard Form of Agreement between Owner and Contractor where the Basis of Payment is the Cost of the Work Plus a Fee, without a Negotiated Guaranteed Maximum Price (GMP)" (2001).
   - Document A201, "General Conditions of the Contract for Construction" (1997).
   - Document A701, "Instructions to Bidders" (1997).
4. CSI Documents MP-2-1, *MasterFormat,* 2004 ed.; MP-2-2. *SectionFormat* (1997); MP2-3, *PageFormat* (1999), The Construction Specifications Institute, Alexandria, VA.
5. *ASTM Standards,* 70 vols., American Society for Testing and Materials, Philadelphia, PA (revised annually).
6. *AWWA Standards,* American Water Works Association, Denver, CO (latest edition).
7. AIA. *MASTERSPEC* (Vols. I-V). The American Institute of Architects, Washington, DC (updated periodically).

Chapter 29

Costs

STEFAN M. ABELIN
MARC T. PRITCHARD
ROBERT L. SANKS

CONTRIBUTORS
Kirk Blanchard
Robert H. Brotherton
Ronald W. Duncan
Howard N. Godat
Charles J. Jeelack
James L. Mohart
Michael C. Mulbarger
Donald Newton
Michael R. Olson
Carl W. Reh
William H. Richardson
Ronald Rosie
Larry R. Smith
LeRoy R. Taylor
William R. Taylor
Michael G.Thalhamer
Patricia A. Trager
James C. Well
Eric L. Winchester
John E. Wiskus
Alvord, Burdick & Howson
Black & Veatch, Engineers-Architects
Brown and Caldwell Consultants
Camp Dresser & Mckee, Inc.
CH2M-Hill, Inc.
City of Virginia Beach
Flygt Corporation
Havens and Emerson, Inc.
King County (Washington) Dept. of Municipal Services
Raymond Vail and Associates
Wilson & Company, Engineers & Architects

The cost of a project must be estimated at several stages: (1) at the time the project is conceived and before any funds have been expended; (2) during the planning stage while the facility plan report is being prepared and the best two or three alternatives must be selected; (3) during the design period when the most cost-effective plan must be chosen; and (4) after completion of the plans for use as a basis for both informing the owner of probable cost and judging bids.

Construction cost curves based on data for several types of pumping stations obtained from the contributors to this chapter are presented for rough cost estimating. The construction costs are keyed to a cost index to avoid obsolescence and to facilitate estimating future costs. Formulas for dealing with interest, escalation, and inflation are given, and an example of the economic comparison of alternatives is worked in detail. The level of detail in this chapter is necessarily limited, so a review of the literature before undertaking complex cost analysis is recommended.

29-1. Cost Indexes

Unfortunately, much of the cost data reported in the literature is worthless because of incomplete descriptions or a failure to reference the cost data to a cost index. Some form of cost index must be used to account for inflation. Those of merit include the Engineering News-Record Construction Cost Index (ENRCCI) [1], the Handy-Whitman Index [2], and several EPA cost indexes [3]. The ENRCCI, which is the oldest, is regularly updated, easy to find, and in common use. If the extreme variations in the costs of pumping stations similar in size and type are considered, it seems futile to attempt to improve on the accuracy of the ENRCCI.

Engineering News-Record Cost Index

The ENRCCI, which begins with an index of 100 for the year 1913, is based on constant quantities of

structural steel (weighted 15%), portland cement (2%), lumber (10%), and common labor (73%) in 20 cities. The average of these is considered to be the national average, and plots of yearly national averages are shown in Figure 29-1.

The Engineering News-Record Building Cost Index (ENRBCI) [1] was introduced in 1928 to include the impact of skilled labor, which is weighted 55%, with the remainder assigned as 25% for structural steel, 17% for lumber, and 3% for portland cement.

Both cost indexes are updated weekly in the "Market Trends" section of the *Engineering News-Record* (now called "ENR"). Historical curves of ENRCCI and ENRBCI and the materials component are shown in each mid-March "Quarterly Cost Report" along with forecasts to the end of the year. Indexes 12 months ahead are predicted in the mid-December "Quarterly Cost Report." The cost indexes can be found on the Internet.

Using the ENRCCI

To use the ENRCCI (from Figure 29-1), follow these steps:

- Estimate the date of construction and extrapolate the curve to that date.
- If desired, correct for the region of the site by choosing the ENRCCI ratio (F' in Equation 29-1) for the nearest city as compared to the national average.
- If the construction site is more than 30 mi from an urban area, labor cost increases about 10% for each additional half-hour of driving time. Investigate local union rules for more accuracy. As labor is roughly half of the total cost of construction, add about 5% for each additional half-hour of driving time beyond the 30-mi limit (F'' in Equation 29-1). If the site is in a congested area (such as in the center of a city), the cost index increases greatly

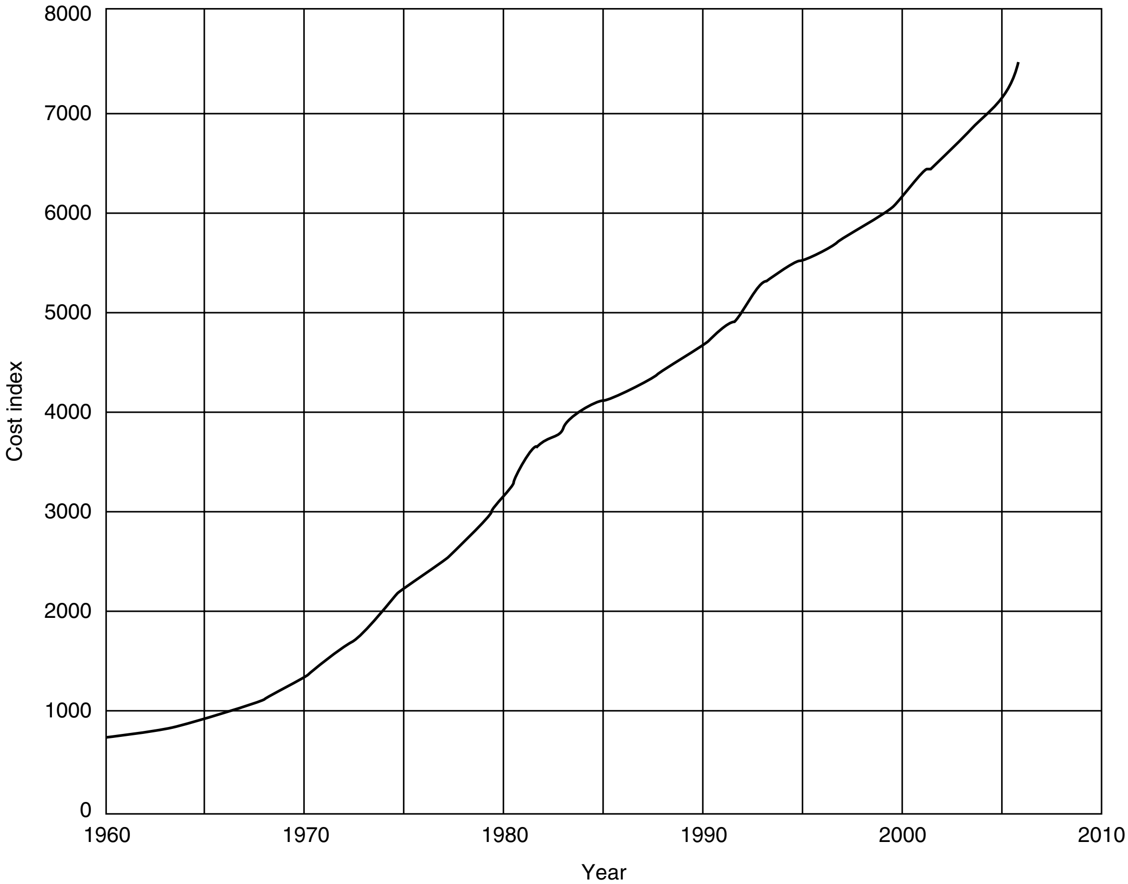


Figure 29-1. Inflation as measured by the ENRCCI.

due to traffic interference, limited work area, and underground and overhead interferences. The advice of experienced local contractors is helpful in estimating the impact on costs.

- Calculate the final cost index as follows:

$$C_{\mathrm{f}} = C_{\mathrm{p}} \frac{\mathrm{ENRCCI_f}}{\mathrm{ENRCCI_p}} \times F' \times F'' \qquad (29\text{-}1)$$

where the subscript f represents a future date, the subscript p represents a past date for which the construction cost is known, and F' and F'' are correction factors for the region and locale. The terms F' and F'' indicate a supposed accuracy that is entirely overshadowed by the construction conditions, the designer's concept of appropriate design, the amount of instrumentation, the addition of standby power, and, especially, the bidding climate factors that combine to make prices scatter far more than the effect of terms F' and F''.

29-2. Cost Curves

The data for the cost curves presented in this section were obtained from a survey of the contributors to this chapter. These cost data represent construction between 1966 and 1987, but more than 90% of the data are more recent than 1974. All costs were corrected to an ENRCCI of 4500 by means of Equation 29-1; the trivial terms, F' and F'', were omitted in consideration of the scatter that often exceeded 300%. Only data for "typical" pumping stations are included. Data for "atypical" pumping stations (such as those excessively deep or those built to unusual specifications) were either discarded or are explained in the text. All costs are contract prices for construction plus costs for extra work, and all are limited to the construction allocated only to the pumping station (and excluding costs of other works such as force mains and treatment plants). To obtain the total cost of a pumping station, charges for engineering and legal fees, land, administration, and interest during construction must be added.

No distinct pattern was found to explain the scatter—neither the number of pumps, variable- versus constant-speed drivers, the presence or absence of standby power, high versus low head, nor difficult foundation conditions. The factors discussed at the end of Section 29-1 account for the variations, and these factors are not quantifiable. Deep foundations, high head, standby power, and variable speed, however, tend to increase costs toward the upper limit lines in the figures.

The data are limited, and the cost envelopes in the following figures should be used with caution. In particular, it should not be assumed that the cost "curves" continue in a straight line beyond the limits of the data points. It seems logical to suppose that the cost curves become flatter as the size of the pumping station decreases below about 23 m^3/h (100 gal/min), although no data are presented here to support such a conclusion. Costs in Figures 29-2 through 29-9 are given in k$ (thousands of dollars).

Wastewater Pumping

Pumping stations equipped with standby power are shown by open circles in the figures; those without standby power are indicated by smaller solid dots. Of stations for which complete data on standby power were given, 35% had diesel engine-generators, 30% had dual electricity sources, 15% had gas engine-generators, 10% had propane and diesel engine-generators, and 10% depended on portable engine-generators. The lines depicting upper and lower limits are merely estimates that indicate a likely range of probable costs.

Custom-Built Wet Well–Dry Well Stations

The numbers adjacent to plotted points are TDH in feet. Although there is no clear correlation between head and cost, data for pumping stations with a TDH above 21.3 m (70 ft) tend to lie along the upper limit of costs in Figure 29-2. Of the three points well above the upper limit, one is for a station (TDH = 47) that required automatic screens, odor control, and, during construction, blasting. Another is for a station (TDH = 55) that included comminutors, chlorination equipment, and provision for future telemetering. The third station (TDH = 38) had odor-control equipment and room to increase the number of pumps from four to six. The cluster of 25 points at the lower left is for one location, Virginia Beach, where nearly all of the pumping stations were built by developers to city requirements.

Submersible Pumps

The TDH for half of the stations is less than 15.2 m (50 ft). Data points for all but two stations with a TDH of more than 15.2 m (50 ft) lie close to or above the upper limit line shown in Figure 29-3.

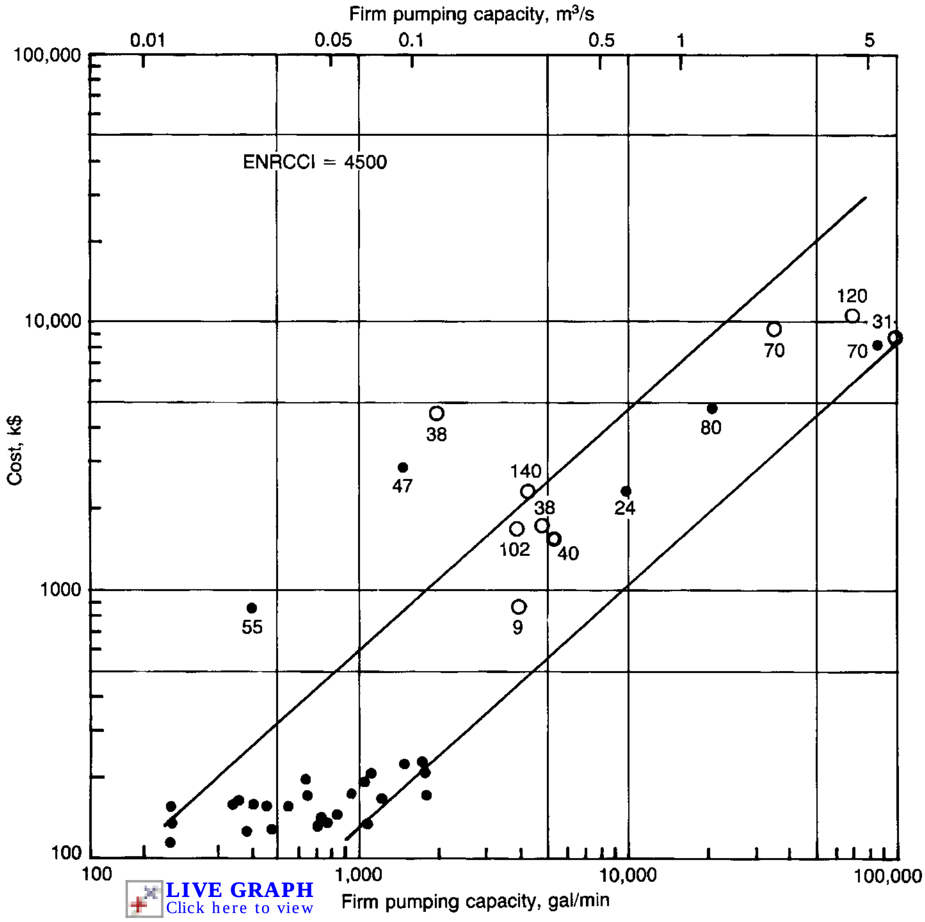


Figure 29-2. Construction costs of custom wet well–dry well wastewater pumping stations. No standby power = solid circles; has standby power = open circles. The numbers are TDH in feet.

Self-Priming Pumps

No data are shown for self-priming pumps at grade, but construction cost is slightly less than for submersible pumps. The maintenance of self-priming pumps is lower than that for submersible pumps because the equipment is at grade, the access is easy, the wet well can be sealed so that there are no fumes to corrode electrical equipment, and there is no need for ventilation other than louvers for cooling the motors [4].

Prefabricated Pumping Stations

Prefabricated pumping stations can be obtained in a number of types: (1) wet well-dry well with flooded suction (i.e., the dry well is as deep as the wet well); (2) wet well–shallow dry well with suction lift; (3) wet well with a submerged pump connected to the motor with a long shaft; (4) wet well with suction lift to self-priming pumps at grade; and (5) submersible pump and close-coupled (submersible) motor. The data for prefabricated pumping stations in Figure 29-4 are for the wet well–dry well type except for two, which utilize self-priming pumps, and one, which is an enclosed (Archimedes) screw pump. With one exception, the variation in heads does not affect the costs. The high cost of one pumping station is due to the inclusion of headworks (comminutor, Parshall flume, manual screen bypass). The Archimedes screw pump is included with prefabricated stations because it is an enclosed screw and, thus, "prefabricated" to a degree.

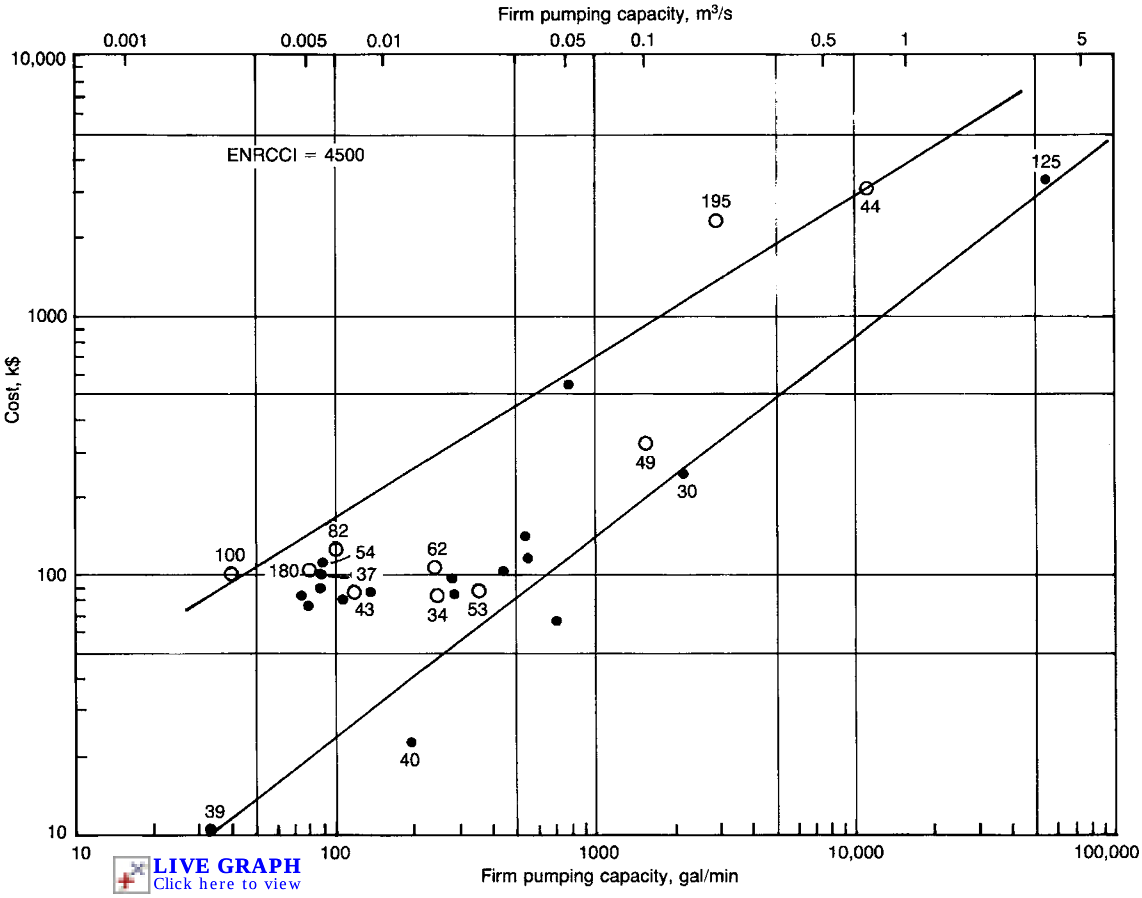


Figure 29-3. Construction costs of submersible-pump wastewater pumping stations. No standby power = solid circles; has standy power = open circles. The numbers are TDH in feet.

Water Pumping

In general, high speed (1800 rev/min) is common up to 150 kW (200 hp), and 1200 rev/min is usual for larger pumps.

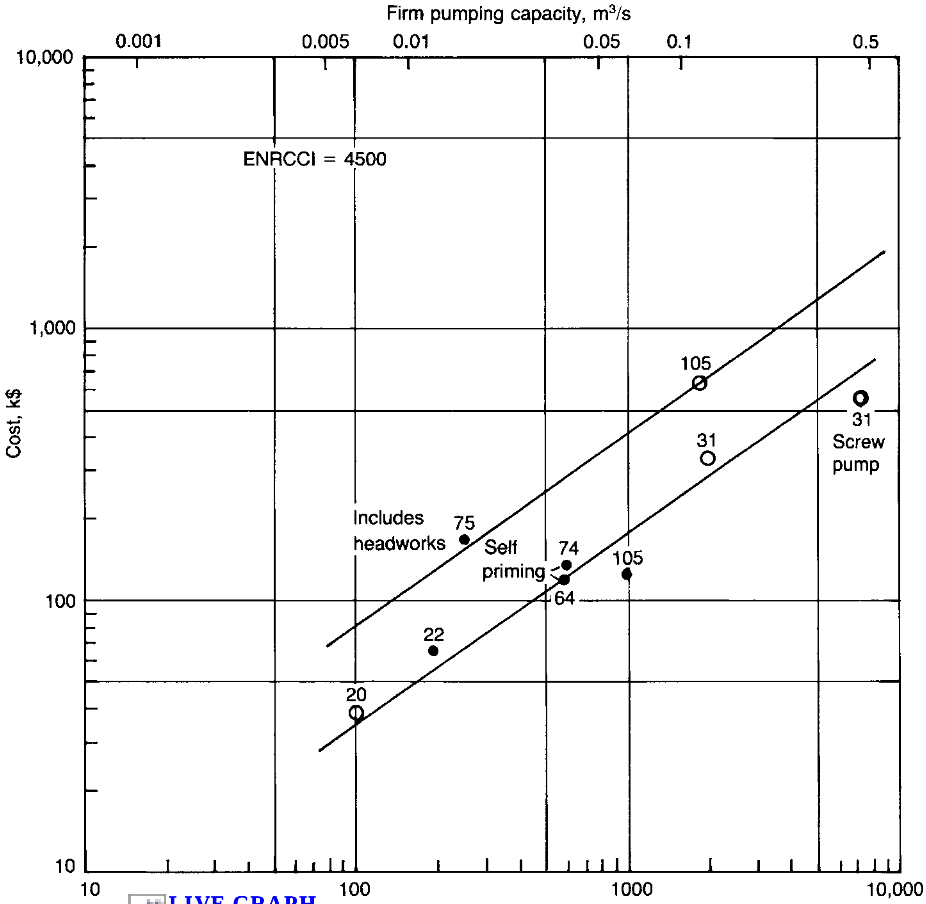
Raw Water Pumping

All of the pumping stations shown in Figure 29-5 were constructed in lakes or on the banks of rivers where deep structures, sheet piling, or coffer dams were required. The high cost of pumping station No. 1 is attributed (1) partly to facilities designed for two pumps with space for four more and a dual electrical feeder for standby power, and (2) partly to minimal bid competition and the construction of a lake intake. Pumping stations Nos. 3 and 4 have traveling screens, No. 3 has a massive (3.3-m- or 11-ft-thick) foundation of tremie concrete, and No. 4 has prestressed rock anchors.

The TDH varied from 15.8 to 73 m (52 to 240 ft). The stations with the lowest relative cost (Nos. 2 and 6) had nearly the highest and lowest heads, whereas the stations with highest relative costs (Nos. 1, 3, 4, and 5) also had the lowest (No. 1) to the highest (No. 5) heads. Head is evidently a minor consideration in raw water pumping.

Service Pumping

The TDHs for the finished water pumping stations shown in Figure 29-6, which vary from 43 to 114 m (140 to 375 ft), seem to have no correlation with cost. The station with the highest head was moderate in relative cost, and the station with the lowest head was the most expensive. The remaining data fit a similarly



LIVE GRAPH
Click here to view

Figure 29-4. Construction costs of prefabricated wastewater pumping stations. No standby power = solid circles; has standby power = open circles. The numbers are TDH in feet.

random pattern, and costs do not appear to correlate with other features such as standby power, ventilation, or foundation problems.

Booster Pumping

The TDHs for the booster pumping station costs in Figure 29-7 vary from 9.1 to 95 m (30 to 310 ft). Again, cost appears to be unrelated to either head or the inclusion of standby power. Seven pumping stations are in-line boosters, and five are distribution boosters. The type of booster also seems to have no effect on cost.

Well Pumping

The total costs of pumping stations, including the pumps and drilling and casing the well, are shown in Figure 29-8. Adequate data for the costs excluding the well were not available. Standby power in pumping station No. 5 consists of a natural gas engine-generator. Standby power for No. 3 is a direct-drive natural gas engine with a manual switchover. In general, any correlation between depth and cost is poor.

Summary

Median lines for each type of pumping station shown in Figures 29-2 through 29-8 are shown for comparison in Figure 29-9. If the upper and lower limits in Figures 29-2 through 29-8 were plotted, the costs for all of the pumping station types would overlap; a custom station, therefore, could be less expensive than a prefabricated or submersible station.

If the cost for one size (capacity) of a pumping station is known, the cost for another size (of the same type) can be found by means of the "six-tenths

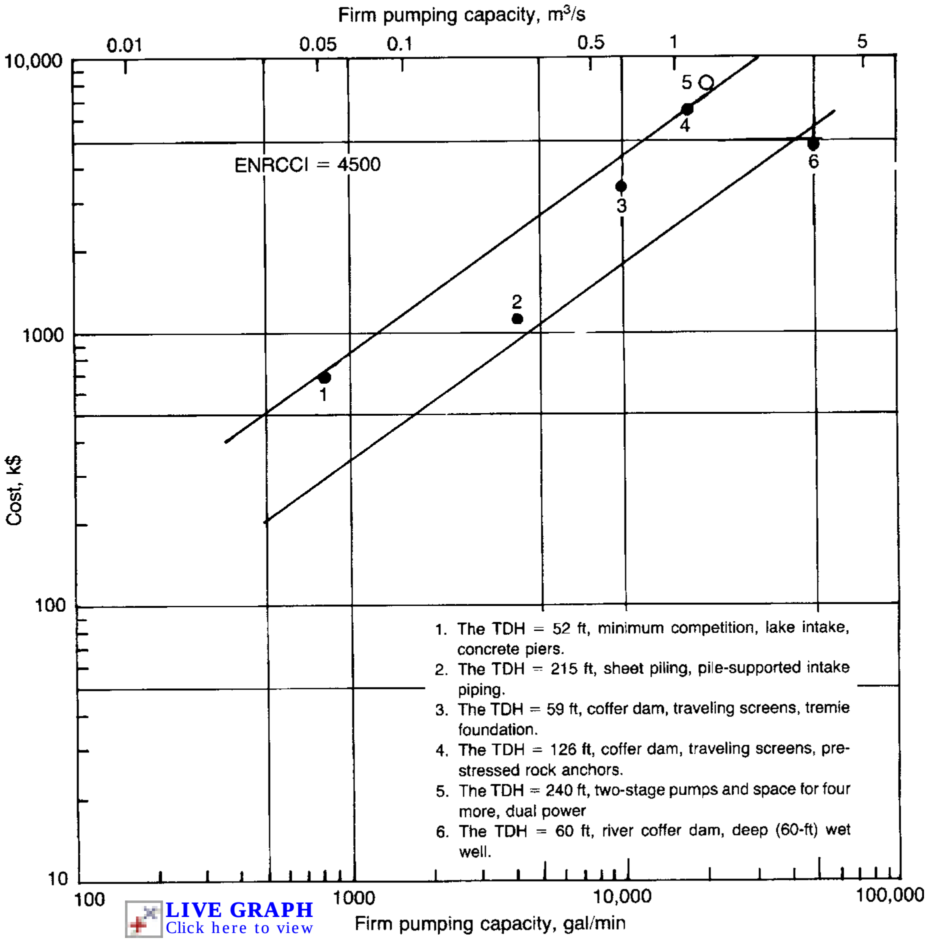


Figure 29-5. Construction costs of prefabricated wastewater pumping stations. No standby power = solid circles; has standby power = open circles. The numbers are TDH in feet.

factor." Simply multiply the known cost for one facility by the ratio of sizes raised to the exponent 0.6. For most of the plots of Figure 29-9, however, a more realistic exponent is 0.75. For water wells, the exponent is about 0.42.

29-3. Maintenance and Energy

The cost for maintenance is elusive partly because (1) it depends upon the owners' policies, which vary greatly from utility to utility (and also with time and personnel), and (2) so few owners have reliable, accurate records of maintenance labor and supplies for each pumping station. Because pumping stations differ considerably in maintenance requirements, records for several stations lumped together are of little use. The true cost of labor should include fringe benefits, supervision, costs for support (tools, vehicles, insurance, etc.), and the expenses associated with maintaining support facilities such as shops, clerks, records, etc. Costs of operating and maintenance labor, material, and supplies for wastewater pumping are shown as curves by Patterson and Banker [5], but these data should be used with caution because (1) labor time can easily double the values shown, and (2) there is no correlation with owners' policies or types of stations.

Maintenance of equipment is also equipment-specific. For example, the maintenance for a wound-rotor motor with its brushes and slip rings is much greater than for a squirrel-cage motor. A self-priming pump can be repaired by the maintenance crew, but, to prevent voiding the contractor's guarantee, a submersible pump might have to be shipped to a service center with shipping costs added to the high (perhaps

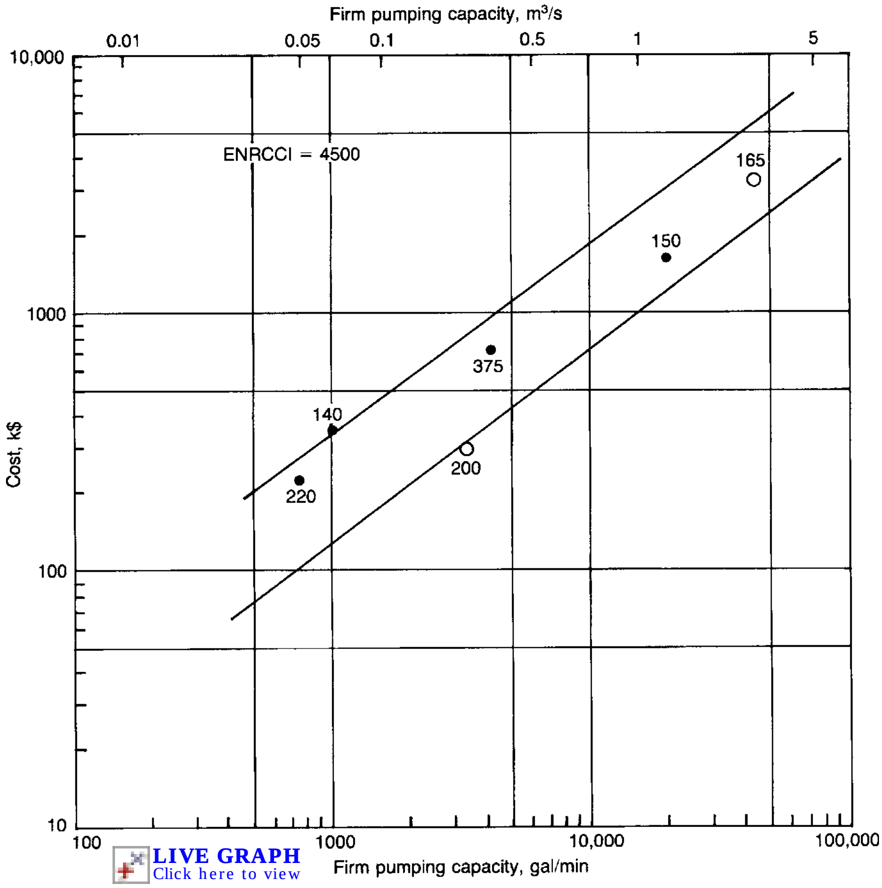


Figure 29-6. Construction costs of service water pumping stations. No standby power = solid circles; has standby power = open circles. The numbers are TDH in feet.

30% of the original purchase price) cost of repair. Repairs made to any equipment at the job site by factory-authorized service personnel are likely to be extremely expensive, but rational maintenance costs can be assessed by investigating service contracts.

Excessive dependence on the effect of either size or purchase price in estimating maintenance costs is unwise. For example, the maintenance costs for a large pump might actually be less than for a small, less expensive pump because of better accessibility (and, hence, reduced labor time, for replacing packing, for example) and because of the lower speed of the larger pump. Finally, two identical machines in similar circumstances may have, inexplicably, quite different maintenance needs and costs. However, to ignore maintenance costs because they are difficult to evaluate is to skew cost comparisons and decisions. To put it simply, investigate as necessary and do the best you can. Some equipment (such as solid-state electronic controls) become obsolete, so a sinking fund to replace such items in a reasonable period (say, 8 to 15 years for electronic equipment) should be added to annual maintenance costs.

Pumping station design is (or should be) profoundly influenced by the cost of power. Dealing with energy costs can be very complex. Electric power rates vary with the classification of the user, the amount of power used, and, often, with the time of day, the season of the year, the power factor, and the power demand. The rates differ by as much as 500% or more depending on location. Furthermore, the escalation of cost may exceed, keep pace with, or even lag behind inflation. The costs of natural gas and diesel fuel are less complex and subject to fewer influences, but they still depend on location and politics. Electric power and gas utilities are the most reliable source of information on past and present costs of electricity and gas. Some utilities make careful predictions of future power costs. For critical problems, a good procedure is to present the appropriate utility with a

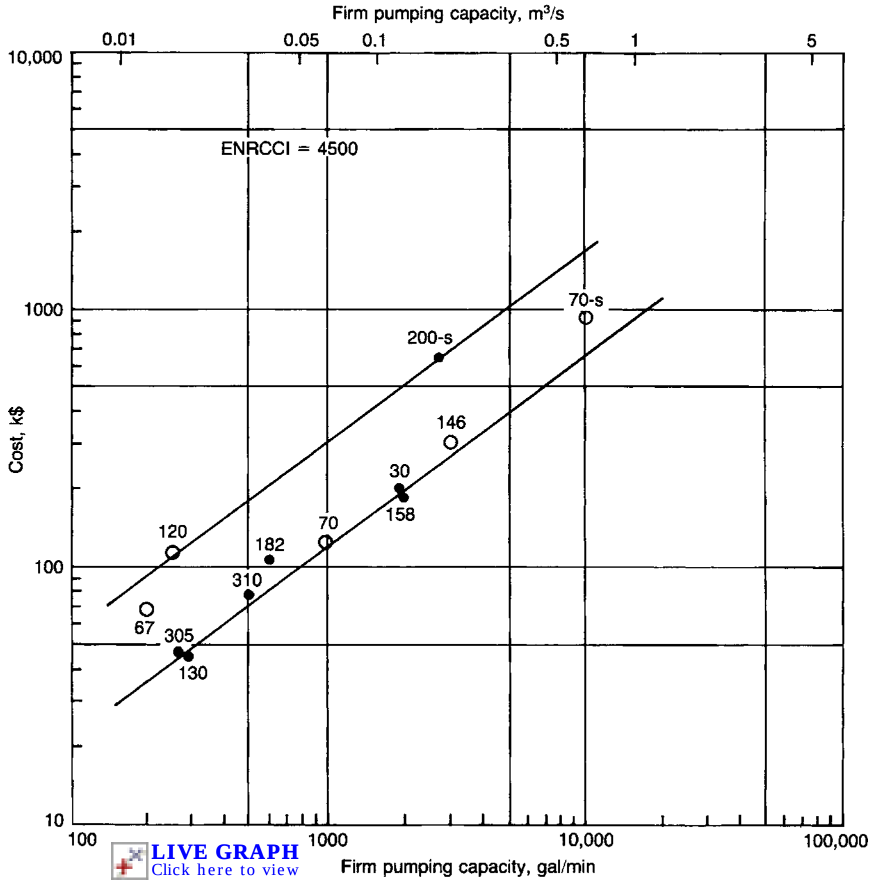


Figure 29-7. Construction costs of booster pumping stations. In-line booster = solid circles; residential booster = open circles; standby power = –s. The numbers are TDH in feet.

complete spectrum of power needs and let them calculate the annual cost. By following their method of calculation, the designer can then accurately evaluate power costs for alternatives.

29-4. Interest Formulas

The comparative worth of two or more alternative plans can be evaluated on the basis of either annual cost or present worth.

Equivalence

Costs can be compared only on some equivalent basis such as present worth or a uniform series of annual payments. The two basic formulas for the relationship between present worth, future sum, and periodic payments are

$$F = P\,[(1+i)^n] \tag{29-2}$$

$$F = A\left[\frac{(1+i)^n - 1}{i}\right] \tag{29-3}$$

where i is the interest rate (in decimals or percentage/100) per payment period, n is the number of payment periods, P is a present (lump) sum of money, F is the future (lump) sum of money at the end of n periods equivalent to P at interest rate i, and A is the uniform end-of-period (usually annual) payment equivalent to a present (lump) sum. If the payment period is, for example, monthly, the annual interest rate is divided by 12 and n is multiplied by 12.

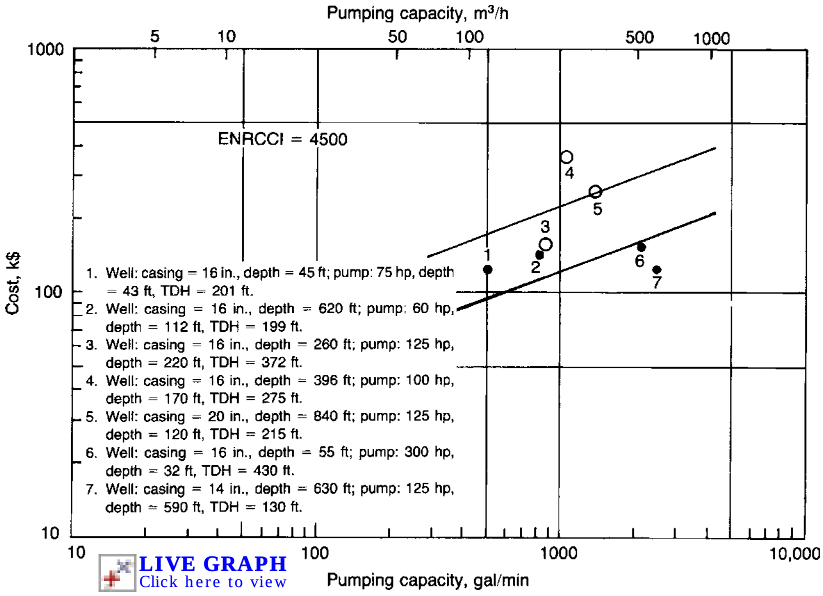


Figure 29-8. Construction costs of water wells and pumping stations. No standby power = solid circles; has standby power = open circles.

Other formulas can be derived algebraically from Equations 29-2 and 29-3 as follows:

$$P = \frac{F}{(1+i)^n} = A\left[\frac{(1+i)^n - 1}{i\,(1+i)^n}\right] \tag{29-4}$$

$$A = P\left[\frac{i\,(1+i)^n}{(1+i)^n - 1}\right] = P\left[i + \frac{i}{(1+i)^n - 1}\right] \tag{29-5}$$

The bracketed terms in these equations are named as follows:

- *Equation 29-2:* single-payment compound-amount factor (spcaf)
- Inverse of spcaf: single-payment present-worth factor (sppwf)
- *Equation 29-3:* uniform-series compound-amount factor (uscaf)
- Inverse of uscaf: sinking-fund deposit factor (sfdf)
- *Equation 29-4:* uniform-series present-worth factor (uspwf)
- Inverse of uspwf: capital-recovery factor (crf).

The annual cost, A, in Equation 29-5 is limited to the debt service (retiring the bonded indebtedness), but it can be expanded to include annual operation (O) and maintenance (M) costs as follows:

$$A = P\left[i + \frac{i}{(1+i)^n - 1}\right] + O + M \tag{29-6}$$

and this is a most useful formulation for comparing alternatives.

Arithmetic-Series Gradients

Costs (or quantities such as power usage) sometimes increase uniformly with time. Let the increase in cost (or quantity) be G and the constant cost be C. Then the total payments for each period are

- C at the end of year 1
- $C + G$ at the end of year 2
- $C + 2G$ at the end of year 3
- $C + (n - 1)G$ at the end of year n.

The total equivalent uniform annual series of payments or quantities is

$$A = G\left\{\frac{1}{i} - \left[\frac{n}{(1+i)^n - 1}\right]\right\} + C \tag{29-7}$$

The term in braces is called the arithmetic-series factor (asf). The quantity A can be converted to present worth by means of Equation 29-4.

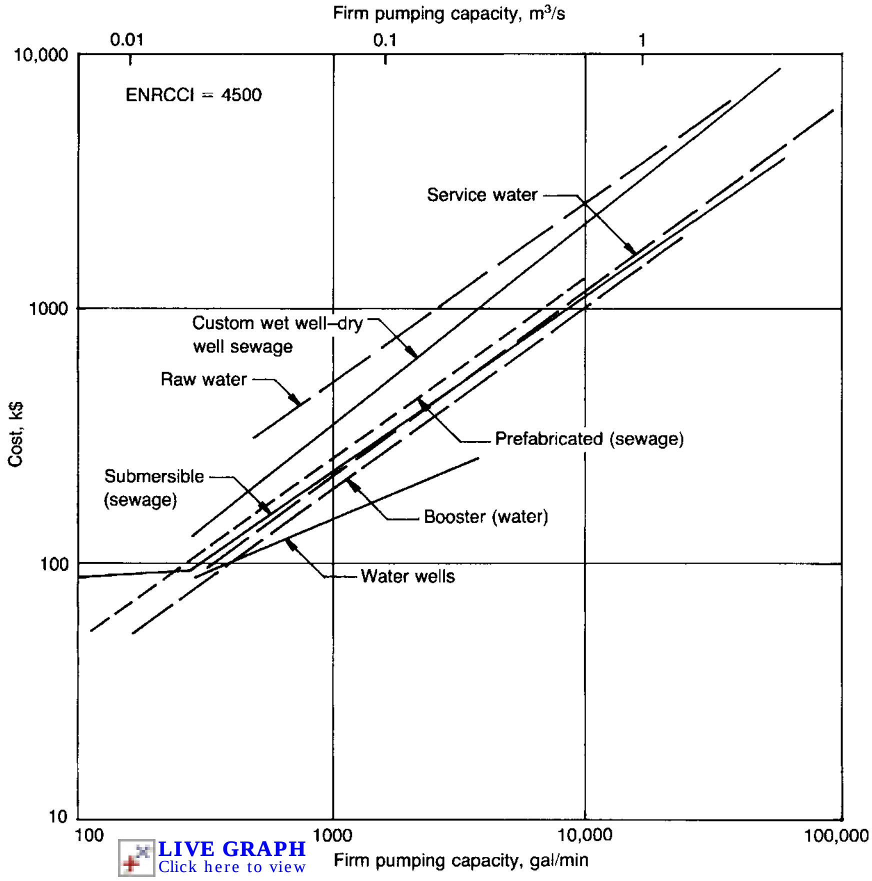


Figure 29-9. Comparison of average construction costs for pumping stations.

Geometric-Series Factor

If the costs or quantities increase geometrically with time, the total payments for each period are

- C at the end of year 1
- aC at the end of year 2
- a^2C at the end of year 3
- $a^{n-1}C$ at the end of year n.

The geometric increase $(aC + a^2C + \ldots + a^nC)$ can be converted to present worth by using the equation

$$P = C\left[\frac{a^n - (1+i)^n}{a - 1 - i}\right] \tag{29-8}$$

The term in brackets is called the geometric-series, compound-amount factor (gscaf). The present worth, P, can be converted to a uniform annual amount, A, by means of Equation 29-5.

The derivations of these equations, along with a complete discussion of their use, are given by Taylor [6].

Inflation and Escalation

Inflation is the decrease in the value of money. Escalation is the increase in price of a particular commodity of concern, such as energy. Although both are common experiences, there is some uncertainty about how they should be treated (if at all) in an economic analysis. Equations 29-2 through 29-6 can be modified for inflation, or inflation can be calculated from Equation 29-7 or 29-8, but predictions of the inflation rate are apt to be faulty, and, in many kinds of problems, the results are not altered by inflation.

If the escalation of energy costs equals the inflation rate, it, too, can be ignored. But if it is expected to exceed inflation, there is an undue penalty on a

more costly but more efficient pump. White et al. [7] proposed the concept of "constant-worth" dollars expressed by the equation

$$C_t = \frac{T(1+e)^t}{(1+j)^t} \tag{29-9}$$

in which C_t represents purchasing power at time t relative to year 0, T is the fixed, constant-worth dollar amount, e is the escalation rate in percentage/100, and j is the inflation rate in percentage/100. Note that if escalation equals inflation, C_t equals T.

Dell'Isola and Kirk [8] proposed modifying Equation 29-5 to include escalation as follows:

$$A = P\left\{\frac{\dfrac{1+m}{1+i} - 1}{\left(\dfrac{1+m}{1+i}\right)\left[\left(\dfrac{1+m}{1+i}\right)^n - 1\right]}\right\} \tag{29-10}$$

where $m = e - j$ and e and j are defined above. If the escalation rate equals the inflation rate, Equation 29-10 reduces to Equation 29-5. Consult the literature for more complete discussions.

For finding the present worth of energy (or, more exactly, a "discounted break-even value") in an escalating market, Wood [9] proposed

$$P = E\left[\frac{1 - x^n}{(1-x)y^n}\right] \tag{29-11}$$

where $x = (1+e)/(1+j)$, $y = (1+i)/(1+j)$, and E is the present annual cost of energy. If the escalation rate equals the inflation rate, the equation becomes indeterminate. The formula is not strictly comparable to Equation 29-10 because the derivation is based on payment at the beginning of each pay period, whereas all previous formulas are based on payment at the end of each pay period. Instead of Equation 29-10 or 29-11, however, Equation 29-7 or 29-8 can be used to account for inflation and escalation as well as for an increase in energy needs provided that the increases occur in either a straight-line or a geometric progression. For irregular increases in the quantity of energy to be used and to account for inflation, escalation, or both, set up a table of computations in which all quantities are reduced to an equivalent present worth.

For projects funded or controlled by the U.S. EPA, comparisons of alternatives may be required to follow a specified format. For example, inflation (except for land and energy) should be ignored [10]. Historical bases for the escalation of energy costs can be obtained from public utility commissions and from public relations personnel of power utilities.

The cost of energy over the life of a pumping station is surprisingly high and, hence, has a profound influence in any objective analysis on the selection of pumps and drivers. In today's labor market, maintenance is also likely to be a major cost. To ignore either is to skew the results of a comparative cost analysis. The cost of a pump is roughly proportional to the square of the impeller diameter, but, because energy is approximately proportional to the fifth power of the impeller diameter, the ratio of energy to capital cost increases rapidly as the pump becomes larger. Evidently, first cost is more important for small pumps, whereas for large pumps the cost of power may be many (up to 50) times the cost of the pump itself.

29-5. Cost Estimates

The types of cost estimates discussed in this chapter include:

- Order-of-Magnitude
- Budget
- Definitive
- Comparative
- Life-cycle.

Order-of-Magnitude Cost Estimates

There are various methods available to create an order-of-magnitude estimate for a pumping system. The use of Figures 29-2 through 29-9 is one.

Budget Cost Estimates

At the selection phase of a proposed pumping system project, a more detailed budget estimate is created to help identify the most cost-effective system. The types of estimating methods used in the past to establish these budgetary costs have had a history of minimal accuracy. With today's modern computers, the means to achieve a higher level of accuracy in establishing the budgets for various types of pumping systems is possible. Accuracy is important because owners use the costs estimated at the early stages of the project to establish the budgets for the project. By using the latest computer technology, the level of accuracy that can be obtained in the budgetary phase can be surprising. For example, a computer-based program called *BAC-PAC*© (a cost estimating program of Brown and Caldwell) [11, 12] has been used to obtain budget estimates

that (since 1991) have had an accuracy between +5% and −15% of the final, in-place construction costs.

Definitive Cost Estimates

After the budgetary costs have been established and the design becomes more definitive, the cost estimates also become more definitive and more accurate. Using the latest computer technology, it is relatively easy and quick to upgrade the budget cost estimates and to use these improved editions to help keep the design within the budget or to determine the effect on costs that various alternatives can have. For example, based on various soils conditions, *BACPAC*© computes the thickness and cost of concrete walls together with the amount and cost of reinforcement automatically—the reason for omitting the thickness of walls in Figures 29-12 and 29-15. *BACPAC*© cost estimates for final designs have had an historical accuracy (since 1991) between +2% and −5% of final, in-place costs.

During the design process, it is important to make cost estimates from time to time to inform both owner and designer of the consequences of various design decisions on the cost of the project and so avoid overruns. One system for coping with this objective is the "Design-to-Cost" system described in Section 1-10.

Regardless of the system used, the final cost estimate is based on a complete listing of the quantities of all materials described in the plans and specifications multiplied by the prices for materials and labor plus an allowance for the contractor's profit and overhead. Even with a computer program, good estimating requires great skill and an intimate knowledge of construction practices and market conditions. An apprentice, however, if careful to miss nothing, can make reasonably acceptable estimates with the help of current publications that are updated annually. Useful and authoritative publications include Means [13, 14] and the Dodge Cost Systems [15]. Means is particularly useful for buildings, and the Dodge Cost Systems includes heavy construction and remodeling. Walski [16] discusses costs for planning. European engineers may find the publications of the Water Research Centre [17] useful. Articles such as that by Hall et al. [18] can remain useful for many years if the costs are updated by the ENRCCI [1].

Obtain prices from manufacturers wherever possible. List prices are usually reduced by large percentages that depend on the customer. The "customer" for a pumping station is the general contractor or a subcontractor. Shipping and installation costs (including profit and overhead) must be added to the manufacturer's cost. Installation involves (1) anchoring the unit in place; (2) "hooking up" to pipes, conduits, and electricity; and (3) start-up and testing. Installation costs are often surprisingly high even for a skid-mounted unit. Depending on the size of a machine or device and the complexity of the hook-up, typical installation costs might normally range from 10 to 40% of the purchase price, but the cost can be either less or more (see Means [13, 14] or Dodge [15]). Costs are affected by many factors, including:

- Competition and bidding climate
- Accessibility to a labor market [remote locations cause an increase in labor prices—about 10% for every 48 km (30 mi) of travel]
- Foundation conditions (dewatering, sheet piling, unstable soils, rock excavation)
- Earthquake resistance
- Surrounding conditions (such as nearby buildings, buried conduits, etc.)
- Need for standby power
- Whether the owner is a private or public agency.

Local contractors can be of great help in assigning values to these considerations—if their cooperation can be secured—but there is no substitute for developing some appreciation for construction practices and the problems confronting a contractor who must do a specified job for a fixed price. Because of planning, land acquisition, legal fees, owner's involvement, interest during construction, administrative costs for the project, start-up, operational training, and the costs for spare parts, the total cost to the owner is about twice the contract bid price.

Comparative Cost Estimates

A rational choice between alternative designs can be made only when life-cycle cost estimates are compared. *Basing decisions on capital (construction) costs only is misleading*. However, to obtain a direct measure of the relative costs between different design alternatives, it is not necessary to include features common to all alternatives. Finally, to reach a decision on the best alternative—not necessarily the cheapest—the intangibles (which cannot be quantified in dollars) must also be considered. Intangibles include (but are not limited to) historical reliability, construction time, operational ease, simplicity, aesthetics, compatibility with other facilities, familiarity, and the preferences of the owner, operators, and designer.

Life-Cycle Cost Estimates

Life-cycle cost is the combined cost of engineering, design, construction, and equipment procurement to construct and activate a pumping station during year 1 plus the accumulated operational costs of energy, maintenance, and repair for the established life of the pumping station. Depending upon the project circumstances, the life of the pumping station can be based on either economic or technical considerations. Important factors include cost of money (interest), the need for equipment repair, inflation, and future energy costs. The initial cost together with each of the annual costs (corrected to present worth value by means of the equations in Section 29-4) yields the total life-cycle cost. Although the life-cycle cost may not be exact, it is the only way in which the relationship between different alternatives can be objectively assessed.

Unfortunately, owners are seldom willing to pay for a life-cycle cost analysis that is truly complete and realistic , and it is customary to consider capital costs only and, even so, to ignore the effects of the project on the cost of the whole system. (For example, savings using constant-speed (C/S) instead of variable-speed (V/S) pumping may stress—and hence increase the cost of—the treatment process downstream.) The designer, as a consequence, is often required to guess at answers, and such guesses can be misleading. In Example 29-1 of the first (1989) edition of this book, stations with C/S, V/S with AFDs, and V/S with eddy-current couplings for pump drivers were compared for an 880-L/s (20-Mgal/d) wastewater pumping station. In view of the inefficiency of eddy-current couplings, the results were surprising. The comparative costs were 126, 107, and 100, respectively. Thus, the lowest cost was for eddy-current couplings. The results might have been different if energy costs had been higher than those assumed—essentially a peak power rate of 6.37 ¢/kW · h and an off-peak rate of 3.15 ¢/kW · h. Nowadays, the prices of eddy-current couplings have increased, and the number of manufacturers has decreased. However, the prices of AFDs have decreased, so in today's market the AFDs would probably be the most cost effective.

Example 29-1
Life-Cycle Cost Comparison of Three Alternative Wastewater Pumping Stations

---

Two common problems today involve: (1) the choice between using a few large pumps and more smaller ones (with the consequent increase in piping and space requirement), and (2) the choice between using C/S and V/S drives. Both problems as applied to submersible pumps are illustrated in this example. The example is hypothetical, but the life-cycle cost analyses are realistic and supportable. Three options are compared:

- Three V/S pumps (2 duty, 1 standby) of equal size
- Three C/S pumps of the above size, plus a smaller pump
- Three C/S pumps of equal size.

To avoid tedious repetition in tables, this example is worked only in U.S. customary units.
Problem: A new pumping station with a trench-type sump is to replace an old, outdated station in a fully built-up, older residential neighborhood. Any potential increase in inflow is expected to be offset by water conservation measures. Therefore, it is assumed that the flow duration data given in Figure 29-10 will essentially hold steady for the life of the station.

Flow Rates, Heads, and Controlling Elevations

- Minimum: 1.9 Mgal/d (1300 gal/min). See also Figure 29-10.
- Average: 5.4 Mgal/d (3750 gal/min).
- Maximum: 13.3 Mgal/d (9230 gal/min or 20.6 ft^3/s) for two duty pumps.
- Sewer: 36 in. at 0.1% slope. Invert elevation 24.05 ft at 60 ft from wet well.
- Station piping: typical piping to manifold. Parabolic curve for friction headloss = 9.0 ft at 7000 gal/min.
- Force main: 20-in., mortar-lined, ductile iron pipe, 4000 ft long. Discharge elevation: 60.10 ft.
- Friction headloss based on $C = 135$ is 50 ft at 10,000 gal/min.

Other Data

- Design life: 25 yr
- Interest: 7%

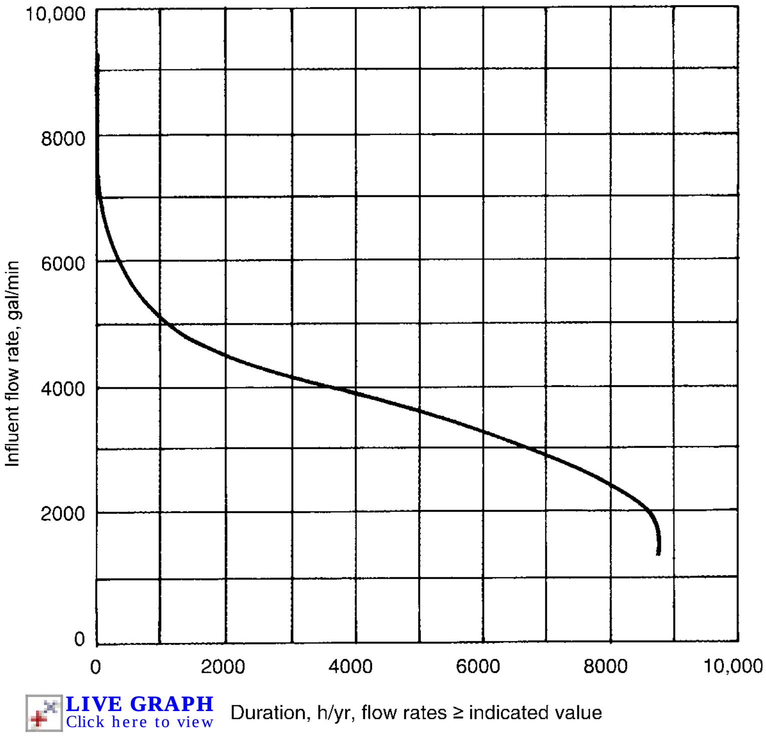


Figure 29-10. Influent flow rate duration curve.

- Inflation: 4.5%
- Energy: \$0.06/kW · h with annual inflation increase of 5%.

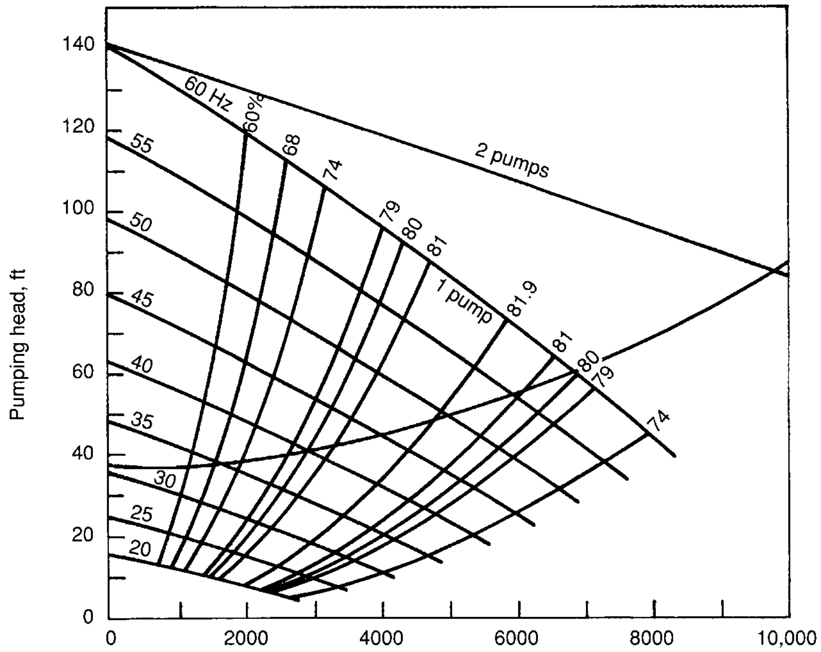
Design Option 1 (Three V/S Pumps)

Three 12-in. Flygt CP3312, 175-hp, close-coupled submersible pumps are selected. The system curve for the force main is shown in Figure 29-11 together with pump characteristic curves. Station piping headlosses for a pump are assumed to follow a parabola to a loss of 7 ft at a discharge of 6000 gal/min.

To make sure that the station can meet its requirements, the H-W *C* is often considered to vary between 145 or 150 and 120. For a life-cycle cost analysis, however, the use of excessive safety factors is misleading, and the best—not the safest—values are needed, so the chosen value of *C* is 135. Two pumps together can discharge a flow rate of 9860 gal/min at 89.5 ft of head, whereas a single pump can discharge a flow rate of 6790 gal/min at 69.6 ft of head.

Each pump is equipped with a top-of-the-line pulse-width modulated (PWM) type of AFD to produce the highest drive efficiencies available to date. Because of their sensitivity to heat, fumes, and dust, the AFDs are housed in a small air-conditioned building where incoming air is filtered to remove all dust and contaminants, particularly H_2S. Motors controlled by AF converters are subjected to severe service and are consequently derated. Despite the derating, they require more maintenance than do motors for C/S pumps.

The principal advantage of V/S pumping is that discharge equals inflow at all times, and thus there are no sudden changes of the inflow to the treatment plant—an advantage that may be serious or negligible depending on the relative sizes of pumping station and treatment plant and the sensitivity of the treatment plant facilities to shock loads. The other advantages are the increased operational flexibility, the reduced energy consumption, the smaller size of the wet



LIVE GRAPH
Click here to view

Figure 29-11. Pump and system curves for variable-speed pumping.

well, and a shallower excavation if the wet well is to be self-cleaning. The disadvantages are the substantial costs of the AFDs, the increased control system complexity, the potential problems with harmonics and power quality, plus the increased maintenance both for the AF converters themselves and for the air-conditioning unit required to protect them from dust, contaminants, and heat as well as increased maintenance for the pump motors. Furthermore, AFDs require highly skilled personnel for servicing and repairs. Overall station reliability is reduced somewhat, but AFDs have become more dependable in recent years. High-quality equipment is reliable when well maintained and housed in an air-conditioned space.

The design follows the guidelines in Chapter 12. At an entrance velocity of 5.5 ft/s, the diameter of the suction bell for 6800 gal/min (Figure 29-11) would be 1.88 ft, but the next standard flange and flare is 1.96 ft in diameter. The maximum entrance velocity is 5.03 ft/s. The trench width is $2\,D = 3.92$ ft and the depth is $2\,D +$ floor clearance $= 4.90$ ft. Submergence required (Equation 12-1) is 4.8 ft. The top of the trench was set 0.95 ft below the invert of the sewer so that at neither peak inflow rate nor at half the peak inflow rate (at a lower elevation, of course) would the average velocity in the wet well above the trench and around the motors exceed 1.0 ft/s. The actual submergence is about 5.2 ft at LWL and 7.5 ft at HWL.

The pumps can be centered in the trench by recessing the discharge elbow (see Figure 29-12b). The pumps chosen are furnished with short suction nozzles. The floor is lowered below the last pump to ensure strong currents along the trench and minimum air ingestion during cleaning, so its nozzle is longer (see Figure 29-12c). The pumps are set on 4.66-ft centers as required by the manufacturer. Closer spacing might be tolerated pending model studies. The plans, excluding the valve vault and a small building for the AF converters, are shown in Figure 29-12.

Design Option 2 (Four C/S Pumps)

The parallel pump array consists of three C/S,175-hp Flygt CP3312 pumps and a C/S, 75-hp Flygt CP3300 pump selected for flow rates below 3600 gal/min. Compared with V/S pumping,

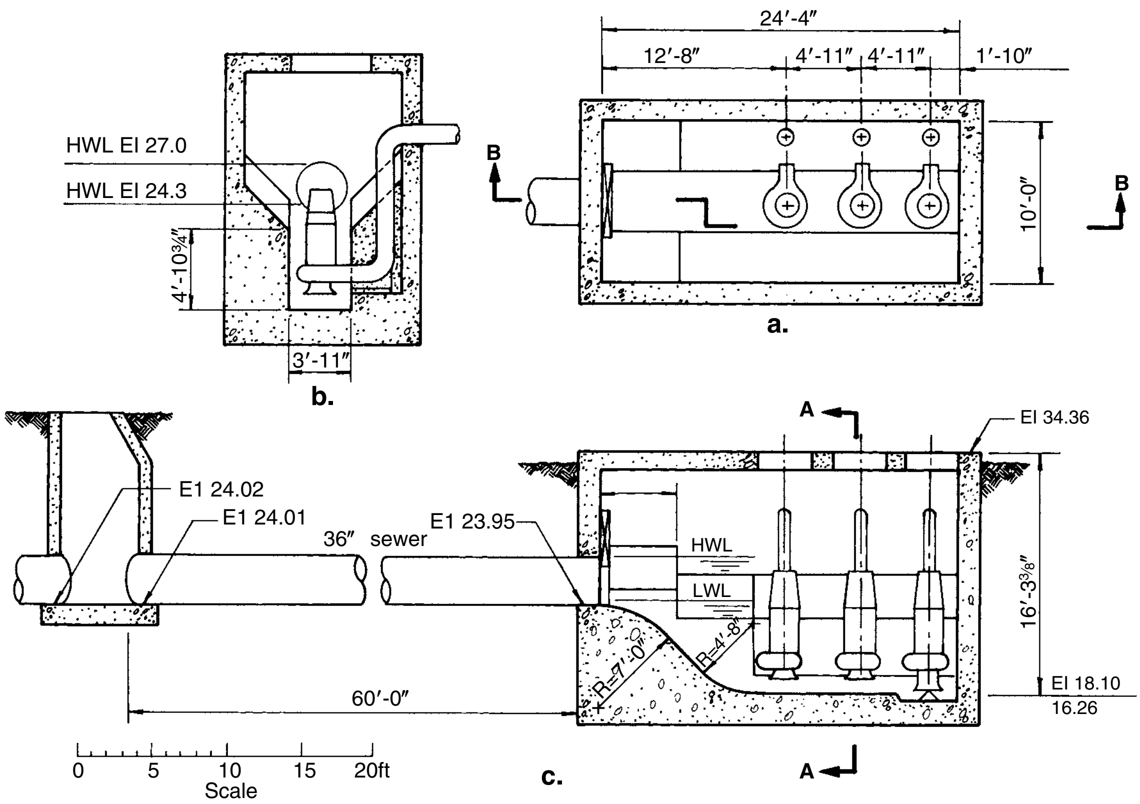


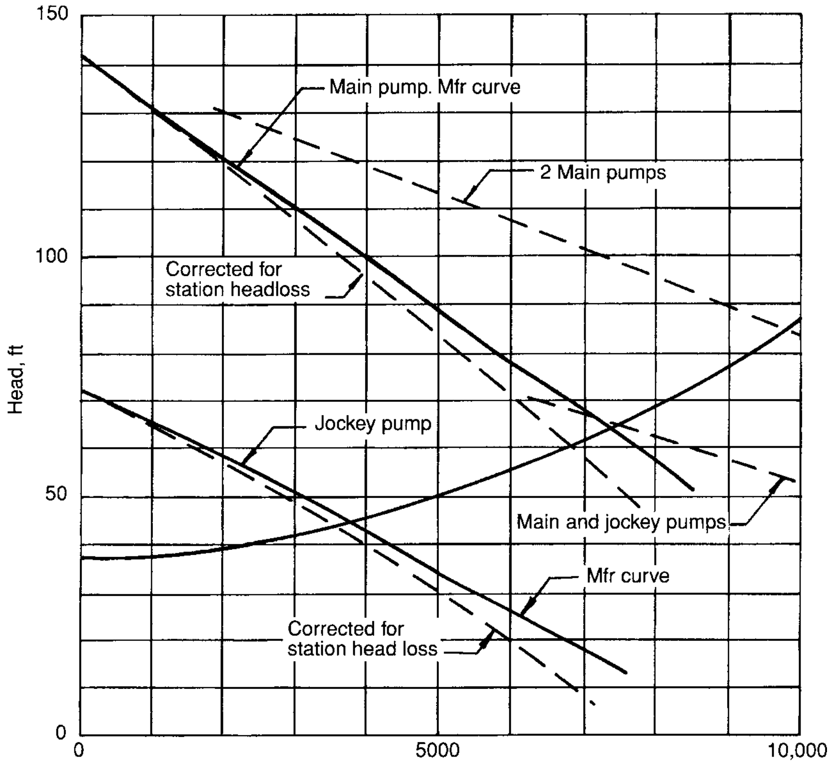
Figure 29-12. Plans for a pumping station with three V/S pumps. (a) Plan at elev. = 28.0 ft; (b) Section A-A; (c) Section B-B.

the advantages of C/S pumping include lower equipment cost, reduced maintenance, simplicity, and somewhat greater reliability, whereas the disadvantages include a deeper wet well with an approach pipe to augment the storage volume so as to allow pumps to cycle on and off at intervals long enough to avoid overloading motors and motor starters. The surges accompanying essentially instantaneous pump starts may be a significant disadvantage. The characteristic pump curves and an average system curve are shown in Figure 29-13.

In C/S pumping stations, the advantage of a four-pump facility that includes one relatively small pump is greater flexibility and the improvement of efficiency at lower flow rates because of the reduced friction losses. The disadvantages are the somewhat longer wet well and valve vault needed for the extra pump and its discharge piping, and an increase in mechanical equipment, maintenance, and repair costs.

To keep excavation cost to the absolute minimum, the wet well elevation was set as high as possible by making the approach pipe as short as possible and allowing the HWL for any pump or pump combination to reach the nominal water level in the upstream sewer. At the transition manhole (Figure 29-14), the crown of the approach pipe was set at the same elevation as that of the sewer to ensure supercritical velocities at both intermediate and maximum flow rates.

Little is known about headloss coefficients for flows in transition manholes where the shape of the water cross-section changes from nearly a full circle to a shallow segment of a larger circle. A K-factor of 0.1 applied to the exit velocity seems to be a reasonable assumption from which the drop in the energy grade line (EGL) is $0.1(v_2)^2/2g = 0.25$ ft, as shown in Figure 29-14.



LIVE GRAPH
Click here to view

Figure 29-13. Pump and system curves for station with constant-speed pumps.

Fortunately, an error in calculating headloss is of little consequence because velocity varies only as the square root of the head (water surface to EGL) and the increase in sequent depth (after the hydraulic jump) is only about one-third as much as the percentage change in velocity. Hence, at the maximum flow rate, the exit velocity could be 21% too high with a sequent depth of, therefore, 67%—both acceptable values.

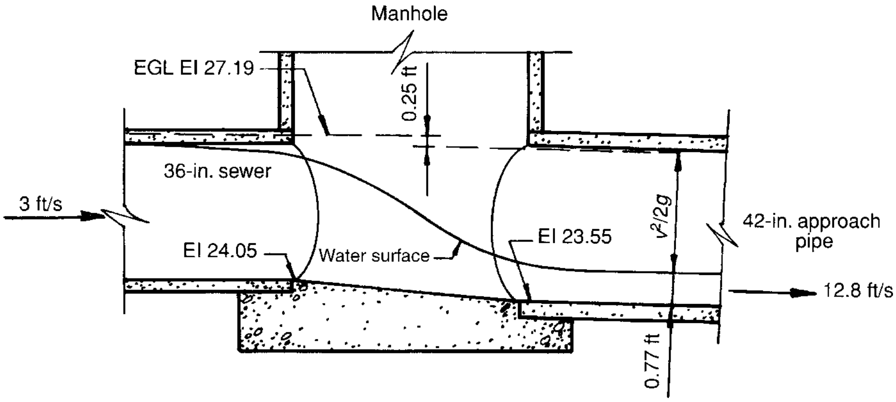


Figure 29-14. Transition manhole for constant-speed pipes.

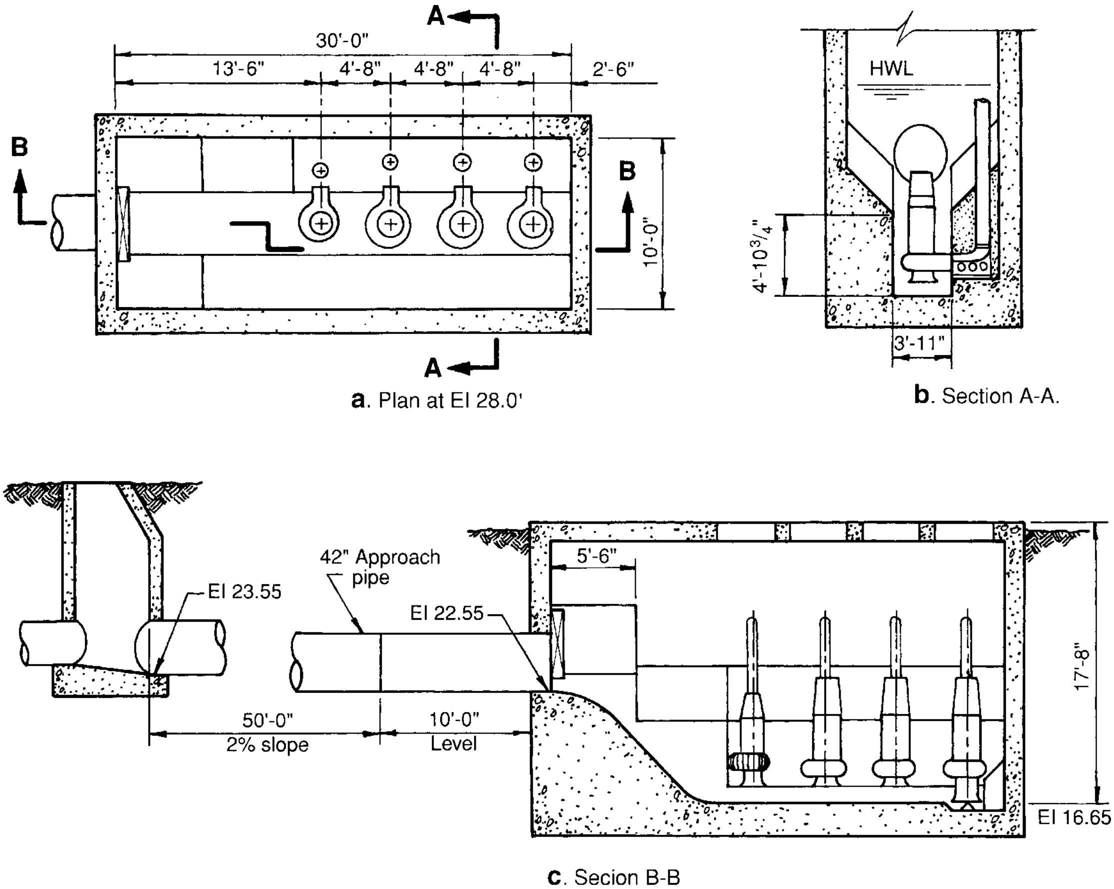


Figure 29-15. Plans for pumping station with four constant-speed pumps. (a) Plan at Elev. = 28.0 ft; (b) Section A-A; (c) Section B-B.

The design closely follows that of the foregoing V/S station except for the need for fluctuating water levels as pumps cycle on and off. An approach pipe is needed to prevent a free fall of water and to contribute to the active storage volume. The cross-section (Figure 29-15b) is the same as shown in Figure 29-12b, but the wet well was lengthened to accommodate an extra small pump, chosen to discharge a flow rate of 3500 gal/min. The complete plans are shown in Figure 29-15.

The volume of the wet well was found in preliminary calculations to be almost sufficient, so the approach pipe was made only 60 ft long with its last 10 ft level. A level exit for the approach pipe seems desirable although model studies do not show a discernable advantage over a 2% slope. To obtain a graph of pump-cycling frequency (similar to Figure 12-30), a plot of storage volume versus elevations above the approach pipe invert was made by (1) dividing the wet well and approach pipe into several simpler geometric shapes, (2) plotting the volume of each shape versus depth referenced to the influent pipe invert, and (3) adding the volumes graphically. (The volume occupied by the inflow moving at supercritical velocity in the approach pipe was not included.) These data were transferred to Figure 29-16.

The entrance fluid velocity into the wet well at the end of every pump cycle should, ideally, be more than 3 ft/s (to scour the approach pipe) but less than 4 ft/s (to avoid strong currents). With fixed stop-level controls for each pump (or combination thereof), some compromise may be necessary. For the small pump, the stop level was set at 0.45 ft above the invert for

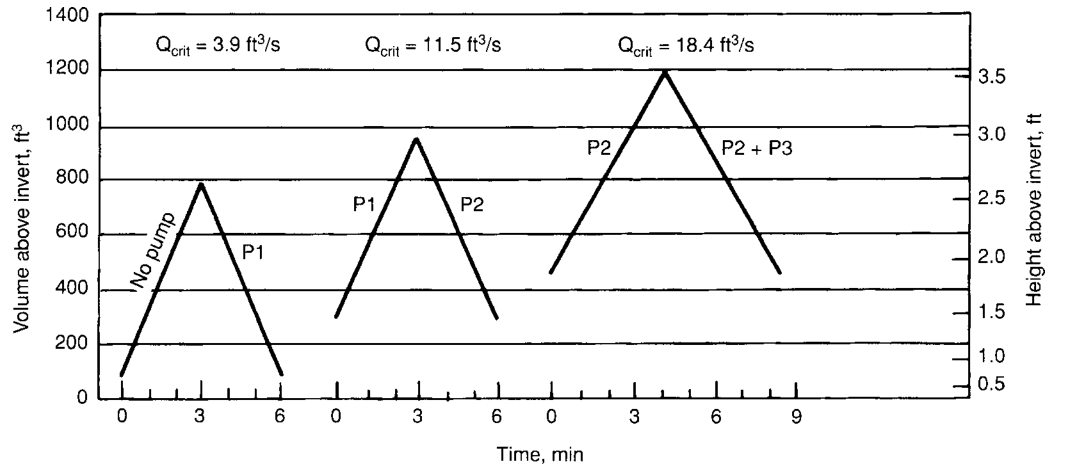


Figure 29-16. Wet well plus approach pipe volume and pump-cycle times. Small pump (7.80 ft^3/s) is P1, main pumps (15.1 ft^3/s each) are P2, P3, or P4.

end-of-cycle entrance fluid velocities varying from 2 to 4.6 ft/s. For 97% of the time, however, the minimum velocity is 3 ft/s. The stop level for P2 was set at 1.47 ft to produce velocities of 2.1 to 4.0 ft/s, and the stop level for P2 + P3 was set at 1.86 ft to produce fluid velocities of 3 to 4 ft/s (see Figure 29-16).

Start levels were set to yield a minimum cycle time of 6 min and for minimum successive start levels differing by at least 0.5 ft. The corresponding volumes can be found by Equation 12-3 ($V = Tq/4$) or by plotting the first leg of the cycle time at a slope of $Q_{critical}$ to an intersection with half the cycle time (note the first and third legs in Figure 29-16). $Q_{critical}$ is half of the pump capacity or the average capacity of P1 + P2 or P2 + P3, etc.

The start and stop levels were not allowed to interfere with the fluid velocities in the upstream sewer.

Design Option 3 (Three C/S Pumps)

Three pumps constitute the simplest type of installation—typical of many, especially those of small and medium sizes. The principal attraction is the perceived low first cost, but the drawbacks include higher energy consumption and less operational flexibility than either a four-pump C/S station or a V/S station.

The pump and station curves are similar to those of Figure 29-15 except that there is no small pump. Omitting the small pump would allow the wet well to be shortened. But, as the active storage volume required was no less, the needed volume could be obtained with either (1) a longer approach pipe and a shorter, deeper wet well, or (2) by retaining the same dimensions. The latter option was chosen, and the plans are the same as those of Figure 29-15 except that there are three pumps instead of four.

A diagram of the pump-cycling frequency is shown in Figure 29-17. The start and stop elevations had to be changed to accommodate the omission of the small pump, but the greater volume required for the larger starting pump could be reduced by alternating the pumps in successive cycles—a common expedient that cuts the required cycle time in half.

One unresolved difficulty is that the use of fixed stop levels produces end-of-cycle entrance velocities that, for P1, would vary from 4.0 to less than 1 ft/s. To avoid the buildup of deposits not expelled at velocities much less than about 2.5 ft/s, a PLC could be programmed to lower the stop level for low flows. Inflow can be quantified approximately by programming the computer to measure the time period for the up leg of the pump cycle.

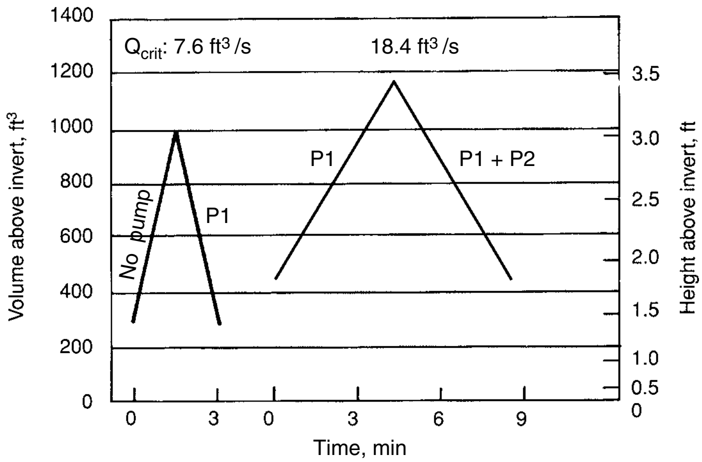


Figure 29-17. Wet wells plus approach pipe volumes and pump cycle times for two main duty pumps (15.1 ft^3/s each) alternated.

Construction Costs

Construction costs estimated for the three design options above are based on the wage rates prevailing in 1997 at an ENR 20-cities average of 5798 for the Atlanta, Georgia, area. For a graphic presentation of the effects that specific ground and soils conditions can have on the costs of a particular pumping station, two estimates were made for each design option: (1) one for good soil conditions, and (2) one for poor soils with high groundwater. The difference in costs is due to the need for watertight sheet piling and dewatering in poor soils.

A single cost estimate, even at this budget stage, requires seven pages of fine print and is thus too lengthy to include herein, but a typical summary is presented in Table 29-1. Grand totals for all six cost estimates are given in Table 29-2.

It is of considerable interest to see how each option affects the costs attributable only to the pumps and unaffected by all of the rest of the pumping station costs. These costs, shown in Table 29-3, are included in the total construction costs of Table 29-2.

The costs for installation, piping, vaults, conduit, wiring, contractor indirects, overhead, profit, bonding, and insurance must be added to the costs in Table 29-2. These additional costs are significant. For example, the fob cost of the small pump, \$31,000, increases by a factor of 4.5 to \$140,000 when the foregoing components of costs are included.

Energy Usage, Design Option 1 (Three V/S Pumps)

Adjustable-frequency drives cause a loss of efficiency as the frequency and speed decrease. A plot of the representative combined efficiency of the AF converter and the induction motor are shown in Figure 29-18. The graph represents a modern pulse-width modulated (PWM) type of adjustable-frequency drive. These drives feature a linear volts-to-hertz ratio and typically achieve high efficiencies at full speed. The efficiency curve in the figure differs from that commonly published as AFD efficiency, because it accounts for (1) the lower motor efficiency resulting from a non-sinusoidal wave form emitted from the AF converter, and (2) the usual motor loadings at lower speeds. Published data, on the other hand, are based on the assumption that power is proportional to speed cubed—a good assumption for fans and blowers but quite erroneous for pumping stations where static head is a substantial portion of the total design head. The curve of Figure 29-18 is based on actual testing of pump motors driven by PWM-type AFDs. The losses are manifested in the form of additional heat in the motor and in the AFD.

Table 29-1. Estimate Summary Sheet for a Pumping Station in Poor Soil. Three V/S Pumps. ENRCCI=5798. Courtesy of Brown and Caldwell, Construction Management Division

ESTIMATE SUMMARY SHEET

Project: Pumping station cost, poor soil 3 pump V/S system	Estimate : 1111C1 Job: Figure 29-12 Est date: 6-17-97
Made by: Pritchard/Abelin	PRT date: 10-11-97
Checked by: _____	PRT time: 8:49 pm

Summary totals		Markup data	
Labor: $ 105,933	Sales Tax: $ 14,449	Labor M/U: 15.00%	Sales: Tax (Maff): 6.00%
Material: $ 240,832	Markup: $ 66,730	Material M/U: 12.00%	Sales Tax (Equip): .00%
Subs: $ 386,253	Subtotal: $ 836,082	Subs M/U: 5.00%	Bond Rate: 1.00%
Equipment: $ 21,885	Bond: $ 8,361	Equipment M/U: 12.00%	Contingence :15.00%
	Revised Subtotal: $ 844,443		
	Contingence: $ 126,666		
	Grand Total: $ 971,109		

GRP #	Process Area	Labor M/C[a], Hrs	Labor Amount, $	Material Amount, $	Sub Amount, $	Equipment Amount, $	Sales Tax, $	Markup, $	Total, $	Job%
010	Contractor Indirects	1,384	65,394	608	2,450	14,490	36	11,744	94,722	11.3
015	Yard Development	192	3,955		6,570	480		981	11,986	1.4
018	Sewer Pipe	84	2,494	7,040	5,645	1,230	422	1,648	18,479	2.2
020	Pump Station	1,042	25,125	215,997	111,588	4,616	12,962	35,822	406,110	48.6
022	Valve Vault	354	8,965	17,187		1,069	1,029	3,535	31,785	3.8
176	Elect/instr				260,000			13,000	273,000	32.7
	Estimate Subtotal:	3,052	$ 105,933	$ 240,832	$ 386,253	$ 21,885	$ 14,449	$ 66,730	$ 836,082	100.0
	Plus Bond, if Required:								$ 8,361	1.0
	Revised Subtotal:								$ 844,443	101.0
	Contingence:								$ 126,666	15.0
	GRAND TOTAL:								$ 971,109	

[a]Man-h/Crew-h.

Table 29-2. Grand Total Capital Costs

Option	Grand total capital cost, $ Good soil	Poor soil
1 (3 V/S pumps)	859,910	971,109
2 (4 C/S pumps)	920,160	1,054,875
3 (3 C/S pumps)	782,184	914,209

Table 29-3. Capital Costs (fob) of Pumps, Motors, Guide Rails, and Controls

Option	Pump and accessory	Control cost, $	Total $
1 (3 V/S pumps)	165,000	160,999	325,000
2 (4 C/S pumps)	191,000	55,000	246,000
3 (3 C/S pumps)	165,000	50,000	215,000

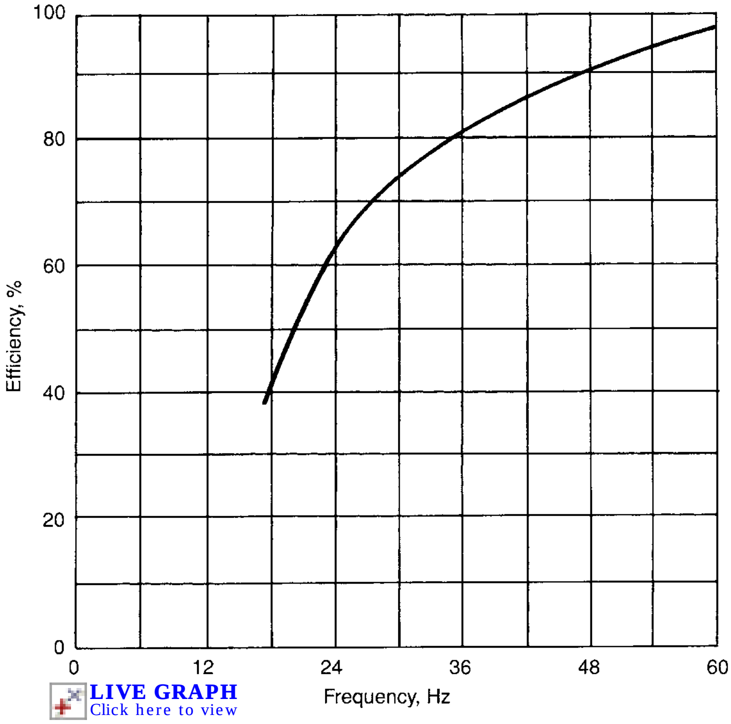


Figure 29-18. Representative efficiency of an AFD/motor system.

Specific energy (E_s) is a useful concept for comparing energy usage for various selections of pumping equipment. The term is a measure of the energy used per unit volume pumped, and in this example is expressed as kilowatt hours per million gallons pumped (kW · h/Mgal). The calculations used to obtain E_s are shown in Table 29-4.

The total energy use per year is 462,000 kW · h. As the inflow increases from a minimum, a single V/S pump is operated until its flow rate capacity is exceeded. Then two scenarios are possible:

- The first pump is ramped down as the second pump is ramped up until both are running at equal speeds. "Load-sharing operation" is shown as a heavy solid line in Figure 29-19 (see also Figure 15-8).
- The first pump continues to run at top speed while the second discharges the excess inflow, called "staggered operation" and shown as a dashed line in Figure 29-19. See "Conclusions and Critique" and also see "Comparison of Operating Modes" in Section 15-4.

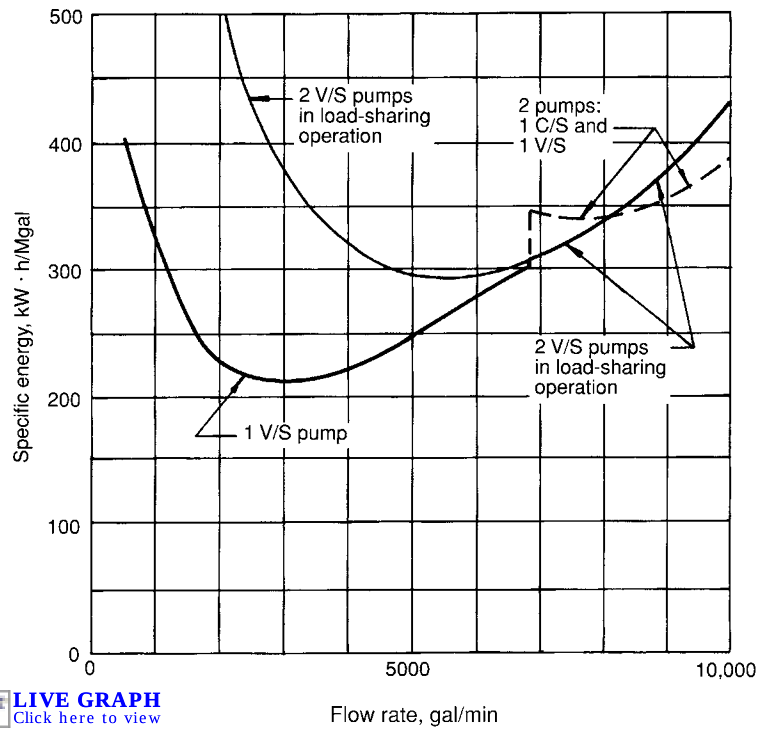
Energy Usage, Design Option 2 (Three Large C/S Pumps + One C/S Small Pump)

Only the small pump operates until the flow exceeds 3500 gal/min at Point A in Figure 29-20. The E_s is constant in this range, because the small pump, operating on and off, always pumps against nearly the same head. For flow rates from 3500 to 6800 gal/min (Points A to B), a large pump alternates with the small pump. Because the E_s of the small pump alone is only 210, the average E_s in this operating range (between Points A and B) becomes a slightly curved line.

At Point B, the small pump is no longer useful and only the large pump with its E_s of 300 is required. Above a flow rate of 6800 gal/min, one large pump operates continuously while the other cycles on and off. Again a curved line is developed between Points B and C. Instead of cycling one pump on and off, the two can be alternated to double the time for a cycle. The data for calculating E_s for Figure 29-20 are given in Table 29-5.

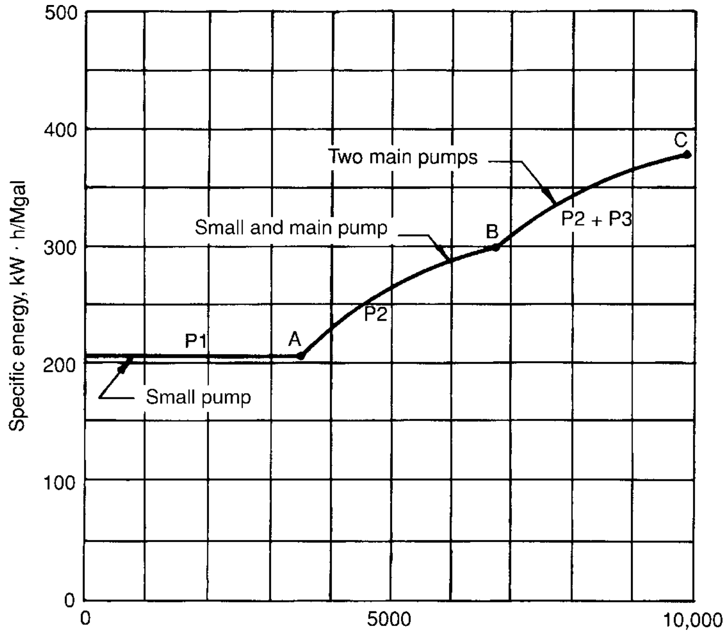
Table 29-4. Computation of Specific Energy for One V/S Pump and (in Staggered Operation) Two V/S Pumps

CP3312 pumps operating at:		Total head, ft		Flow rate, gal/min			Total pump	Specific energy,
V/S	C/S	Pump 1	Pump 2	Pump 1	Pump 2	Station	kW	kW · h/Mgal
1	0	37.8	0	1000	0	1000	20.1	335
1	0	38.6	0	1500	0	1500	23.5	261
1	0	39.8	0	2000	0	2000	27.2	227
1	0	41.4	0	2500	0	2500	32.3	215
1	0	43.2	0	3000	0	3000	38.2	212
1	0	45.5	0	3500	0	3500	45.6	217
1	0	48.5	0	4000	0	4000	53.4	223
1	0	51.0	0	4500	0	4500	62.9	233
1	0	54.3	0	5000	0	5000	73.5	245
1	0	57.9	0	5500	0	5500	85.1	258
1	0	61.8	0	6000	0	6000	98.6	274
1	0	66.2	0	6500	0	6500	113.1	290
1	1	69.7	61.6	6594	406	7000	144.9	345
1	1	72.7	65.6	6274	1256	7500	153.5	341
1	1	75.8	70.0	5942	2058	8000	163.7	341
1	1	79.1	74.9	5576	2924	8500	177.5	348
1	1	82.5	80.3	5162	3837	8999	193.3	358
1	1	86.6	86.6	4750	4750	9500	212.8	373



LIVE GRAPH
Click here to view

Figure 29-19. Specific energy for two V/S pumps versus one V/S and one C/S pump.



LIVE GRAPH
Click here to view

Figure 29-20. Specific energy for a small and two main constant-speed pumps.

Table 29-5. Computation of Specific Energy for One Small and Two Large C/S Pumps

Pumps operating	Head, ft			Flow rate, ad/min		Total, kW	Specific energy, kW · h/Mol
	Piping	Main	Total	Per pump	Station		
1-CP3300	2.6	44.4	47.0	3600	3600	48.6	225
1-CP3312	8.6	61.4	70.0	6786	6786	122.0	300
2-CP3312	4.5	85.4	89.9	4930	9860	221.7	376

Energy Usage, Design Option 3 (Three C/S Pumps)

The E_s curve between Points A and B in Figure 29-21 is exactly the same as in Figure 29-20. However, at flow rates less than 6800 gal/min (Point A), only a 175-hp pump is available for operation and the E_s is constant at 300. Consequently, the overall specific energy required is higher, and pumping is, on the average, less efficient than with three large pumps and one small pump.

Annual Cost Analysis

The total annual costs for energy and maintenance (including repairs and service) are given in Table 29-6. Maintenance is taken as 5% of the capital cost of pumps, accessories, and control given in Table 29-5. The 5% is an estimated value that is not completely realistic. Objective figures, however, are not only difficult but probably impossible to obtain. There are many circumstances where identical pumps in identical circumstances have a history of decidedly different repair costs for no discernible reason. However, experience shows that care

in the selection of equipment and the use of custom engineered pumps with heavier shafts and bearings (at 15 to 20% cost premium) can extend the mean time between repairs by factors of 10 or more. This type of improvement in life cycle costs, though difficult to quantify, is at the heart of the engineer's task of minimizing the total cost of ownership. For example, the bearing replacement interval in four off-the-shelf 75 hp horizontal, end-suction sewage pumps increased from three or four months to four years after the pumps were replaced with custom engineered units. The reduced maintenance was well worth the cost of the new equipment. Had the original pumps been custom engineered, the owner's overall cost of maintenance and repairs would have been very low.

Total Life-Cycle Costs

The total life-cycle costs, given in Table 29-7, differ very little so that on the basis of cost alone, a logical choice cannot be made.

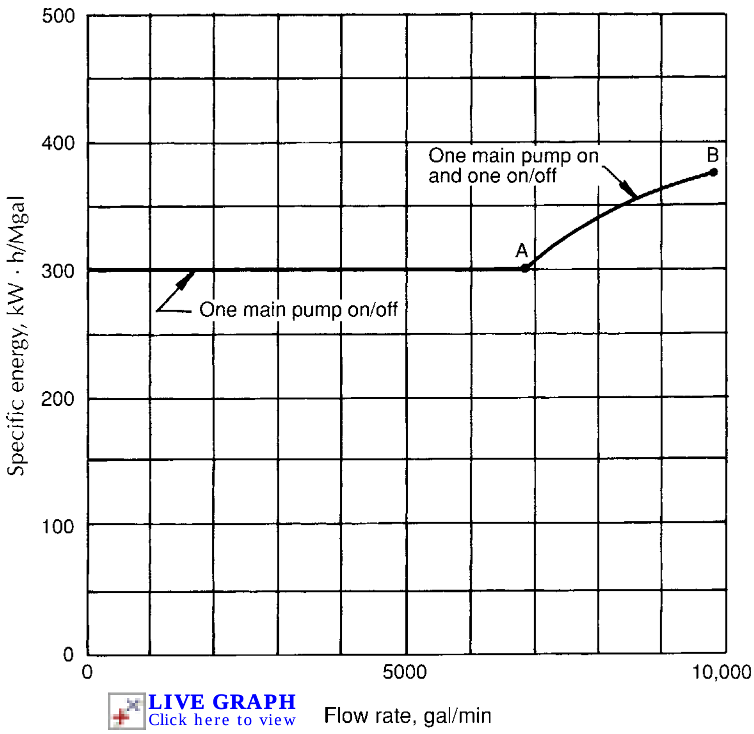


Figure 29-21. Specific energy for two main C/S pumps.

Table 29-6. Annual Energy Consumption and Cost of Operations

	Energy			
Option	Use, kW · h	Cost, $	Maintenance and service, $	Annual operation cost, $
1 (3V/S pumps)	498,041	28,882	16,250	46,132
2 (4 C/S pumps)	519,986	31,199	12,300	43,499
3 (3 C/S pumps)	635,086	38,105	10,750	48,855

Table 29-7 Total Life-Cycle Costs Based on Present Worth (PW) for 25 Years

Option	Operation, $ (Present worth)	Capital, $	Total life-cycle, $[a] (Present worth)	Comparison, %
		Good soil		
1 (3 V/S pumps)	871,210	859,910	1,730,000	102
2 (4 C/S pumps)	821,481	920,160	1,740,000	102
3 (3 C/S pumps)	922,629	782,184	1,700,000	100
		Poor soil		
1 (3 V/S pumps)	871,210	971,109	1,840,000	100
2 (4 C/S pumps)	821,481	1,054,875	1,880,000	102
3 (3 C/S pumps)	922,629	914,209	1,840,000	100

[a]Rounded to three significant figures.

Conclusions and Critique

The cost differences for the three options are less than the errors in cost estimating. There is, in other words, no practical difference at all. Many engineers have assumed that a V/S pumping station is more costly than a C/S station because of the added cost of the AF converters. Such short-sighted reasoning ignores the comparative sizes and depths of wet wells and cost of electricity. Furthermore, none of the costs herein include the added cost of treatment operations because of the added burden of meeting effluent standards due to the effect of hydraulic shock loads generated by C/S pumps going on and off. If the entire life-cycle system costs were evaluated carefully, it seems likely that V/S pumping (at least for similar pumping stations) is actually less expensive than C/S pumping in the long run.

Treatment systems simply work better when changes in hydraulic loading vary gradually instead of abruptly. For example, in 1986, the chlorination system in the Lompoc (California) Wastewater Treatment Plant could not meet the required discharge standards. An examination of the plant revealed that the operators did not like (or, perhaps, did not understand) the V/S drives and had switched off the AF converters to make the pumps operate as C/S units. Returning the station to V/S pumping solved the chlorination problem.

Owners should be advised to avoid consideration of first cost only. If capital cost were the only criterion, Options 1, 2, and 3 would, in the same order, compare as 110, 118, and 100 in good soil and 106, 115, and 100 in poor soil.

Example 29-1 is representative of a rather typical pumping station, but the results should be viewed with objective skepticism when applied to markedly different circumstances. In general, however, small differences in dimensions or elevations would have a negligible effect on cost comparisons.

Location

Great care should be taken to ensure that comparative cost estimates are made with the very best available information. The ENRCCI is a reliable method for comparative cost estimating using historical data. Note that the region of the country has a profound effect on the costs of a pumping station. The costs in Example 29-1 would be 75% of those given if the station were to be built in New Orleans and 113% if built in San Francisco.

Matching Pump and System Curves

It is important in V/S pumping to select the highest flow-based duty point well to the right of the BEP (but within the POR) so that, as the flow decreases, the operating point does not stray too far from the high-efficiency zones (see also Section 15-3). By following the above guideline, the hydraulic efficiencies can be kept at or above 80% for flows from 3000 to 7000 gal/min. Note as well that the high-efficiency zone coincides with the zone of the most stable pump operation. If the guideline is followed, the pump is less likely to suffer cavitation, vibration, and radial thrust damage. It is also important to analyze the movement of duty points when pumps are turned on or off, or when their speed is varied. Two significant considerations are (1) ensuring operation within the recommended operational range of the pump (i.e., avoid approaching shut-off head or runout); and (2) pumping economy. With respect to

pumping economy, the flow duration curve should be carefully considered, and the duty points with the best efficiency should be chosen to coincide with flows of the longest duration. When duty points for contitions representing the most frequent operation have been selected close to the BEP, the pump will run smoothly and give the best possible service life.

Staggered Versus Load-Sharing Pump Operation

Staggered operation seems to be widely used—usually to save the cost of one or more AF converters by providing a mix of V/S and C/S drives—and is a common practice in Europe. To prevent operation at excessively low discharges, the station is designed to operate in C/S mode until inflow increases to the minimum allowable pumping capacity of one V/S unit. Consider the cost data, the statements above, and Sections 15-1 and 15-5. You should form your own conclusions about the advisability of a mix of V/S and C/S pumps after considering the cost data, the statements above, and Section 15-5.

Note that staggered operation is somewhat less efficient than parallel operation at flows slightly greater than the capacity of a single pump, as shown in Figure 29-19. Above 8300 gal/min in this example, however, staggered operation is more efficient, and the pump controller could be programmed to switch from load-sharing to staggered operation. But flows above 8000 gal/min occur for such a negligible period of time (see Figure 29-10) that switching to staggered operation is excessively fussy. Note, however, that a small energy savings could be obtained if flows between 7000 and 8000 gal/min were pumped with two V/S units in load-sharing operation instead of staggered operation with one V/S and one C/S unit.

A V/S pump should not be operated at (usually) less than 35% of its maximum flow rate. Consequently, in staggered operation as the influent flow rate begins to exceed the capacity of a single pump, the second pump must operate in on-off mode at 35% capacity. Such an operation defeats the reason for choosing V/S in the first place. On the other hand, in parallel operation, when the flow rate exceeds the capacity of a single pump, a second pump is ramped up while the first pump is ramped down so that each operates at slightly more than 50% capacity.

29-6. References

1. *ENR,* published weekly by McGraw-Hill, New York. (Available from Construction Economics, Engineering News-Record.) www.enr.com.
2. Whitman, Requardt & Associates, Engineers–Consultants, Baltimore, MD.
3. U.S. Environmental Protection Agency, Office of Municipal Pollution Control, Municipal Utilities Division, Washington, DC.
4. Sloan, L., Chief Engineer, ENPO Pump Co. Private communication (April 1987).
5. Patterson, W. L., and R. F. Banker, *Estimating Costs and Manpower Requirements for Conventional Wastewater Treatment Facilities*, U.S. EPA, 17080 DAN, Washington, DC (1971).
6. Taylor, G. A., *Management and Engineering Economy, Economic Decision-Making,* 3rd ed., Brooks/Cole Engineering Division, Monterey, CA (1980).
7. White, J. A., M. H. Agee, and K. E. Case, *Principles of Economic Analysis,* 2nd ed., John Wiley, New York (1984).
8. Dell'Isola, A. J., and S. J. Kirk, *Life-Cycle Costing for Design Professionals,* McGraw-Hill, New York (1981).
9. Wood, K. Private communication (January 3, 1986).
10. *Construction Grants 1985 (CG-85),* U.S. EPA 430/9-84-004, Office of Water Program Operations (WH-546), Washington, DC (July 1984).
11. Pritchard, M. T., D. S. Parker, and A. Malik, "Design-to-Cost with BACPAC." In-house publication, Brown and Caldwell Consultants, Pleasant Hill, CA.
12. Pritchard, M. T., D. S. Parker, and A. Malik, "Keeping costs from creeping," *Water Environment & Technology,* 4(12): 38–39 (December 1992).
13. Means, R. S., *Means Facilities Cost Data,* R. S. Means Co., Inc., Kingston, MA (updated annually).
14. Means, R. S., *Means Mechanical Cost Data.* R. S. Means Co., Inc., Kingston, MA (updated annually).
15. Dodge Cost Systems, *Vol. 1: Dodge Assemblies Cost Data; Vol. 2: Dodge Unit Cost Data; Vol. 3: Dodge Square Foot Cost Data; Vol. 4: Dodge Heavy Construction Cost Data, Vol. 5: Dodge Remodeling & Retrofit Cost Data,* McGraw-Hill, Princeton, NJ (updated annually).
16. Walski, T. M. (Ed.), *Proceedings of the Symposium on Cost Estimating for Water Supply Planning Studies,* Miscellaneous Paper EL-83-6, U.S. Army Corps of Engineers, Waterways Experiment Station, Vicksburg, MS (September 1983).
17. Water Research Centre, *Cost Information for Water Supply and Sewage Disposal,* TR 61, Water Research Centre, Medmenham, Marlow, Buckinghamshire, SL7 2HD, UK (1977).
18. Hall, R. S., J. Morley, and K. J. Naughton, "Current costs of process equipment," *Chemical Engineering,* 89: 80–116 (April 1982).

Appendix A
Physical Data

CONTRIBUTORS

Carl N. Anderson
W. Eric Hopkins
Constantine N. Papadakis
Daniel L. Shaffer
Earle C. Smith

In this volume, SI means System International (a metric system) and U.S. means U.S. customary units (an English system). The notation 6.22 E–2 means 6.22×10^{-2}. Most tabular entries are shown to only three significant digits because the errors in engineering data are rarely less than 1%. If greater accuracy is needed, use the conversion factors in Table A-3.

By convention, lb_f (pound force) is written as lb, but lb_m (pound mass) is always written lb_m.

Table A-1. Base SI Units

Quantity	Name	Symbol
Length	Meter	m
Mass	Kilogram	kg
Time	Second	s
Electric current	Ampere	A
Thermodynamic temperature	Kelvin	K
Amount of substance	Mole	mol
Luminous intensity	Candela	cd
Plane angle[a]	Radian	rad
Solid angle[a]	Steradian	sr

[a]Supplementary units.

Table A-2. Derived SI Units and Physical Quantities

Quantity	Symbol	Name	Formula
Acceleration, angular	—	Radian/second^2	rad/s^2
Acceleration, linear	—	Meter/second^2	m/s^2
Angle	rad	Radian	rad
Area	—	Meter^2	m^2
Capacitance	F	Farad	C/V
Charge	C	Coulomb	A · S
Concentration, molar	—	Molecular mass/meter^3	mol/m^3
Concentration, gravimetric	—	Milligram/liter or Gram/meter^3	mg/L g/m^3
Conductance	S	Siemen	A/V
Density, mass	—	Kilogram/meter^3	kg/m^3
Electric field intensity	—	Volt/meter	V/m
Electric flux	C	Coulomb	A · s
Electric flux density	—	Coulomb/square meter	C/m^2
Electric potential	V	Volt	V
Electric resistance	Ω	Ohm	V/A
Energy	J	Joule	N · m
Force	N	Newton	kg · m/s^2
Frequency	Hz	Hertz	cycle/s
Impedance	Ω	Ohm	V/A
Inductance	H	Henry	Wb/A
Luminous flux	lm	Lumen	cd · sr
Magnetic field intensity	—	Ampere/meter	A/m
Magnetic flux	Wb	Weber	V · s
Magnetic flux density	T	Tesla	Wb/m^2
Moment of force	—	Newton-meter	N · m
Power	W	Watt	J/s
Power, apparent	—	Volt-ampere	V · A
Power, reactive	—	Volt-ampere reactive	var
Pressure	Pa	Pascal	N/m^2
Reactance	Ω	Ohm	V/A
Resistance	Ω	Ohm	V/A
Resistivity	—	Ohm-meter	W · m
Stress	—	Newton/meter^2	N/m^2
Velocity, angular	—	Radian/second	rad/s
Velocity, linear	—	Meter/second	m/s
Viscosity, dynamic	—	Pascal · second	Pa · s
Viscosity, kinematic	—	Meter^2/second	m^2/s
Voltage	V	Volt	W/A
Volume	—	Meter^3	m^3
Volume, specific	—	Meter^3/kg	m^3/kg
Work	J	Joule	N · m

Table A-3. Physical Constants

Name	SI units	U.S. customary units
Acceleration due to gravity[a]	9.80665 m/s^2	32.174 ft/s^2
Atmosphere (standard)	101.325 kPa	14.696 $lb/in.^2$
Atmosphere (standard)	10.333 m of WC	33.899 ft of WC
Bar[b]	100.0 kPa	14.504 $lb/in.^2$
Mass density of water[b]	1000 kg/m^3	1.94 $slugs/ft^3$
Specific heat of air[c]	1003 $N \cdot m/(kg \cdot °C)$	0.2395 $Btu/lb_m \cdot °F =$ 6000 $ft \cdot lb/(slug \cdot °F)$

[a]Standard value, changes with latitude and elevation.
[b]At 5°C (41°F).
[c]At 20°C (68°F) and 1 atm.

Table A-4. SI Prefixes

Multiplication factor		Prefix[a]	Symbol
1,000,000,000,000	E+12	tera	T
1,000,000,000	E+9	giga	G
1,000,000	E+6	mega	M
1,000	E+3	kilo	k
100	E+2	hecto[b]	h
10	E+1	deka[b]	da
0.1	E−1	deci[b]	d
0.01	E−2	centi[b]	c
0.001	E−3	milli	m
0.000 001	E−6	micro	μ
0.000 000 001	E−9	nano	n
0.000 000 000 001	E−12	pico	p

[a]The first syllable of every prefix is accented so that the prefix retains its identity. Therefore, the preferred pronunciation of kilometer places the accent on the first syllable, *ki*-lometer.
[b]The use of these prefixes should be avoided, except for the measurement of areas and volumes and for the nontechnical use of centimeter.

Table A-5. Conversion Factors, U.S. Customary Units to Commonly Used SI Units

Multiply the U.S. customary unit		by	To obtain the SI unit	
Name	Symbol		Symbol	Name
Acceleration feet per second squared[a]	ft/s^2	3.0480 E−1[a]	m/s^2	Meters per second squared[a]
Angle	degrees (°)	1.745 E−2	rad	Radian
Area				
Acre	acre	4.047 E−1	ha	Hectare
Square foot	ft^2	9.2903 E−2	m^2	Square meter
Square inch	$in.^2$	6.4516 E−4[a]	m^2	Square meter
Concentration (for gases), parts per million	ppm	[b]	mg/m^3	Milligrams per cubic meter
Energy				
British thermal unit	Btu	1.0551 E+0	kJ	Kilojoule
Foot-pound (force)	$ft \cdot lb$	1.3558 E+0	J	Joule
Horsepower-hour	$hp \cdot h$	2.6845 E+0	MJ	Megajoule
Kilowatt-hour	$kW \cdot h$	3.600 E+3[a]	kJ	Kilojoule

[a]Exact value or exact conversion.

(*Table continues on next page*)

[b]$mg/m^3 = \frac{(ppm)(grams/mol)}{22.4\ L/mol} \times \frac{273}{273 + °C}$. Note: Regulations use °C = 25, but source measurements are referenced to °C = 21.1.

Table A-5. Continued

Multiply the U.S. customary unit		by	To obtain the SI unit	
Name	Symbol		Symbol	Name
Force				
Pound force	lb	4.4482 E+0	N	Newton
Flow rate				
Cubic feet per second	ft^3/s	2.8317 E−2	m^3/s	Cubic meters per second
Gallons per minute	gal/min	6.3090 E − 2	L/s	Liters per second
Gallons per minute	gal/min	2.2712 E − 1	m^3/h	Cubic meters per hour
Million gallons per day	Mgal/d	4.3813 E + 1	L/s	Liters per second
Million gallons per day	Mgal/d	3.7854 E + 0	ML/d	Million liters per day
Length				
Foot	ft	3.0480 E − 1[a]	m	Meter
Inch	in.	2.5400 E + 1[a]	mm	Millimeter
Mile	mi	1.6093 E + 0	km	Kilometer
Luminous intensity				
Foot-candle	ft · cd	1.076 E + 1	lx	Lux
Mass				
Pound	lb_m	4.5359 E − 1	kg	Kilogram
Slug (32.174 lb_m)	slug	1.4594 E + 1	kg	Kilogram
Moment				
Pound-inch	lb · in.	1.1298 E − 1	N · m	Newton-meter
Power				
Horsepower	hp	7.457 E − 1	kW	Kilowatt
Pressure (force/area)				
Atmosphere (standard)	atm	1.0133 E + 2	kPa	Kilopascal (kN/m^2)
Pounds (force) per square inch	$lb/in.^2$	6.8948 E + 0	kPa	Kilopascal
Specific heat	$Btu/(lb_m \cdot R)$	4.188 E + 0	$J/(g \cdot K)$	Joules per gram-degrees Kelvin
	$ft \cdot lb/(slug \cdot R)$	1.672 E − 1	$N \cdot m(kg \cdot K)$	Newton-meters per kilogram-degrees Kelvin
Stress				
Pounds (force) per square inch	$lb/in.^2$	6.8948 E + 0	kPa	Kilopascal
Kips (force) per square inch	$k/in.^2$	6.8948 E + 0	MPa	Megapascal
Temperature				
Degrees Fahrenheit	°F	0.5556(°F − 32)	°C	Degrees Celsius
Degrees Fahrenheit	°F	0.5556(°F + 459.67)	K	Degrees Kelvin
Degrees Rankine	R	0.5556 (R)	K	Degrees Kelvin
Velocity				
Feet per second	ft/s	3.048 E − 1[a]	m/s	Meters per second
Miles per hour	mi/h	1.6093 E + 0	km/h	Kilometers per hour
Volume				
Cubic foot	ft^3	2.8317 E + 1	L	Liter
Cubic foot	ft^3	2.8317 E − 2	m^3	Cubic meter
Gallon	gal	3.7854 E − 3	m^3	Cubic meter
Gallon	gal	3.7854 E + 0	L	Liter
Weight, specific				
Pounds per cubic foot	lb/ft^3	1.5708 E − 1	kN/m^3	Kilonewtons per cubic meter

[a]Exact value or exact conversion.

[b]$mg/m^3 = \dfrac{(ppm)(grams/mol)}{22.4\,L/mol} \times \dfrac{273}{273 + °C}$. Note: Regulations use °C = 25, but source measurements are referenced to °C = 21.1.

Table A-6. Atmospheric Pressure (SI Units)

Elevation above sea level (m)	Atmospheric pressure[a] kPa	Water column, mm	Mercury column, mm	Specific weight, γ, of air at 20°C, kN/m^3[b]
0	101.3	10.33	760	1.18 E−2
500	95.6	9.74	717	1.11 E−2
1000	90.1	9.19	676	1.05 E−2
1500	84.8	8.64	636	9.87 E−3
2000	79.8	8.13	598	9.29 E−3
2500	73.3	7.47	550	8.53 E−3
3000	70.3	7.17	527	8.19 E−3
3500	66.1	6.74	496	7.70 E−3

[a]Storms commonly reduce atmospheric pressure by about 1.7%.

[b]At other temperatures and pressures, use $p_1 v_1/K_1 = p_2 v_2/K_2$ where p is pressure, v is volume, and K is degrees Kelvin (°C + 273).

Table A-7. Atmospheric Pressure (U.S. Customary Units)

	Atmospheric pressure[a]				
			Mercury column		
Elevation above sea level, ft	$lb/in.^2$	Water column, ft	in.	mm	Specific weight, γ, of air at 68°F, lb/ft^3[b]
0	14.7	33.9	29.92	760	7.52 E−2
1,000	14.2	32.7	28.90	734	7.26 E−2
2,000	13.7	31.6	27.87	708	7.00 E−2
3,000	13.2	30.4	26.81	681	6.73 E−2
4,000	12.7	29.4	25.90	658	6.51 E−2
5,000	12.1	28.2	24.92	633	6.26 E−2
6,000	11.8	27.2	24.01	610	6.04 E−2
7,000	11.4	26.3	23.19	589	5.83 E−2
8,000	10.7	24.8	21.89	556	5.50 E−2
9,000	10.5	24.2	21.38	543	5.37 E−2
10,000	10.1	23.4	20.63	524	5.18 E−2

[a]Storms commonly reduce atmospheric pressure by about 1.7%.

[b]At other temperatures and pressures use $p_1 v_1/T_1 = p_2 v_2/T_2$, where $T = {}^\circ F + 460$, or use the general formula for atmospheric pressure

$$\frac{p_b}{p_a} = \exp\left[-\frac{gM(z_b - z_a)}{g_c RT}\right]$$

where $g_c = 32.2\,\text{ft}\cdot\text{lb}_m/\text{lb}\cdot\text{s}^2$, $g = 32.2\,\text{ft/s}^2$, $M = 29\,\text{lb}_m/\text{lb}-\text{mol}$, $R = \dfrac{1545\text{ft}\cdot\text{lb}}{\text{lb}_{mol}\cdot T}$, and $T = 460 + {}^\circ F$.

Table A-8. Physical Properties of Water (SI Units)[a]

Temperature, °C	Specific weight[b], γ, kN/m^3	Density, ρ, kg/m^3	Bulk modulus of elasticity[c], K, kPa	Dynamic viscosity, μ, Pa · s	Kinematic viscosity, ν, m^2/s	Surface tension[d], σ, N/m	Vapor pressure, p_v, kPa	Vapor pressure, p_v, kPa, *WC* m
0	9.805	9.998 E+2	1.98 E+6	1.78 E−3	1.79 E−6	0.0765	0.61	0.06
5	9.807	1.000 E+3	2.05 E+6	1.52 E−3	1.52 E−6	0.0749	0.87	0.09
10	9.804	9.997 E+2	2.10 E+6	1.31 E−3	1.31 E−6	0.0742	1.23	0.13
15	9.798	9.991 E+2	2.15 E+6	1.14 E−3	1.14 E+6	0.0735	1.70	0.17
20	9.789	9.982 E+2	2.17 E+6	1.00 E−3	1.00 E−6	0.0726	2.34	0.24
25	9.777	9.970 E+2	2.22 E+6	8.90 E−4	8.93 E−7	0.0720	3.17	0.32
30	9.764	9.957 E+2	2.25 E+6	7.98 E−4	8.00 E−7	0.0712	4.24	0.43
40	9.730	9.922 E+2	2.28 E+6	6.53 E−4	6.58 E−7	0.0696	7.38	0.76
50	9.689	9.880 E+2	2.29 E+6	5.47 E−4	5.53 E−7	0.0679	12.33	1.27
60	9.642	9.832 E+2	2.28 E+6	4.66 E−4	4.74 E−7	0.0662	19.92	2.07
70	9.589	9.778 E+2	2.25 E+6	4.04 E−4	4.13 E−7	0.0644	31.19	3.25
80	9.530	9.718 E+2	2.20 E+6	3.54 E−4	3.64 E−7	0.0626	47.34	4.97
90	9.466	9.653 E+2	2.14 E+6	3.15 E−4	3.26 E−7	0.0608	70.10	7.41
100	9.399	9.584 E+2	2.07 E+6	2.82 E−4	2.94 E−7	0.0589	101.33	10.78

[a]Adapted from Vennard and Street [1].

[b]Specific weight, γ, is force per unit volume. The relationship between γ, ρ, and the acceleration due to gravity, g, is $\gamma = \rho g$. See Table A-3 for the value of g.

[c]At atmospheric pressure.

[d]In contact with air.

Table A-9. Physical Properties of Water (U.S. Customary Units)[a]

Temperature, °F	Specific weight[a], γ, lb$_m$/ft^3	Density[b], ρ, slug/ft^3	Bulk modulus of elasticity[c], K, lb/in.^{3}	Dynamic viscosity, μ, lb × s/ft^3	Kinematic viscosity, ν, ft^2/s	Surface tension[d], σ, lb/ft	Vapor Pressure p_v, lb/in.^{2}	Vapor Pressure *WC*, ft
32	62.42	1.940	2.87 E+5	3.746 E−5	1.93 E−5	0.00518	0.09	0.2
40	62.43	1.940	2.96 E+5	3.229 E−5	1.66 E−5	0.00614	0.12	0.3
50	62.41	1.940	3.05 E+5	2.735 E−5	1.41 E−5	0.00509	0.18	0.4
60	62.37	1.938	3.13 E+5	2.359 E−5	1.22 E−5	0.00504	0.26	0.6
70	62.30	1.936	3.19 E+5	2.112 E−5	1.06 E−5	0.00498	0.36	0.8
80	62.22	1.934	3.24 E+5	1.799 E−5	9.30 E−6	0.00492	0.51	1.2
90	62.11	1.931	3.28 E+5	1.595 E−5	8.26 E−6	0.00486	0.70	1.6
100	62.00	1.927	3.31 E+5	1.424 E−5	7.39 E−6	0.00480	0.95	2.2
110	61.86	1.923	3.32 E+5	1.284 E−5	6.67 E−6	0.00473	1.27	2.9
120	61.71	1.918	3.32 E+5	1.168 E−5	6.09 E−6	0.00467	1.69	3.9
130	61.55	1.913	3.31 E+5	1.069 E−5	5.58 E−6	0.00460	2.22	5.2
140	61.38	1.908	3.30 E+5	0.981 E−5	5.14 E−6	0.00454	2.89	6.8
150	61.20	1.902	3.28 E+5	0.905 E−5	4.76 E−6	0.00447	3.72	8.8
160	61.00	1.896	3.26 E+5	0.838 E−5	4.42 E−6	0.00441	4.74	11.2
170	60.80	1.890	3.22 E+5	0.780 E−5	4.13 E−6	0.00434	5.99	14.2
180	60.58	1.883	3.18 E+5	0.726 E−5	3.85 E−6	0.00427	7.51	17.9
190	60.36	1.876	3.13 E+5	0.678 E−5	3.62 E−6	0.00420	9.34	22.3
200	60.12	1.868	3.08 E+5	0.637 E−5	3.41 E−6	0.00413	11.52	27.6
212	59.83	1.860	3.00 E+5	0.593 E−5	3.19 E−6	0.00404	14.70	35.4

[a]Specific weight, γ, is the weight per unit volume. The relationship between γ, ρ, and the acceleration due to gravity, g, is $\gamma = \rho g$. See Table A-3 for the value of g.

[b]At atmospheric pressure, 1 slug = 32.2 lb$_m$ × s^2.

[c]In many formulas, K must be in pounds per square foot. Multiply by 144 in.^{2}/ft^2.

[d]In contact with air.

Table A-10. Physical Properties of Pipe Materials

Material	Poisson's ratio, μ, dimensionless	Modulus of elasticity (E) SI units, N/m^2	Modulus of elasticity (E) U.S. customary units $lb/in.^2$	Modulus of elasticity (E) U.S. customary units lb/ft^2
Aluminum	0.33	7.30 E+10	1.05 E+7	1.51 E+9
Asbestos-cement	0.30	2.30 E+10	3.40 E+6	4.90 E+8
Brass	0.34	1.03 E+11	1.50 E+7	2.16 E+9
Copper	0.30	1.10 E+11	1.60 E+7	2.30 E+9
Ductile iron	0.28	1.66 E+11	2.40 E+7	3.46 E+9
Gray cast iron	0.28	1.03 E+11	1.50 E+7	2.16 E+9
HDPE[a]	0.45	1.03 E+9	1.50 E+5	2.16 E+7
PVC	0.45	2.70 E+9	4.00 E+5	5.76 E+7
Steel	0.30	2.07 E+11	3.00 E+7	4.32 E+9

[a]Varies.

Table A-11. Equivalent Weights and Measures

Length

U.S. customary units: Miles	Yards	Feet	Inches	SI units: Kilometers	Meters	Millimeters
1	1.76 E+3	5.28 E+3	6.36 E+4	1.61 E+0	1.61 E+3	1.61 E+6
5.68 E−4	1	3.00 E+0	3.60 E+1	9.14 E−4	9.14 E−1	9.14 E+2
1.89 E−4	3.33 E−1	1	1.20 E+1	3.05 E−4	3.05 E−1	3.05 E+2
1.58 E−5	2.78 E−2	8.33 E−2	1	2.54 E−5	2.54 E−2	2.54 E+1
6.21 E−1	1.09 E+3	3.28 E+3	3.94 E+4	1	1.00 E+3	1.00 E+6
6.21 E−4	1.09 E+0	3.28 E+0	3.94 E+1	1.00 E−3	1	1.00 E+3
6.21 E−7	1.09 E−3	3.28 E−3	3.94 E−2	1.00 E−6	1.00 E−3	1

Area

U.S. customary units: Square miles	Acres	Square feet	Square inches	SI units: Square kilometers	Hectares	Square meters
1	6.40 E+2	2.79 E+7	4.01 E+9	2.59 E+0	2.59 E+2	2.59 E+6
1.56 E−3	1	4.36 E+4	6.27 E+6	4.05 E−3	4.05 E−1	4.05 E+3
3.59 E−8	2.30 E−5	1	1.44 E+2	9.29 E−8	9.29 E−6	9.29 E−2
2.49 E−10	1.59 E−7	6.94 E−3	1	6.45 E−10	6.45 E−8	6.45 E−4
3.86 E−1	2.47 E+2	1.08 E+7	1.55 E+9	1	1.00 E+2	1.00 E+6
3.86 E−3	2.47 E+0	1.08 E+5	1.55 E+7	1.00 E−2	1	1.00 E+4
3.86 E−7	2.47 E−4	1.08 E+1	1.55 E+3	1.00 E−6	1.00 E−4	1

(*Table continues on next page*)

Table A-11. Continued

Volume

U.S. customary units					SI units		
Cubic yards	Cubic feet	Imperial gallons	U.S. gallons	Cubic inches	Cubic meters	Liters	Milliliters
1	2.7 E+1	1.68 E+2	2.02 E+2	4.67 E+4	7.65 E−1	7.65 E+2	7.65 E+5
3.70 E−2	1	6.23 E+0	7.48 E+0	1.73 E+3	2.83 E−2	2.83 E+1	2.83 E+4
5.95 E−3	1.61 E−1	1	1.20 E+0	2.77 E+2	4.55 E−3	4.55 E+0	4.55 E+3
4.95 E−3	1.34 E−1	8.33 E−1	1	2.31 E+2	3.79 E−3	3.79 E+0	3.79 E+3
2.14 E−5	5.79 E−4	3.60 E−3	4.33 E−3	1	1.64 E−5	1.64 E−2	1.64 E+1
1.31 E+0	3.53 E+1	2.20 E+2	2.64 E+2	6.10 E+4	1	1.00 E+3	1.00 E+6
1.31 E−3	3.53 E−2	2.20 E−1	2.64 E−1	6.10 E+1	1.00 E−3	1	1.00 E+3
1.31 E−6	3.53 E−5	2.20 E−4	2.64 E−4	6.10 E−2	1.00 E−6	1.00 E−3	1

Flow rate

U.S. customary units			SI units			
Million gallons per day	Cubic feet per second	Gallons per minute	Cubic meters per second	Cubic meters per hour	Liters per second	Millions of liters per day
1	1.55 E + 0	6.94 E + 2	4.38 E − 2	1.58 E + 2	4.38 E + 1	3.79 E + 0
6.46 E − 1	1	4.49 E + 2	2.83 E − 2	1.02 E + 2	2.83 E + 1	2.45 E + 0
1.44 E − 3	2.23 E − 3	1	6.31 E − 5	2.27 E − 1	6.31 E − 2	5.45 E − 3
2.28 E + 1	3.53 E + 1	1.58 E + 4	1	3.60 E + 3	1.00 E + 3	8.64 E + 1
6.34 E − 3	9.81 E − 3	4.40 E + 0	2.78 E − 4	1	2.78 E − 1	2.40 E − 2
2.28 E − 2	3.53 E − 2	1.59 E + 1	1.00 E − 3	3.60 E + 0	1	8.64 E − 2
2.64 E − 1	4.09 E − 1	1.83 E + 2	1.16 E − 2	4.17 E + 1	1.16 E + 1	1

Force

U.S. customary units				SI units	
Tons	Kilopounds	Pounds	Grains	Kilonewtons	Newtons
1	2.00 E + 0	2.00 E + 3	1.40 E + 7	8.90 E + 0	8.90 E + 3
5.00 E − 1	1	1.00 E + 3	7.00 E + 6	4.45 E + 0	4.45 E + 3
5.00 E − 4	1.00 E − 3	1	7.00 E + 3	4.45 E − 3	4.45 E + 0
7.14 E − 8	1.43 E − 7	1.43 E − 4	1	6.35 E − 7	6.35 E − 4
1.12 E − 1	2.25 E − 1	2.25 E + 2	1.57 E + 6	1	1.00 E + 3
1.12 E − 4	2.25 E − 4	2.25 E − 1	1.57 E + 3	1.00 E − 3	1

Mass

U.S. customary units		SI units		
Slugs	Pounds mass	Kilograms	Grams	Milligrams
1	3.22 E + 1	1.46 E + 1	1.46 E + 4	1.46 E + 7
3.11 E − 2	1	4.54 E − 1	4.54 E + 2	4.54 E + 5
6.85 E − 2	2.20 E + 0	1	1.00 E + 3	1.00 E + 6
6.85 E − 5	2.20 E − 3	1.00 E − 3	1	1.00 E + 3
6.85 E − 8	2.20 E − 6	1.00 E − 6	1.00 E − 3	1

	Pressure or stress (force/area)					
	U.S. customary units			SI units		
Standard atmospheres	Pounds per square inch	Hg, in.	Water, ft	Water, m	Kilopascals	Hg, mm
1	1.47 E + 1	2.99 E + 1	3.40 E + 1	1.04 E + 1	1.01 E + 2	7.60 E + 2
6.80 E − 2	1	2.04 E + 0	2.31 E + 0	7.04 E − 1	6.90 E + 0	5.17 E + 1
3.34 E − 2	4.91 E − 1	1	1.14 E + 0	3.46 E − 1	3.39 E + 0	2.54 E + 1
2.94 E − 2	4.33 E − 1	8.81 E − 1	1	3.05 E − 1	2.98 E + 0	2.24 E + 1
9.66 E − 2	1.42 E + 0	2.89 E + 0	3.28 E + 0	1	9.79 E + 0	7.34 E + 1
9.87 E − 3	1.45 E − 1	2.95 E − 1	3.35 E − 1	1.02 E − 1	1	7.50 E + 0
1.32 E − 2	1.93 E − 2	3.94 E − 2	4.47 E − 2	1.36 E − 2	1.33 E − 1	1

Power				
U.S. customary units				SI units
British thermal units per second	Horsepower	Foot-pounds per second	Foot-pounds per minute	Kilowatts
1	1.41 E + 0	7.78 E + 2	4.67 E + 4	1.06 E + 0
7.07 E − 1	1	5.50 E + 2	3.30 E + 4	7.46 E − 1
1.29 E − 3	1.82 E − 3	1	6.00 E + 1	1.36 E − 3
2.14 E − 5	3.03 E − 5	1.67 E − 2	1	2.26 E − 5
9.48 E − 1	1.34 E + 0	7.38 E + 2	4.43 E + 4	1

Energy				
U.S. customary units			SI units	
Horsepower-hours	British thermal units	Foot- pounds	Kilowatt-hours	Joules
1	2.54 E + 3	1.98 E + 6	7.46 E − 1	2.68 E + 6
3.93 E − 4	1	7.78 E + 2	2.93 E − 4	1.05 E + 3
5.05 E − 7	1.29 E − 3	1	3.77 E − 7	1.36 E + 0
1.34 E + 0	3.41 E + 3	2.66 E + 6	1	3.60 E + 6
3.73 E − 7	9.48 E − 4	7.38 E − 1	2.78 E − 7	1

A-1. References

1. Vennard, J. K., and R. L. Street. *Elementary Fluid Mechanics*, 5th ed. John Wiley, New York (1975).
2. Metcalf & Eddy, Inc. *Wastewater Engineering, Treatment, Disposal, and Reuse*, 4th ed. Revised by G. Tchobanoglous, F.L. Burton, and D.H. Stensel. McGraw-Hill, New York (2003).

INDEX

Index Terms	Links			

Index Terms	Links		

Index Terms	Links				

Index Terms / Links

Index Terms	Links	

Index Terms	**Links**	

Index Terms	Links	

Index Terms | Links

Index Terms	**Links**	

Index Terms	**Links**			

Index Terms	Links

Index Terms	Links	

Index Terms Links

Index Terms	**Links**	

Index Terms	Links		

Index Terms	**Links**	

Index Terms **Links**

Index Terms	Links	

Index Terms	**Links**

Index Terms — Links

Index Terms	**Links**		

Index Terms	Links	

Index Terms	Links	

Index Terms	Links		

Index Terms	Links	

Index Terms	Links			
Ease of operation (*Cont.*)				
pressure gauges	24.3			
surge control	24.4			
wet wells and other pump intakes	24.4			
landscaping for	24.2			
mechanical considerations	24.6			
access	24.8			
piping	24.7			
pumps	24.8			
operator preferences for	24.14			
County Sanitation Districts, Orange County CA	24.15			
Los Angeles County Sanitation Districts	24.14			
Seattle Metro pumping stations	24.15			
site selection for	24.1			
smooth-running and reliable pumps	24.8			
achieving	24.10			
baseplates and foundations	24.9			
reduction in annual O&M costs through	24.10	24.11		
rotating equipment balance quality and bearing longevity	24.9			
shaft alignment	24.9			
torsional vibration and	24.10			
vibration effects on bearing lift	24.10			
specifications for	24.14			
standby facilities design for	24.13			
Ebullient cooling systems	14.10			
advantages and disadvantages	14.13			
schematic diagram	14.15			
unsuitability of lean-burn engines for	14.8			
Eccentric plug valves	5.4	5.5	5.9	5.10
for sludge pumping	19.25			
Economics				
for buried piping	4.16			
design blunders related to	27.18			

Index Terms	**Links**	

Index Terms	Links

Index Terms Links

Index Terms | **Links**

Index Terms	Links			

Index Terms | Links

Index Terms Links

Index Terms	**Links**	

Index Terms	Links

Index Terms / Links

Index Terms	Links			

Index Terms	Links		

Index Terms **Links**

Index Terms	Links

Index Terms	Links	

Index Terms	Links	

Index Terms	**Links**

Index Terms Links

Index Terms	Links				

I

Index Terms	**Links**		

Index Terms Links

Index Terms	**Links**

Index Terms	Links		

Index Terms	Links	

Index Terms | Links

Index Terms Links

Index Terms	Links	

M

Index Terms	Links		

Index Terms	Links		

N

Index Terms	Links	

Index Terms	Links	

O

Index Terms | Links

P

Index Terms	Links		
Palmer-Bowlus flume	3.26	12.50	20.21
Palmer-Bowlus meter	3.17		
advantage of	3.20		
in flow rate measurement	3.25		
Parametric resonances	22.17		
Parking lot lighting	8.34		
Parshall flume	3.17		
Part winding starters	8.19		
PARTFULL program	3.12	3.14	
Partially closed valves	3.23		
Particulates	14.17		
Partnering, avoiding litigation through	1.7		
Parts per billion (ppb)	2.6		
Parts per million (ppm)	2.6		
Pascal	2.11		
Patterson electronic catalog	12.76		
design point selection list	12.76		
Patterson performance curves	12.76	12.77	
Payback analysis	13.10		
PBM T-port valves	7.3		
Peak demand	12.50		
Peak flow rate, in small distribution system booster	18.37		
Peak heat gain	23.18		
Peak influent rate (PIR)	2.6		
Peak operating conditions, centrifugal pumps	10.31		
Peak shaving, and adjustable-speed drives	15.30		
Performance, manufacturer guarantee at specified conditions	16.5		
Performance and payment bond	28.3		
Performance curves, vortex pump	19.23		
Performance degradation, centrifugal pumps	10.16		
Performance requirements			
calculation of	12.55		
in specification examples	C.18		

Index Terms | Links

Index Terms	Links		
Pig stop	4.37		
Pigging			
design blunders related to	27.15		
in sludge pumping systems	19.27		
Pigs	2.11	19.25	
use in field measurement	3.23		
Pinch valves	5.13		
for sludge pumping	19.25		
Pipe			
effects on wave speed and pressure	6.6		
flow data	B.1	B.2	B.15
flow measurements in	20.13		
geometric properties	B.13		
hydraulic elements of circular	B.14		
maximum allowable flow rates	B.17	B.18	
safety factors for	7.19		
Venturi meter in	3.15		
Pipe bending modes, boundary conditions for	22.37		
Pipe cleaning			
flowmeter concerns	19.26		
pigging	19.27		
rodding	19.27		
in situ	3.8		
in sludge pumping applications	19.27		
Pipe diameter			
calculation of average	3.24		
field measurement of	3.23		
and valve slam	5.16		
Pipe fittings			
headlosses in	3.11		
symbols	2.14		
Pipe flushing/draining, in sludge pumping systems	19.27		
Pipe formulas	3.6		

Index Terms	Links	

Index Terms	**Links**

Index Terms | Links

Index Terms	Links	

Index Terms	Links	

Index Terms | Links

Index Terms	Links

Index Terms Links

Index Terms | Links

Index Terms	Links	

Index Terms	**Links**	

Index Terms	Links	

Index Terms Links

Index Terms	Links	

Index Terms	Links	

Index Terms Links

Index Terms	**Links**	

Index Terms	Links

Index Terms Links

Index Terms	Links		

Index Terms	**Links**	

Index Terms	Links			
Startup costs	1.4			
State fish and game authority	18.8			
Static deflection	22.22			
Static differential pressure	6.12			
Static discharge head	2.12	10.3		
Static head	2.13	3.23		
determining for pump testing	16.4			
of pump intake water	10.22			
in V/S wet well pumping	15.4			
and valve slam	5.16			
Static imbalance, vibration and	12.60			
Static lift variations	10.37	10.41	12.49	12.55
in motors for wastewater lift station	13.17			
Static suction head	2.13	10.3		
Static suction lift	2.13			
Station auxiliaries	25.30			
alarms	25.34			
compressed air systems	25.33			
hoisting equipment	25.31			
sump pumping system	25.32			
ventilation	25.32			
water systems	25.34			
Station curves, centrifugal pumps	10.38			
Station head-capacity (H-Q) curves	1.3			
Station location, and pump selection	12.48	12.53		
Station losses	2.13			
Station piping	2.13			
Station utilities design, for wastewater pumping systems	17.5			
Station voltage selection	8.39			
Stationary field eddy-current couplings	15.39			
Stator winding temperature	13.16			
Stators	13.3	13.7		
Steady bearing	2.13			

Index Terms	Links	

Index Terms	Links	

Index Terms	Links				

Index Terms	**Links**		

Index Terms	Links			
Surge control	5.24			
air chambers with turbine pumps	7.18			
centrifugal high-service pumps in	7.18			
deep well pumps	7.15			
design for ease of operation and low maintenance	24.4			
with high inertia	7.9			
low suction-lift turbine pumps for	7.18			
pipeline design for	7.19			
pump-control valves in	7.18			
quick-closing check valves for	7.18			
for raw wastewater pumping stations	7.18			
rotational inertia increase	7.19			
turbine booster pumps in	7.18			
in wastewater pumping design	17.3			
for water pumping stations	7.15			
Surge protection	8.31			
water booster pumping station	12.79			
Surge relief valves	7.12			
advantages and disadvantages of	7.14			
in deep well pump systems	7.15			
Surge tanks				
avoiding in wastewater service	7.9			
one-way	7.10			
two-way	7.10			
Surge theory	6.1			
Suspended material	2.13			
in pump selection process	12.50			
Swan Island Pumping Station (SIPS) FSI	12.9			
Swedish pumping stations	17.27			
Swing check valves	5.17	5.18	5.20	7.8
advantages and disadvantages of	7.14			
design blunders of	27.13			
headlosses	B.12			
proximity to air chambers	27.4			
in transient control	7.12			

Index Terms	Links	

Index Terms	**Links**		

Index Terms	Links	

Index Terms	Links			
Tranquil flow	2.13	3.16		
Transformers	8.10			
factory testing per ANSI	C.57	12.90	9.23	
primary and secondary protection sizing	9.14			
primary fuses	9.9			
sizing in electrical design	9.6	9.13		
Transient analysis	6.12	17.10	17.15	17.22
of high-lift pumping systems	6.13			
Transient control strategies	7.1			
check valve selection	7.8			
for column separation	7.5			
filling empty pipelines	7.8			
in power failure	7.4			
preventing valve slam	7.7			
for pump shut-down	7.6			
for pump start-up	7.5			
site-specific nature of	7.1			
undesirable options				
for wastewater systems	7.4			
for wastewater systems	7.2			
for water systems	7.4			
Transient control valves	7.11			
air and vacuum control	7.11			
air release valves	7.11			
vacuum relief valves	7.11			
in wastewater force mains	7.12			
in water mains	7.11			
check valves	7.12			
surge anticipation valves	7.12			
surge relief valves	7.12			
Transient voltages	13.24			
Transients				
analysis checklist	6.13			
analysis methods	6.2			
causes of	6.1	7.25		

Index Terms | Links

Index Terms	Links	

Index Terms	Links	

Index Terms	Links

V

Index Terms Links

Index Terms	Links	

Index Terms | Links

Index Terms	Links	

Index Terms	Links		

Index Terms   Links

Index Terms / Links

W

Index Terms Links

Index Terms	Links

Index Terms | Links

Index Terms	**Links**

Index Terms	Links				

Index Terms | Links

Index Terms	Links	

Index Terms   Links