ANSI/HI 9.6.6-2009

Approved American National Standard

ANSI

American National Standard for

Rotodynamic Pumps

for Pump Piping

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Hydraulic Institute

6 Campus Drive
First Floor North
Parsippany, New Jersey
07054-4406
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Approved July 28, 2009

American National Standards Institute, Inc.

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Foreword (Not part of Standard)

Scope

This standard applies to rotodynamic pump types, in all worldwide markets. It provides required and recommended practices for pump piping which, if followed, should reduce the risk of the pump failing to perform properly due to interaction with the system. Excluded is any piping integral to the pump unit, such as auxiliary or lubricant piping. This document is intended to complement ANSI/HI 9.8 *Pump Intake Design*. In order to eliminate the possibility of overlapping and possibly conflicting standards, this document is considered to be applicable as follows:

- All piping downstream and upstream from the pump
- Upstream from the pump this document ceases to be applicable when the pipe enters a tank, vessel, or other intake structure

Purpose and aims of the Hydraulic Institute

The purpose and aims of the Institute are to promote the continued growth of pump knowledge for the interest of pump users and manufacturers and to further the interests of the public in such matters as are involved in manufacturing, engineering, distribution, safety, transportation and other problems of the industry, and to this end, among other things:

a) To develop and publish standards for pumps;

b) To collect and disseminate information of value to its members and to the public;

c) To appear for its members before governmental departments and agencies and other bodies in regard to matters affecting the industry;

d) To increase the amount and to improve the quality of pump service to the public;

e) To support educational and research activities;

f) To promote the business interests of its members but not to engage in business of the kind ordinarily carried on for profit or to perform particular services for its members or individual persons as distinguished from activities to improve the business conditions and lawful interests of all of its members.

Purpose of Standards

1) Hydraulic Institute Standards are adopted in the public interest and are designed to help eliminate misunderstandings between the manufacturer, the purchaser and/or the user and to assist the purchaser in selecting and obtaining the proper product for a particular need.

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Definition of a Standard of the Hydraulic Institute

Quoting from Article XV, Standards, of the By-Laws of the Institute, Section B:

"An Institute Standard defines the product, material, process or procedure with reference to one or more of the following: nomenclature, composition, construction, dimensions, tolerances, safety, operating characteristics, performance, quality, rating, testing and service for which designed."

Comments from users

Comments from users of this standard will be appreciated, to help the Hydraulic Institute prepare even more useful future editions. Questions arising from the content of this standard may be directed to the Hydraulic Institute. It will direct all such questions to the appropriate technical committee for provision of a suitable answer.

If a dispute arises regarding contents of an Institute standard or an answer provided by the Institute to a question such as indicated above, the point in question shall be sent in writing to the Technical Director of the Hydraulic Institute. The inquiry will then be directed to the appropriate technical committee for provision of a suitable answer.

Revisions

The Standards of the Hydraulic Institute are subject to constant review, and revisions are undertaken whenever it is found necessary because of new developments and progress in the art. If no revisions are made for five years, the standards are reaffirmed using the ANSI canvass procedure.

Disclaimers

This document presents accepted best practices based on information available to the ISO/TC-115 SC-3 Work Group 4 and Hydraulic Institute as of the date of publication. Nothing presented herein is to be construed as a warranty of successful performance under any conditions for any application.

Units of measurement

Metric units of measurement are used and corresponding US customary units appear in brackets. Charts, graphs, and sample calculations are also shown in both metric and US customary units.

Because values given in metric units are not exact equivalents to values given in US customary units, it is important that the selected units of measure be stated in reference to this standard. If no such statement is provided, metric units shall govern. See Section 9.6.6.8, List of acronyms.

In the application of this standard, the pump rated flow shall be used as the design flow for the basis of the piping design.

In this document references to pipe diameter will be understood to mean the nominal pipe size, not the pipe inside diameter.

Consensus for this standard was achieved by use of the Canvass Method

The following organizations, recognized as having an interest in the standardization of rotodynamic pumps, were contacted prior to the approval of this revision of the standard. Inclusion in this list does not necessarily imply that the organization concurred with the submittal of the proposed standard to ANSI.

4B Engineering
Alden Research Laboratory, Inc.
Berryman & Henigar
Black & Veatch
Brown and Caldwell
Carollo Engineers
CDM
Cheng Fluid Systems
CTE
Diagnostic Solutions, LLC
E.I. duPont de Nemours & Co., Inc.
ENSR International
Fairbanks Morse Pump
Flowserve Pump Division
Fluid Sealing Association
Flygt - ITT Industries
GIW Industries, Inc.
Greeley-Hansen, LLC
Grundfos Pumps Corporation
Healy Engineering
ITT - Res. & Comm. Water Group
ITT Industrial & BioPharm Group
John Anspach Consulting
John Crane Inc.
MechTronix Engineering
Northwest Hydraulic Consultants
Patterson Pump Company
Peerless Pump Company
Pentair Water
Powell Kugler, Inc.
Pumping Machinery, LLC
Reddy-Buffaloes Pump, Inc.
Rockwell Automation
Shell Global Solutions International BV
Sulzer Process Pumps (US) Inc.
Sulzer Pumps (US) Inc.
Suncor Energy Inc.
TACO, Inc.
Tuthill Pump Group
Union Sanitary District
Weir Floway Pumps
Weir Minerals North America
Weir Specialty Pumps
Whitley Burchett & Associates

Although this standard was processed and approved for submittal to ANSI by the Canvass Method, a working committee met many times to facilitate its development. At the time it was developed, the committee had the following members:

Co-chairman: Pat Moyer - ITT Bell & Gossett

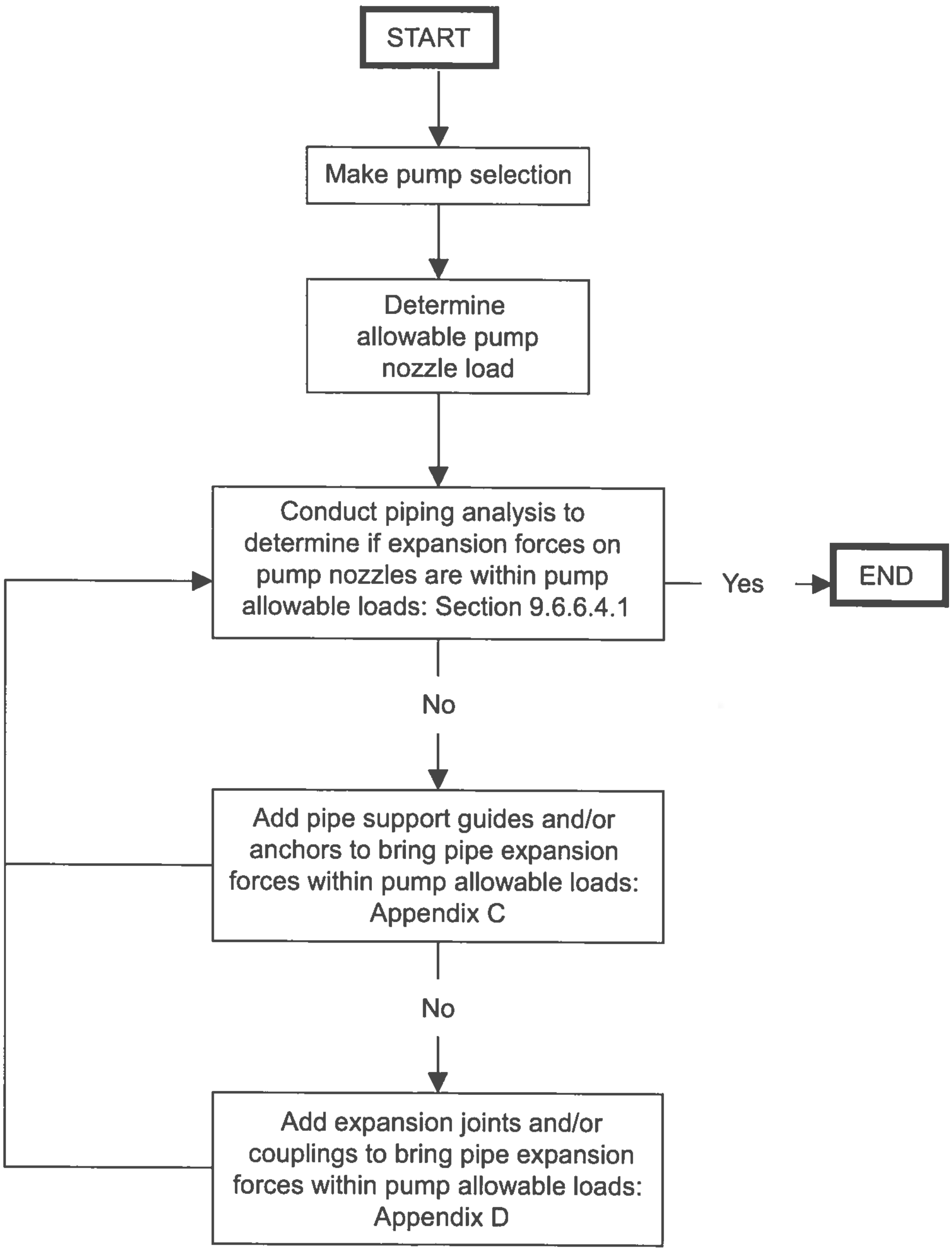
Co-chairman: Tom Angle, Weir Specialty Pumps

Other Members:

Stefan Abelin	ITT Flygt
Charles Allaben	CDM
Ed Allis	Peerless Pumps Co.
John Anspach	John Anspach Consulting
René Barbarulo	David Brown Guinard Pumps
Bill Beekman	Floway Pumps
Alan Budris	Formerly ITT Goulds
Fred Buse	Consultant, Flowserve
Jack Claxton	Patterson Pumps
Mike D'Ambrosia	King County
Tom Hendrey	Whitley Burchett & Associates
Al Iseppon	Pentair Water, Delevan
Garr Jones	Brown & Caldwell
Yuri Khazanov	Yeomans Chigago Corporation
Jim Osborne	A.R. Wilfley & Sons
Y.J. Reddy	Reddy-Buffaloes Pump, Inc.
Y.R. Reddy	Reddy-Buffaloes Pump, Inc.
Jim Roberts	Bell & Gossett – ITT Fluid Technology
Bob Rollings	DuPont
Aleks Roudnev	Weir Slurry North America
Steve Schmitz	ITT Bell & Gossett
Arnold Sdano	Fairbanks Morse Pump
Ernest Sturtz	CDM
Roger Turley	Flowserve

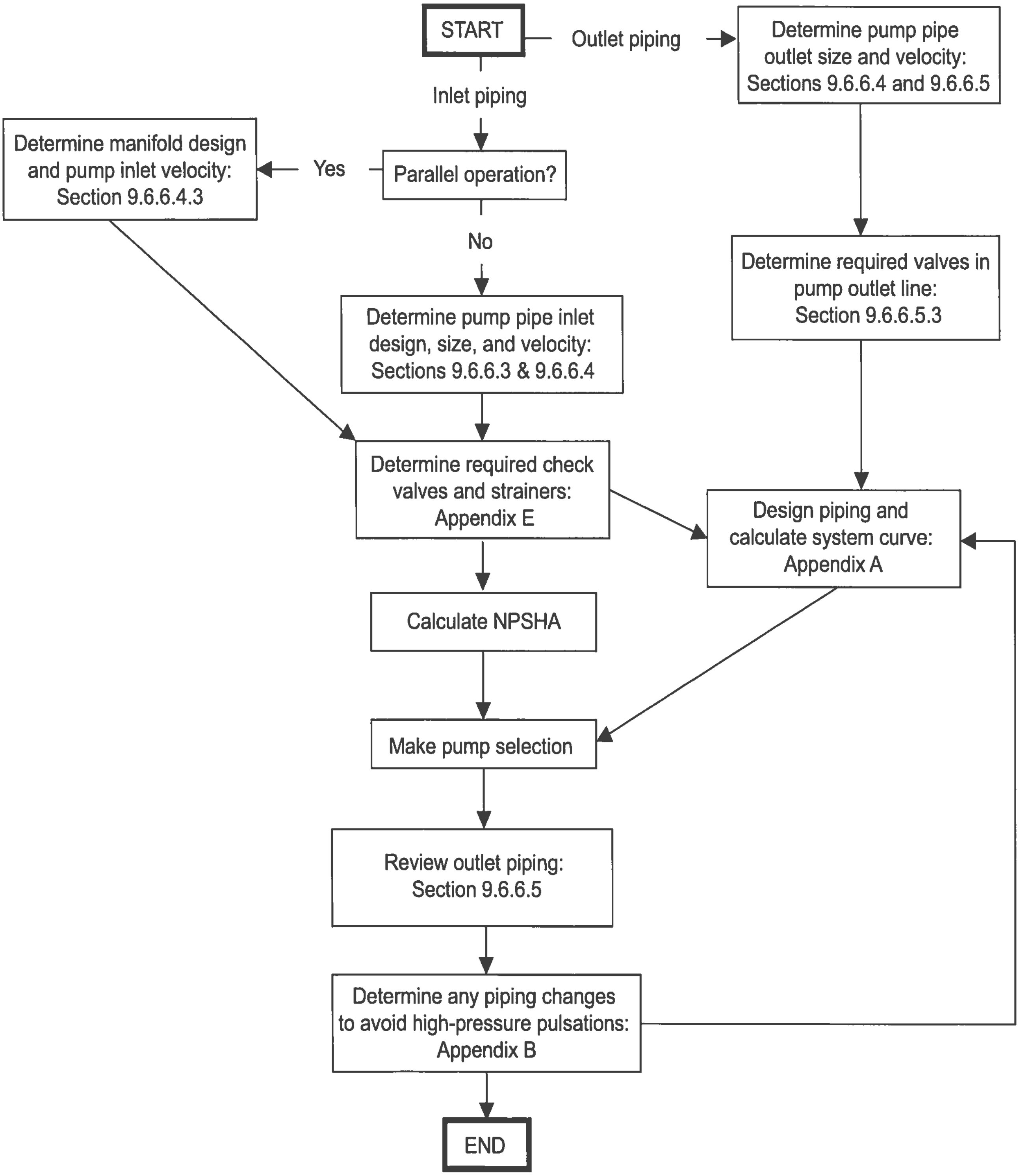
Flowchart for use of standard (mechanical considerations)

NOTE: This flowchart is intended as a guide to the use of this standard and can be used to locate the appropriate sections in this standard. The chart is not a substitute for comprehension of the complete standard.



Flowchart for use of standard (hydraulic considerations)

NOTE: This flowchart is intended as a guide to the use of this standard and can be used to locate the appropriate sections in this standard. The chart is not a substitute for comprehension of the complete standard.



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9.6.6 Pump piping for rotodynamic pumps

9.6.6.1 Objective

The objective of this standard is to provide piping requirements for rotodynamic pump piping, and to educate users about the effects and interactions of inlet (suction) and outlet (discharge) piping on rotodynamic pump performance. The standard covers pump suction liquid conditions, such as the effects of piping on the net positive suction head available (NPSHA) to the pump (which controls cavitation in the pump), the developed pressure of the pump, hydraulic and piping loads on the pump, pump noise, and pump vibration.

9.6.6.2 Introduction

The function of pump piping is to provide a conduit for the flow of liquid to and from a pump, while not adversely affecting the performance or reliability of the pump. In addition it should be noted that a well-designed piping system will usually be more energy efficient than a poorly designed system.

The function of suction piping is to provide a uniform velocity profile or symmetric approaching flow to the pump inlet (suction) connection with sufficient pressure to avoid damaging cavitation in the pump. An uneven distribution of flow is characterized by strong local currents and swirls. The ideal approach is a straight pipe, coming directly to the pump, with no turns or flow-disturbing fittings close to the pump. Failure of the inlet (suction) piping to deliver the liquid to the pump in this condition can lead to noisy operation, random axial load oscillations, premature bearing or seal failure, cavitation damage to the impeller and inlet portions of the casing, and occasionally damage due to liquid separation on the discharge side. See pump intake design standards [1, 2] and *Recommendations for Fitting of Inlet and Outlet on Piping* [3] for additional intake recommendations.

Outlet (discharge) piping flow characteristics normally will not affect the performance and reliability of a rotodynamic pump, with a few exceptions. Sudden valve closures can cause excessively high water-hammer-generated pressure spikes to be reflected back to the pump, possibly causing damage to the pump. Where there may be a sudden closure of a check valve or sudden stopping of the pump, a transient flow analysis may be required (see Section 9.6.6.5.4). Outlet (discharge) piping can affect the starting, stopping, and priming of the pump. The outlet (discharge) piping configuration can also alter any discharge flow recirculation that might extend into the outlet (discharge) piping at very low flow rates.

Three of the more common detrimental effects from poor pump piping details are the excessive loads that the piping can place on a pump because of pipe misalignment with the pump connections, failure to properly restrain the pump to the structure to transfer the unbalanced force generated by the pump, and the weight of unsupported or poorly supported valves and fittings or vertical in-line pumps can place on the piping. Excessive nozzle loads can be caused by thermal expansion of the pipe, unsupported piping and equipment weight, axially unrestricted couplings, and misaligned piping. Excessive pump nozzle loads lead to misalignment of the pump shaft with the driver shaft, mechanical seal failures, bearing failures, binding or rubbing of the pump rotor, and in extreme cases, failure of pump nozzles or feet.

9.6.6.3 Inlet (suction) piping requirements

Inlet flow disturbances, such as swirl, unbalance in the distribution of velocities and pressures, and sudden variations in velocity can be harmful to the hydraulic performance of a pump, its mechanical behavior, and its reliability. Usually the higher the energy level and specific speed of a pump, and the lower the NPSH margin, the more sensitive the pump's performance is to suction conditions.

All inlet (suction) fitting joints shall be tight, especially when the pressure in the piping is below atmospheric, to preclude air leaking into the fluid. Any valves in the inlet (suction) line should be installed with stems horizontal to eliminate the possibility of air accumulation. For pumps operating with a suction lift, the inlet (suction) line should slope constantly upwards toward the pump, with a minimum slope of 1% [11] (Figure 9.6.6.3). For most pumping systems, an inlet (suction) shut-off valve should be installed in the suction piping for system isolation.

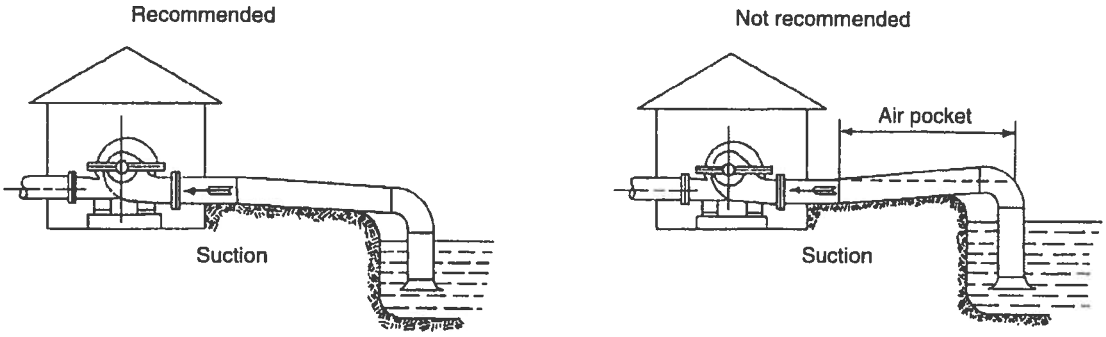


Figure 9.6.6.3 — Suction pipe design

In general, as liquid travels through a piping network, entrained air tends to rise to the highest point. If the pipeline slopes upward, then the velocity of the liquid will move the air bubbles towards this high point. In contrast, if the pipeline is fairly flat and the inside surface of the pipe is very rough, or the pipeline slopes downward, the fluid velocity may not be sufficient to keep the air bubbles moving. As a consequence, it is possible for a pocket of air to collect at high points and gradually reduce the effective liquid flow area. This reduction in area can create a throttling effect similar to a partially closed valve. It is also possible that a slug of air may be swept into the pump during a restart, causing a partial or complete loss of pump prime, especially where the inlet (suction) line is kept full by a foot valve at its intake (see Appendix E).

Any amount of entrained gas in the fluid may adversely affect pump performance. Check with the pump manufacturer to determine allowable levels of entrained gas.

9.6.6.3.1 Inlet (suction) pipe size/velocity requirements

The suction pipe shall be at least as large as the pump suction nozzle. Valves and other flow-disturbing fittings located in pump inlet (suction) piping should be at least one pipe size larger than the pump inlet (suction) nozzle, with the exception of continuous-bore, 100% open valves (such as full-ported ball valves).

The maximum velocity at any point in the inlet (suction) piping is 2.4 m/s (8 ft/s), unless the liquid is a slurry, in which case it may need to be higher to maintain solids suspension. Individual values above 2.4 m/s (8 ft/s) should be evaluated with respect to flow distribution, erosion, NPSH, noise, water hammer, and the manufacturer's recommendations.

For fluids close to the vapor pressure, the velocity must be kept low enough to avoid flashing of the liquid in the piping, especially when fittings are present. A reduction in pressure and/or flashing can liberate dissolved air and vapor, which can be carried into the suction of the pump.

9.6.6.3.2 Effect of piping-generated swirl

The most disturbing flow patterns to a pump are those that result from swirling liquid that has traversed several changes of direction in various planes. Liquid in the inlet (suction) pipe should approach the pump in a state of straight steady flow. When fittings, such as tees and elbows (especially two elbows at right angles) are located too close to the pump inlet (suction), a spinning action, or swirl, is induced. This swirl may adversely affect pump performance by reducing efficiency, head, and NPSH available, and potentially causing noise, vibration, and damage. It is, therefore, recommended that a straight uninterrupted section of pipe be installed between the pump and the nearest fitting (see Figure 9.6.6.3.3a), per the minimum lengths specified in Table 9.6.6.3.2. If the minimum recommended pipe lengths cannot be provided, flow-straightening devices should be considered (see Appendix E.2).

Table 9.6.6.3.2 — Minimum required straight pipe length (L2) before pump suction inlet (refer to Figure 9.6.6.3.3a)

Fitting	Number of pipe diameters[a,b]		Comments
	Long radius	Short radius	
90° elbow	4	5	For double suction pumps add one pipe diameter if the elbow and pump shaft are in the same plane.
Reducing elbow with <30% area reduction	3	4	
Reducing elbow with 30 to <50% area reduction	2	3	
Reducing elbow with >50% area reduction	0	1	
Reducers	Concentric	Eccentric	
1 pipe size reduction	0 (<10°)[c]	0 (<20°)[c]	
2 pipe size reductions	0 (<20°)	1 (<30°)	
3 pipe size reductions	1 (<20°)	2 (<30°)	
4 pipe size reductions	2 (<20°)	3 (<40°)	
5 pipe size reductions	3 (<30°)	4 (<40°)	
45° lateral	4		
Tee, branch flow, 90°	5		
Continuous-bore valve (100% open)[d]	0		
Eccentric plug valve - rectangular 80% port	4		Valve shaft should be parallel to the pump shaft for split case pumps.
Eccentric plug valve - rectangular 100% port	3		
Butterfly valve[e]	2		

Notes:

- The next fitting upstream from the fitting closest to the pump shall have the number of pipe diameters referenced in this table between it and the fitting closest to the pump.
- Globe valves and diaphragm valves shall not be used on the suction side of the pumps.
- For vertical in-line pumps, elbows may be located in an orientation parallel to the pump shaft within two diameters for long radius and three diameters for short radius elbows.
- Length of pipe between fitting and pump may be reduced from that presented in the above table with approval of the pump manufacturer.
- Alternative values may be used if they are substantiated by long-term experience in a particular installation or application, by physical testing, or by CFD (Computational Fluid Dynamics) modeling (see ANSI/HI 9.8, Section 9.8.7, for limitations).
- The data in this table are based on commercially available fittings.
- Although special flow-straightening devices such as internal ribs, vanes, or other arrangements have been applied successfully, such devices are beyond the scope of this document at this time.

[a] For fittings with less than two pipe diameters required, consideration must be made for providing additional pipe length for ease of pump installation and removal.

[b] The pump manufacturer is responsible for successful operation when it supplies fittings that are directly connected to the pump, whether or not they conform to the requirements of this table.

[c] The angles given are the maximum per side for standard commercial fittings. Nonstandard fittings having a greater angle per side should use the number of pipe diameters corresponding to that angle, regardless of the number of pipe reductions.

[d] Ball and gate.

[e] Care must be taken when placing a butterfly valve downstream from a fitting such as an elbow or tee. The disk should not be oriented such that it can channelize flow and thus exacerbate the nonuniform hydraulic conditions generated by the upstream fitting.

9.6.6.3.3 Required straight pipe lengths

The use of flow-disturbing fittings on the inlet (suction side) of the pump needs to be carefully evaluated. In general, pumps shall have an uninterrupted and unthrottled flow into the inlet (suction) nozzle (no flow-disturbing fittings for some minimum length). Flow-disturbing fittings should not be connected directly to the pump inlet without approval from the pump manufacturer. Flow disturbances on the inlet (suction) side of the pump can lead to deterioration in performance (lower head and/or lower NPSHA), and damage leading to shortened impeller, mechanical seal, and bearing life (from cavitation, pulsation surges, and excessive radial and axial forces). Isolation valves, strainers, and other devices used on the inlet (suction) side of the pump shall be sized and located to minimize disturbance of the flow into the pump (Table 9.6.6.3.2).

A concentric reducer is recommended for vertical inlet (suction) pipes or horizontal installations where there is no potential for air or vapor accumulation. Eccentric convergent reducers are normally used for horizontal installations where there is potential for air or vapor accumulation. The flat side shall be located on the top, unless the inlet (suction) line approaches from above, in which case either a concentric or eccentric convergent reducer (with the flat side on the bottom) should be used.

Table 9.6.6.3.2 (reference Figure 9.6.6.3.3a) presents the minimum length of straight pipe required immediately upstream of the pump inlet (suction) nozzle. This straight pipe section is to be the same diameter as that of the pump suction nozzle. The velocity requirements of Section 9.6.6.3.1 do not apply to this section of pipe. This table applies for various fitting types. If several fittings are adjacent, the longest length requirement will govern. Failure to provide the minimum pipe lengths specified in Table 9.6.6.3.2 can lead to the hydraulic and mechanical issues described in the first paragraph of this section. Double suction pumps are a sensitive application because an elbow in the plane of the shaft mounted on the suction flange will direct more flow to one side of the impeller than the other (see Figure 9.6.6.3.3b).

9.6.6.4 Inlet and outlet general piping requirements

9.6.6.4.1 Pipe nozzle alignment/pipe expansion load

Proper piping and pump layout design and analysis prior to installation of the system are absolutely essential to the life and reliability of a pump. This can ensure that nozzle loads remain below acceptable limits for installed pumps by providing a properly supported piping system, which minimizes field adjustments during installation, saving both time and money.

This document presents a general overview on pipe load analysis at the interface with a rotodynamic pump. Additional details can be found in references 4, 5, 6, 7, 8, and 9.

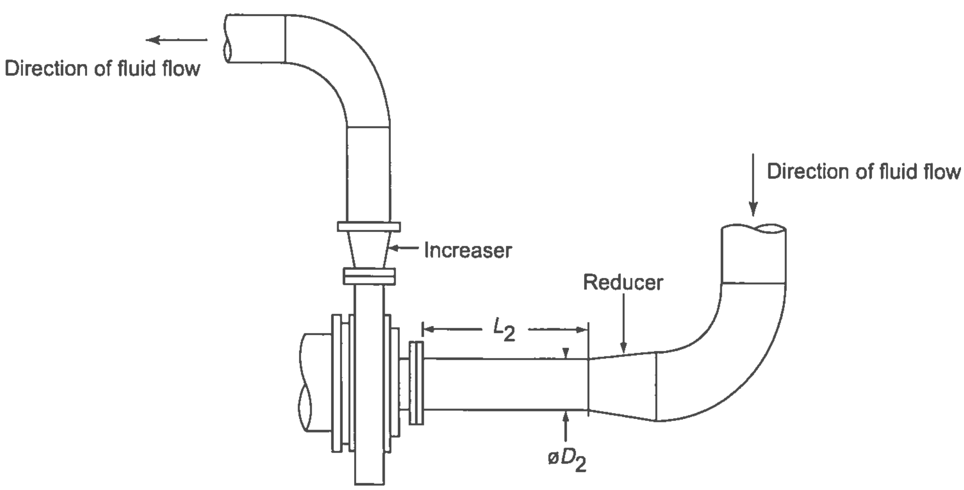


Figure 9.6.6.3.3a — Pump installed with elbows

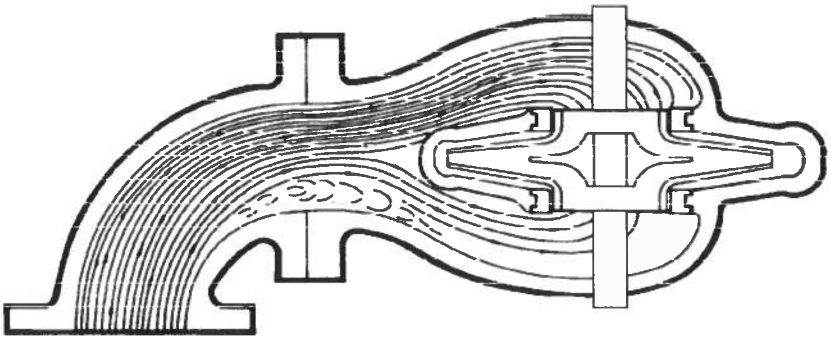


Figure 9.6.6.3.3b — Undesirable effects of elbow directly on suction

Nozzle loads affect pump operation in various ways. At low levels, the effects may be insignificant. At high levels, nozzle loads can contribute to the following:

a) Coupling misalignment, which leads to

- Heat buildup in bearings, which decreases bearing life
- In severe instances, fractures within the pump case
- Decreased coupling life
- Increased noise and vibration levels
- Breakage of pump shaft

b) Shaft movement, which leads to reduced packing, shaft sleeve, and mechanical seal life

c) Fatigue or failure of the shaft

d) Excessive deformation or catastrophic structural failure of

- Pump hold-down bolts or support structure
- Pump nozzles

e) Pump casing gasket leaks

f) Pipe to pump flange leaks

g) Decreased mean time between repair or failure

9.6.6.4.1.1 Pump nozzle flange analysis

Analysis of the thermal and static loads on the pump nozzles are done at both the working and ambient condition, and the effects of supports and braces used in the piping must also be considered (see Section 9.6.6.4.2). If pipe supports are improperly selected and specified in the design, in an effort to relieve working loads on the nozzles at working temperature, the supports can cause unforeseen loads to be transmitted to the pump nozzles in the ambient state. As the pipe attached to the pump increases or decreases in temperature, it results in thermal expansion or contraction of the pipe, which causes the pipe to deform from its original ambient condition. These deformations will produce a strain in the pipe. If the pipe meets any resistance to this strain, such as a pump flange connection, the strain will be translated into a force on the pump in the direction of the deformation. The thermal growth of the pump nozzles (working and ambient) should also be considered in the analysis.

Nozzle loads can be profoundly affected by the degree of misalignment between the pump nozzle and the connection piping. Misalignment can be either at an angle to the centerline of the nozzle and/or a lateral offset to the centerline of the nozzle. Thermal expansion and contraction, acting on the piping, pump, or both can often exacerbate misalignment present under ambient conditions at the time of installation. Frequently both conditions can be present. Piping design features may be configured to accommodate both by providing a pair of flexible couplings, spaced appropriately to accommodate the potential angular misalignment, at each connection. Flexible couplings on the discharge nozzle piping must be restrained, as described in Appendix D. Such restraints must be designed to accommodate the expected misalignment.

Both the analysis of the inlet (suction) and outlet (discharge) lines need to be completed before the forces and moments on the pump can be calculated. The pump analysis, which must be done in conjunction with the pump manufacturer, should consider:

a) Suction and discharge nozzle stresses.

b) Hold-down bolt stress.

c) Pump slippage on the baseplate.

d) Internal clearances and distortion.

e) Movement of pump shaft.

The Hydraulic Institute, American Petroleum Institute (API) [8], and CEN [3, 5, 6] have tabulated the allowable loads for pump nozzles for pumps manufactured using certain pump casing materials. Care must be taken to verify the allowable nozzle loads specified by the pump manufacturer, since they may be less than what is stated in these standards.

9.6.6.4.2 Pipe supports/anchors

Proper installation of piping, supports, and restraint fixtures is imperative to obtain optimum performance and reliability from attached pumps and rotating equipment. Pumps require precise shaft alignment to operate properly. The installed piping should be supported by the surrounding structure, not by the pump itself. Pipe supports are used to hold the weight of the pipe off the pump nozzles, while restraints and guides are used to redirect the forces generated by thermal effects away from the pump nozzle. Pipe supports may be designed to handle vertical, horizontal, axial, thermal, and seismic forces.

Some new installations will have had a flexibility analysis done (discussed later in this section) to determine size and placement of supports and guides. Design shall include an evaluation of the forces from fast-acting, shock-inducing check valves (see Section 9.6.6.5.4). The installation must take place while the pipe is in its ambient condition with no fluid in the lines, where the loads and displacement experienced will not necessarily correspond to the analysis. Careful installation and field checks of all support fixture settings and placements is essential to ensure the piping system will perform under working conditions, as the analysis predicted.

9.6.6.4.2.1 Design considerations

System flexibility is the ability of the piping to absorb the thermal loads and displacements placed on it. All piping systems will include terminal connections of some kind, such as pumps, vessel connections, or anchors, which must be considered as fixed points in the system. These terminal connections, particularly in the case of pumps, are sensitive to the forces generated by the attached pipe. Supports and restraints will aid in transferring or redirecting the load, but the piping system must have adequate flexibility to relieve the forces.

9.6.6.4.2.2 Cold spring

Cold spring is the practice of purposely fabricating certain sections of pipe too short or too long to compensate for thermal movements at system operating conditions. If properly designed, the cold spring will effectively relieve loads on the pump nozzles and terminal connections, but can easily overload attached equipment in the ambient condition. Flange mating and alignment during installation can be extremely difficult when pipes are intentionally made too short. Cold spring adjustment can be easily forgotten or unknown when making future modifications to the existing piping system. Cold spring is not recommended by this standard.

9.6.6.4.2.3 Final installation

Final installation of restraints and anchors should take place after the pipe has been completely installed and all terminal and equipment connections are made. Provisions for a final piping field weld close to the pump may be used to ensure the piping system will be near a zero load condition [11, 12]. Flanges of connecting piping shall not be sprung into position. Pipe flange boltholes shall be lined up with nozzle boltholes, within the limits of the flange bolthole manufacturing clearance, to permit insertion of bolts without applying any external force to the piping. The use of come-alongs or other means of providing a large amount of force to align the piping with the pump flanges shall not be allowed. This should ensure that the piping system will be within allowable pump nozzle load requirements prior to hydrotest and start-up [11]. Dismantling joints and grooved pipe mechanical couplings can be used to ease gasket assembly.

9.6.6.4.2.4 Field adjustment

All restraints and guides should be installed, and terminal and pump connections should be aligned and tight. The shipping stops should be removed after hydrotesting of the system is complete and just prior to start-up with the pipe full of liquid.

Nozzle loads at pumps and other critical terminal connections should be rechecked to ensure that overloading will not occur. Thermal movements in the piping should be measured and compared to the displacements used in the piping analysis. Spring supports should not bottom out under normal operating loads and temperatures. Pump alignments should be checked to ensure that the attached spring support is indeed keeping the pump nozzles within allowable loads.

Periodic inspection of the spring indicators should be performed to ensure that the loads in the system have not changed over time. Also, the supports themselves should be included in a regular preventative maintenance program.

Proper installation and alignment is crucial to the performance and reliability of both the pump and the expansion joint. Improper installation can result in overloaded pump nozzles and expansion joint fatigue and failure. See Appendices C and D for types and application guidelines.

9.6.6.4.3 Parallel operation

Hydraulic piping design considerations for parallel pump systems are similar to the considerations required for single pump installations, but with three basic differences.

First, the pressures on the inlet (suction) side of the pumps should be equal; likewise the pressures on the outlet (discharge) sides of the pumps should be equal (see Figure 9.6.6.4.3a). There shall not be a significant pressure drop between the inlets (suctions) or outlets (discharges) from the first pump in the line to the last pump. The maximum velocity in the inlet (suction) header shall not exceed 2.4 m/s (8.0 ft/s) unless the liquid is a slurry, in which case it may need to be higher to maintain solids suspension. Keeping the velocity below this value will, for practical purposes, keep the inlet (suction) pressures equal at each pump. Likewise, the velocities in the outlet (discharge) manifold shall be low enough so that there are no significant outlet (discharge) pressure differences. Additionally, the fluid velocity in the suction manifold shall not exceed the velocity in the suction lateral for all operating scenarios. To allow for future expansion, manifolds should be terminated at full diameter with blank flanges.

An alternative design is shown in Figure 9.6.6.4.3b that keeps inlet (suction) and outlet (discharge) manifold velocities identical for each pump. This type of construction may be considered for settling slurry applications when a minimum velocity is required to maintain solids in suspension.

The second consideration is that there should be minimum interaction between the inlet (suction) flow of one pump with another. Figure 9.6.6.4.3c illustrates the minimum recommended spacing and length of pump inlet (suction) pipe, based on the ratio of the take-off diameter to manifold diameter. Take-offs directly opposite each other should not be used. The reason for this prohibition is that the prerotation inherent in the inlet (suction) line of one pump can cause significant turbulence in the inlet (suction) line of the second pump. (Although Figure 9.6.6.4.3c illustrates tee connections produced by welding a piece of pipe directly to the manifold, this type of construction is not ideal. A better alternative is to use a weld saddle, a reducing tee, or a tee and a reducer on the manifold. All of these will produce a more uniform flow pattern and minimize friction loss.)

Third, like single pumps (see Section 9.6.6.5.3), pumps operated in parallel should have isolation valves to allow any individual pump to be taken out of service. Additionally, each pump operated in parallel should have a check valve installed in its outlet (discharge) line, between the pump and the manifold, to prevent backflow under operating conditions. When one of the pumps is to be used as an installed spare, a pressure-relief device may be required between the inlet (suction) block valve and the pump inlet (suction) flange. This would ensure that, in the case of block and check valve leakage on the outlet (discharge) side, a nonoperational pump would not be subjected to full outlet (discharge) manifold line pressure.

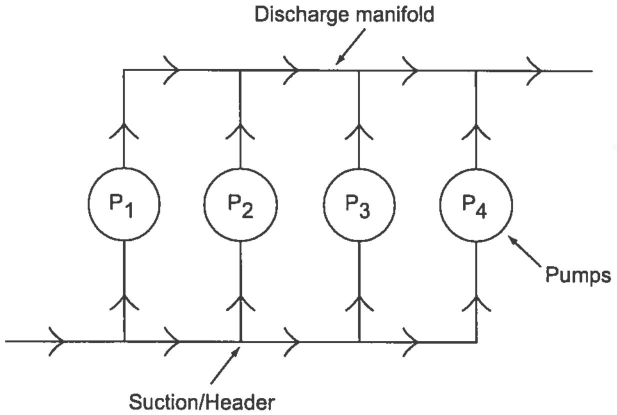


Figure 9.6.6.4.3a — Parallel pump installation

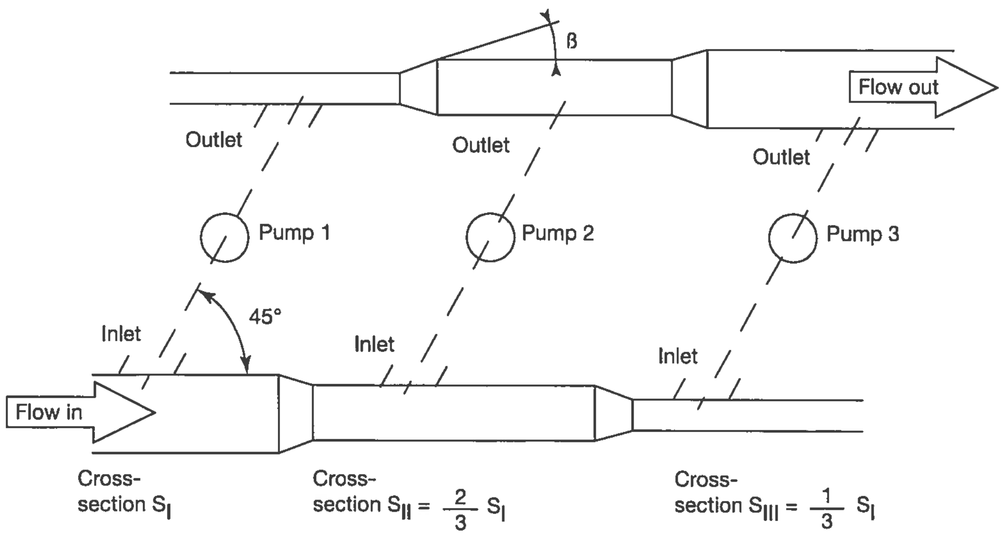


Figure 9.6.6.4.3b — Constant-velocity manifold design, parallel pumps

9.6.6.5 Outlet (discharge) piping requirements

9.6.6.5.1 Pipe size/velocity requirements

The maximum velocity at any point in the outlet (discharge) piping is 4.5 m/s (15 ft/s). This limit shall, however, be reduced if there is a check valve in the outlet piping that will generate a hydraulic shock when it closes. (Note that pump discharge nozzle velocity may exceed this value. Therefore the straight discharge pipe length at the pump discharge nozzle, which will be the same diameter as the discharge nozzle itself, cannot be subject to this velocity limitation.) System pipe friction losses (see Appendix A.1), life-cycle costs, and process considerations normally dictate the size of the discharge piping and fittings. (If the liquid is a slurry, then the velocity in the discharge pipe may be higher in order to keep solids in suspension. See Appendix E.4 for slurry/solids pumping recommendations.)

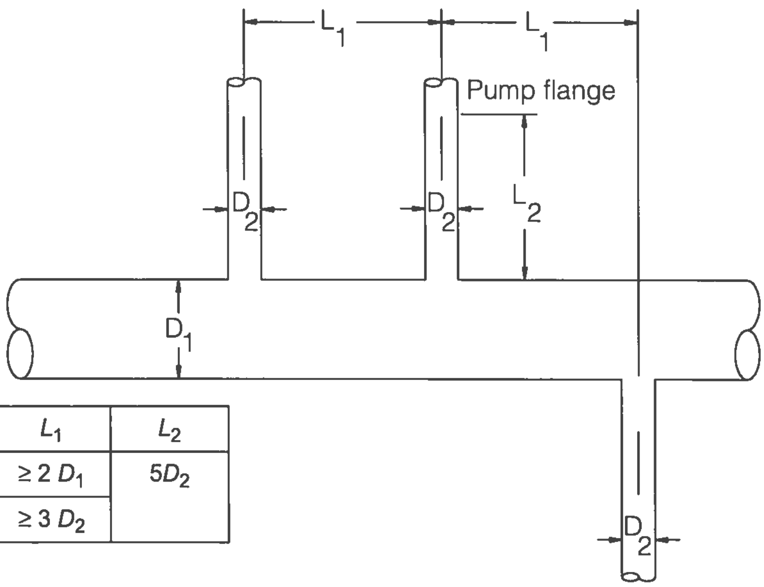
See Appendix E.4 for velocity recommendations for solids/slurries.

9.6.6.5.2 Required straight pipe lengths

Pipe fittings mounted close to the outlet (discharge) flange will normally have minimal effect on the performance or reliability of rotodynamic pumps. However, some pumps can be sensitive to flow-disturbing fittings mounted close to the pump outlet (discharge). This can result in increased noise, vibration, and hydraulic loads. If in doubt, check with the pump manufacturer.

9.6.6.5.3 Recommended valves

For most pumping systems, an inlet (suction) shut-off valve should be installed in the pump inlet piping for system isolation. Likewise an outlet (discharge) shut-off valve should be installed in the pump outlet for system isolation [12]. The shut-off valve is used in priming and when starting or stopping the pump for maintenance. Except for axial flow pumps, where the shut-off horsepower is excessive, it is advisable to close the shut-off valve just before stopping and starting the pump. This is especially important if there is no discharge check valve and the pump is



D_2/D_1	L_1	L_2
≥ 0.3	$\geq 2\ D_1$	$5D_2$
< 0.3	$\geq 3\ D_2$	

Note: The connection from the manifold to the lateral should be designed to minimize flow disturbance. When $D_2 = D_1$, the use of a 90 degree tee is recommended. When $D_2 < D_1$, the use of either a 90 degree reducing tee or a 90 degree tee followed by a reducer is recommended. A fabricated connection should be avoided if possible, since this type of connection tends to cause excessive flow disturbance.

Figure 9.6.6.4.3c — Minimum required suction line lengths and spacing for parallel pumps

operated against a high static head. In addition, for very high horsepower pumps or pumps with thermally sensitive liquids, it is advisable to throttle the pump to minimum flow just before starting or stopping the pump.

If a foot valve is not installed in the inlet (suction) pipe, then a check valve may be necessary between the pump and the shut-off valve to protect the pump from reverse flow and excessive backpressure. If expansion joints are used, they should be placed on the pump side of the check valve to dampen, and not transfer, closure slam. (Do not rely on an expansion joint to dampen the shock of check valve closure. Add a damper to the check valve when shock due to rapid closure of the check valve is likely.) Submersible turbine well pumps must always have a check valve installed in the discharge line, no more than 7.6 m (25 ft) (for water at sea level) above the lowest liquid (pumping) level in the well or sump. This distance limit is necessitated by the vapor pressure limitation to ensure that a fluid void will not form under the check valve, which can cause a hydraulic shock or water hammer (see Section 9.6.6.5.4) the next time the pump is started. At higher elevations, higher temperatures, and for liquids with different vapor pressures, this distance must be evaluated and will be less than the specified 7.6 m (25 ft). The discharge valve and check valve can be replaced by a triple-duty valve (see Figure 9.6.6.5.3) that provides both functions in a single body. Triple-duty valves also provide circuit balance capability and require shorter installed pipe lengths at reduced cost compared to separate valves. (Note that triple-duty valves may have a very high energy cost associated with their use. This should be evaluated as a part of a life-cycle cost analysis.)

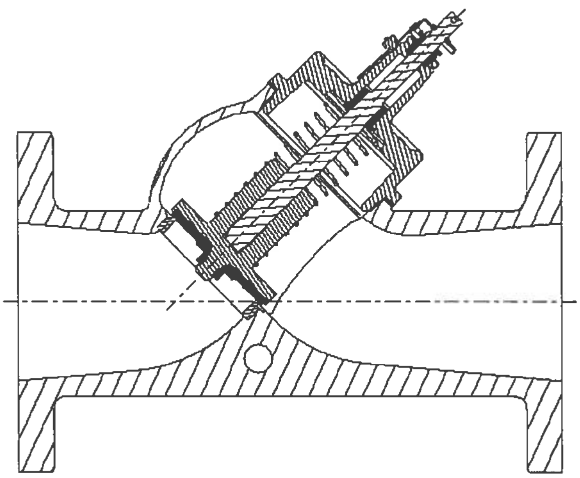


Figure 9.6.6.5.3 — Triple-duty valve

9.6.6.5.4 Water hammer

Water hammer, or hydraulic shock, is a condition that exists when a column of fluid changes velocity quickly. This condition can exist when the power to the driver is lost suddenly, air is finally expelled from the system piping, a valve is closed too quickly, or a check valve closes too slowly and allows backflow to occur before it slams shut. When this occurs, the kinetic energy is rapidly transformed into pressure energy, which will cause a sharp rise above normal system pressure. This transformation produces an acoustic pressure wave that propagates upstream within the pipe. See Appendix B for more detail.

9.6.6.6 References

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2. Committee European for Normalization (CEN). *Rotodynamic Pumps – Design of Pumps Intakes – Recommendations for Installation of Pumps* (CR 13930).

3. Committee European for Normalization (CEN). *Rotodynamic Pumps – Recommendations for Fitting of Inlet and Outlet on Piping* (CR 13932).

4. Hydraulic Institute. ANSI/HI 9.6.2 *Centrifugal and Vertical Pumps for Allowable Nozzle Loads.*

5. Committee European for Normalization. *Rotodynamic Pumps – Forces and Moments on Flanges* (CR 13931).

6. International Organization for Standardization. ISO 9905, *Technical Specification for Centrifugal Pumps, Class I.*

7. International Organization for Standardization. ISO 5199, *Technical Specifications for Centrifugal Pumps, Class II.*

8. American Petroleum Institute. API-610 – *Centrifugal Pumps for Petroleum, Heavy Duty Chemical, and Gas Industry Services.*

9. Process Industry Practices (PIP). *Allowable Piping Loads on Rotating Machinery Nozzles* (PIP RESE002).

10. American Society of Mechanical Engineers (ASME). *Code for Pressure Piping B31*. New York: ASME.

11. Process Industry Practices. *Recommended Practices for Machinery Installation and Installation Design* (API Recommended Practice 686/PIP REIE 686).

12. Process Industry Practices. *Typical Piping Arrangements for Centrifugal Pumps* (PIP REEP004).

13. Europump and Hydraulic Institute. *Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems.*

14. Hydraulic Institute. *Engineering Data Book.*

15. Hydraulic Institute. ANSI/HI 1.3 *Rotodynamic (Centrifugal) Pumps for Design and Application.*

16. *Mark's Standard Handbook for Mechanical Engineers*, 8th ed. New York: McGraw Hill.

17. Hydraulic Institute. ANSI/HI 9.6.3 *Allowable Operating Region.*

18. Mechanical Contractors Association. *Guidelines for Quality Piping Installation.* MCF.

9.6.6.7 Sources of additional information

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James R. Payne and Vincent A. Carucci. *Guidelines for the Design and Installation of Pump Piping Systems.* PVRC Project, January 1993.

National Valve and Manufacturing Company. *NAVCO Piping Datalog.* Pittsburgh: NAVCO, 1974. p.130-135.

Michael Bussler and Tony Paulin. *Pipe Design for Robust Systems, Chemical Engineering.* June 1997. p. 84.

Crocker and King. *Piping Handbook.* New York: McGraw Hill, 1973.

9.6.6.8 List of acronyms

ANSI - American National Standards Institute, www.ANSI.org

API - American Petroleum Institute, www.API.org

ASME - American Society of Mechanical Engineers, www.ASME.org

AWWA - American Water Works Association, www.AWWA.org

HI Pump Piping for Rotodynamic Pumps — 2009

CEN - Committee European of Normalization (the European Committee for Standardization), www.CEN.eu

HI - Hydraulic Institute, www.Pumps.org

ISO - The International Organization for Standardization, www.ISO.org

NPSH - Net positive suction head (in absolute head terms, above vapor pressure)

NPSHA - Net positive suction head available (available to pump)

NPSHR - Net positive suction head required (required at 3% head drop)

PIP - Process Industry Practices, www.PIP.org

psi - pounds per square inch

PVC - polyvinyl chloride

VIL - vertical in-line (vertical pump mounted in pipeline)

VTP - vertical turbine pump (well-type pump)

Appendix A

System Curves

This appendix, which is not a part of this standard, discusses a number of highly technical items and calculation methods that relate to the application of pumps in piping systems. The purpose of this appendix is to provide the reader with the knowledge that there are various analytical methods, hardware, components, and references for pumping systems that can help the specifier and user minimize problems and maximize efficiency.

It is not the intent for this appendix to be a tutorial of how to analyze pump and piping systems. These subjects cannot be covered in adequate detail in a document of this size. The calculations and procedures shown herein are meant to be illustrative only. The reader needs to refer to the many published textbooks, standards, and guidelines that properly cover these subjects in the depth needed for proper analysis and evaluation.

A.1 Calculation of the system curve

When connected to a piping system, a rotodynamic pump will operate at a duty point of equilibrium between the pump and the piping system. At the point of equilibrium, the total head developed by the pump equals the required static head plus head loss in the piping system. The performance of the pump in this respect is usually represented in a diagram format by the so-called *pump H-Q curve*. The corresponding curve for the piping system is called the *system curve*. The volume flow at which the pump and system curves intersect each other is the flow that will pass through the piping system. To correctly determine the sizes of the pump and piping system, knowledge of the characteristics of their respective curves is necessary.

System curves can be very simple, such as when the pump is connected to a single piping system with no control valves and a fixed static head on both the suction and discharge side. On the other hand they can be extremely complex, such as when multiple pumps are connected to a system having several control valves, numerous branches, and varying static heads. In the first instance, the system curve can be calculated by taking into account the static heads and figuring the friction losses within the piping system. In the second case, the static heads and friction losses will vary and a branch analysis must be performed. In practice the actual situation will be somewhere between the two extremes.

Determination of the requirements (head and capacity) for pump performance is at the heart of developing the requirements for any pumping installation. The operating head imposed on the pump is composed of two principal components, the fixed, or static, head that varies over a small, well-defined range and the dynamic losses that vary with the fluid velocity in the piping system. Other factors that will affect these values include fluid viscosity and specific gravity. These losses will vary by the number of pumps in operation and the pumps' speed of operation if variable speed drives are provided. The value associated with the dynamic losses in a pumping system (commonly called "friction") can be approximated by the use of empirical formulae developed from experimental results conducted by researchers in the last century. Very little improvement in calculation methods have been developed in the last 50 years and the methodologies currently in use rely on limited information. Each method requires the estimation of the roughness of the pipe or conduit wall and the flow regime in the pipeline. Calculation of loss values through fittings and valves suffer from the same level of uncertainty because the absolute value will depend a great deal on the nature of the turbulence in the stream approaching a fitting or valve; losses after a reach of straight pipe will be less than losses in a fitting downstream from a valve or fitting. It can be appreciated, therefore, that any calculation of dynamic losses must be viewed as an approximation and that the actual conditions affecting dynamic losses will never be known specifically. It follows, then, that the requirements to be used for selecting pumping equipment must be based on a combination of assumptions that define the most optimistic and most pessimistic estimates for pipe roughness and fitting and valve losses. Designers should be aware that overconservatism in the evaluation of hydraulic losses has been found to be the root cause of many poorly designed pumping systems. A more detailed discussion of the uncertainty of the calculations and the shortcomings and advantages of various methods may be found in Chapter 3, *Pumping Station Design,* 3rd ed., Jones, et al, Elsevier Butterworth-

Heinemann, 2006. Because of the uncertainties as described above, whenever possible the system curve should be measured rather than calculated.

The following paragraphs provide a discussion of factors influencing head losses and commonly used methods for calculating fluid flow losses in piping systems.

A.1.1 Factors that affect head loss

The calculation of friction loss is not an exact science and may lead to errors in sizing the pump. The range of friction factors should be carefully evaluated before making a decision as to what to use.

a) Flow rate: When the flow rate increases, the flow velocity increases and so does the friction or resistance to flow caused by viscosity. The head loss is approximately related to the square of the velocity.

b) Pipe inside diameter: When the inside diameter is made larger, the flow area increases and the liquid velocity for a given rate of flow is reduced. When the liquid velocity is reduced, there is lower head loss in the pipe. Conversely, if the pipe diameter is reduced, the flow area decreases, the liquid velocity increases, and the head loss increases.

c) Roughness of pipe wall: As the roughness of the inside of a pipe increases, so does the thickness of the slow or nonmoving boundary layer. The resultant reduction in flow area increases the liquid velocity and along with it the head loss in the pipe.

d) Corrosion and scale deposits: Scale buildup and corrosion both serve to increase the pipe roughness, which will increase the head loss in the pipe. Scale buildup has the added disadvantage of reducing the inside diameter of the pipe.

e) Liquid viscosity: The higher the viscosity, the higher the friction between two moving streams of liquid. It requires more energy to move a high viscosity liquid than it does for a low viscosity liquid.

f) Length of pipe: The head loss occurs all along a pipe. It will be constant for each foot of pipe and must be multiplied by the number of feet in the pipe. The head loss in pipe is listed in a number of tables per 100 meters (m) or 100 feet (ft) of pipe.

g) Fittings: There are two basic methods to predict the head loss in valves and fittings. One method assigns a K factor to each type or size of fitting, which is applied to the velocity head ($KV^2/2g$) to calculate the loss. Fittings such as elbows, tees, valves, strainers, etc. have all been tested and assigned K factors based on the velocity head and head loss measured through them. The other method assigns an equivalent length of straight pipe to each type or size of fitting. There are charts and tables in various handbooks, including the Hydraulic Institute *Engineering Data Book* [14] that provide these values. The *Engineering Data Book* defines the procedure and lists the data necessary for predicting lead loss for viscous liquids and paper stock.

A.1.2 Friction loss calculations

There were a number of experimental studies conducted many years ago designed to easily predict the head loss in pipes and fittings. These have been published as both formulas and tables for different size pipes, and different flow rates and fittings. The two methods most used are by Darcy and Weisbach, and by Hazen-Williams. The Hazen-Williams relationship, however, applies to a narrow range of pipe sizes.

The following section illustrates the basic mathematics behind each of these two methods. It should be noted that these two methods are not acceptable for use with non-Newtonian fluids. For non-Newtonian fluids, such as paper stock or slurries, specialty algorithms are available from various sources for calculating friction losses.

A.2 Pipe head loss calculation methods

A.2.1 Darcy-Weisbach equation

The Darcy-Weisbach tables, which are similar to the pipe head loss tables in the HI *Engineering Data Book* [14], are based on the head loss in clean steel new pipe. Any fouling factors applicable will have to be determined by the user and applied accordingly. It uses the following basic formula:

$$h_f = (f) \times \frac{L}{D} \times \frac{V^2}{2g}$$

Where:

h_f = Total head loss – m (ft)

f = Friction factor related to the pipe roughness

L = Length of pipe – m (ft)

D = Diameter of pipe – m (ft)

V = Average liquid velocity in the pipe – m/s (ft/s)

g = Acceleration due to gravity – 9.81 m/s^2 (32.2 ft/s^2)

It should be noted that the friction factor (f) for the Darcy-Weisbach method is derived from the "Moody Diagram," which uses the C.F. Colebrook equation to transition from laminar flow to turbulence zones.

A.2.2 Hazen-Williams equation

The Hazen-Williams equations take a different approach. They are based on the head loss in 10-year-old pipe and on actual testing of various size pipes of different degrees of roughness. These tests were conducted in the first part of the last century and were limited to a relatively narrow range of pipe diameters. The tables should not be used for pipe diameters less than 200 mm (8 in) nor for diameters greater than 915 mm (36 in). Their values must be adjusted for pipe age and materials, through the use of correction factors. The Hazen-Williams calculations are strictly designed for water at ambient temperatures and are not applicable to other liquids. The tables are based on the following formula:

(metric units)

$$h_{f\,100} = 4.35 \times 10^{17} \times \frac{Q^{1.85}}{(C^{1.85} \times D^{4.87})}$$

Where:

h_{f100} = Head loss, in m per 100 m of pipe

C = Correction factor to account for pipe roughness

Q = Liquid flow rate – m^3/s

D = Inside pipe diameter – mm

(US customary units)

$$h_{f\ 100} = 0.2083 \times \left(\frac{100}{C}\right)^{1.85} \times \frac{Q^{1.85}}{D^{4.87}}$$

Where:

$h_{f\,100}$ = Head loss, in ft per 100 ft of pipe

C = Correction factor to account for pipe roughness. (Note that C values are the same for both metric and US customary units.)

Q = Liquid flow rate – gpm

D = Inside pipe diameter – in.

NOTE: Correction factor values of C range from 55 for small diameter, 40-year-old, cast-iron pipe, to 140 for large size, steel form, concrete pipe; with a mean value being about 120 for concrete pipe.

Appendix B

Water Hammer

This appendix, which is not a part of this standard, discusses a number of highly technical items and calculation methods that relate to the application of pumps in piping systems. The purpose of this appendix is to provide the reader with the knowledge that there are various analytical methods, hardware, components, and references for pumping systems that can help the specifier and user minimize problems and maximize efficiency.

It is not the intent for this appendix to be a tutorial of how to analyze pump and piping systems. These subjects cannot be covered in adequate detail in a document of this size. The calculations and procedures shown herein are meant to be illustrative only. The reader needs to refer to the many published textbooks, standards, and guidelines that properly cover these subjects in the depth needed for proper analysis and evaluation.

When pumps start or stop or the momentum of the fluid in the piping is changed by any other means (like opening or closing a valve rapidly), a pressure wave is formed. The pressure wave can be either above or below the normal pressure in the piping with the pumps operating depending on the configuration of the piping and pump system. These pressure waves are called "surge pressures" or "water hammer" (when water is the fluid) and their magnitude can be sufficient to burst or collapse piping, valves, machinery casings, and other devices.

The magnitude of the pressure wave can be calculated with reasonable precision if you know the configuration of the piping, the size of the pipes, the materials of the piping, the properties of the fluid and how quickly the pump and/or fluid accelerates or decelerates. When the column of fluid in the piping is either started or stopped, the energy of the system is transformed from velocity energy to head or pressure energy. (Because the fluid and piping material are not completely incompressible, they will absorb a fraction of the energy.)

It is not the intent of this material to provide detailed means to analyze the magnitude and duration of surge waves. This analysis is covered in many hydraulics texts and can be performed with available computer software. This material is provided to inform designers and users of pump piping systems that a surge analysis is necessary because surge will occur in every pumping system. In most cases, the peak pressure of this surge may be below the pressure that may cause damage, but one cannot determine this solely by any simple rules.

However, there is one rule of thumb that does apply. The maximum surge pressure will occur whenever the fluid is stopped in less time than it takes for a pressure wave to travel from the equipment that stopped the flow to the other end of the piping system and back. This may be better illustrated by an example. If a piping system is 2682 m (8800 ft) long and the fluid is water, the highest surge pressure would occur if the water is stopped faster than it takes the pressure wave to travel 2682 m (8800 ft) and then reflect back to the source, a total of 5364 m (17,600 ft). A pressure wave in water travels at the velocity of sound in the fluid. For water in most piping materials, the velocity of sound is about 1341 m (4400 ft) per second so a pressure wave in 2682 m (8800 ft) of pipe will take about 2 seconds. If a valve at the end of the pipe is closed in less than 4 seconds, the pressure of the surge wave will be maximized. The calculation of this value is discussed in Appendix B.1.1.

B.1 Water hammer

Transient forces such as water hammer can produce large displacements in the piping system, which can cause severe damage to the piping system and any attached equipment, and should be evaluated.

B.1.1 Momentum

Water has mass. One cubic meter of water at 15 °C (60 °F) weighs 1000 kg (1 ft^3 of water weighs 62.4 lb or 1 gal weighs 8.3 lb, at sea level). Moving water has momentum, which is directly related to both the mass and the velocity of the liquid.

Liquid momentum = mass of the liquid in the piping system × velocity of the liquid

The faster the liquid is flowing, the greater its momentum. The greater the momentum, the more damage that can occur due to water hammer if the liquid is suddenly stopped.

This sudden change in momentum creates a force (Newton's second law of motion) as follows:

$$F_m = \rho A L \frac{(V_1 - V_2)}{(t_1 - t_2)}$$

Where:

F_m = Momentum force – N (lb_f)

ρ = Fluid density – kg/m^3 (lb_m/ft^3)

g = Acceleration due to gravity – 9.81 m/s^2 (32.2 ft/s^2)

A = Cross-sectional area of pipe – m^2 (ft^2)

L = Total length of pipe – m (ft)

V_1 = Initial velocity – m/s (ft/s)

V_2 = Final velocity – m/s (ft/s)

t_1 = Initial time – s

t_2 = Final time – s

For weight density, the metric units should be kN/m^3 and the equation remains the same.

However, the above equation assumes rigid pipe walls and incompressible liquid, and only applies to relatively slow valve closures [about $(t_2 - t_1) > L/305$, where L is in meters; or $(t_2 - t_1) > L/1000$, where L is in feet].

B.1.2 Acoustic shock wave

When the flow of liquid is suddenly stopped, the liquid tries to continue in the same direction. In the area where the velocity change occurs, the liquid pressure increases dramatically due to the momentum force. As it rebounds, it increases the pressure along the pipe near it and so on and a shock wave or acoustic pressure wave is formed. This is analogous to the waves on a pond after a stone is dropped in. The waves radiate outward in all directions. As the wave travels further from the center, its energy is spread over a larger and larger area and it will dissipate and eventually die out. In a pipe the acoustic pressure wave can only travel along the pipe. It will travel at the speed of sound in water, or if we assume very rigid pipe and ambient temperature, at roughly 1341 m/s (4400 ft/s) (see Appendix F.1). Because it is contained, it cannot dissipate its energy very quickly. As with the wave in the pond that will reflect when it encounters an obstruction, so will the shock wave.

The same series of events that caused the pressure wave in the first place will occur every time the wave encounters an obstruction. This obstruction could be another valve, a fitting, or the pump. If nothing breaks, as with the ripples in a pond, the pressure wave will strike and reflect off any rigid surface. It will continue to travel up and down the pipe until it eventually dissipates its energy through stressing parts and piping. When the pressure wave reaches a component that will not bend to resist it, two things can happen. The pressure wave will be reflected or the part will break.

B.1.3 Magnitude

The potential magnitude of the pressure wave can be tremendous [16]. Based on elastic column theory, the formula for the maximum pressure rise for rapid valve closure (prior to the pressure wave making a round trip back to the valve) and no friction loss is as follows:

(metric units)

$$P = \rho \times a \times V$$

(US customary units)

$$P = \rho \times a \times \left(\frac{V}{144}\right)$$

Where:

P = Pressure rise – Pa (psi)

ρ = Liquid density – kg/m^3 (lb$_m$/ft^3)

a = Velocity of sound in water – m/s (ft/s)

V = Velocity of the liquid in the pipe – m/s (ft/s)

g = Acceleration due to gravity – 9.81 m/s^2 (32.2 ft/s^2)

For weight density, the metric units should be kN/m^3 and the equation remains the same.

Table B.1 shows the increase in pressure at the crest of the shock wave. This is the increase in pressure over the pressure already in the pump casing. As an example, assume a pump is working at 345 kPa (50 psi) and the liquid is traveling 1.5 m/s (5 ft/s). The instantaneous pressure inside the casing will jump to 2503 kPa (368 psi), or 345 kPa (50 psi) static + 2158 kPa (318 psi) as indicated in Table B.1 at 1.5 m/s (5 ft/s). At a liquid velocity of 2.4 m/s (8 ft/s), the instantaneous pressure will hit 3797 kPa (558 psi). Pump casings are not usually designed for this magnitude of pressure increase.

This equation ignores any dampening effect of the fluid compressibility and the piping material elasticity and so this equation slightly overpredicts the maximum surge pressure.

The initial pressure wave will propagate from the source of the wave towards the other end of (and up any branches of) the piping system. Depending on the configuration at the other end, a pressure wave will reflect from the end back to the original source. Because of dampening in the system, the magnitude of this reflected wave is usually smaller than the initial pressure wave. The reflected wave will travel back to the source and be reflected back again. This reflection of the pressure waves from each will continue with the magnitude decaying each time until the pressure reaches steady state. The time for this decay depends on the initial pressure magnitude, the length of the piping system, and fluid and piping properties. In water systems with steel piping, there may be seven or more complete cycles of the reflected waves even with pressure-dampening surge control vessels on the pump discharge.

B.1.4 Damage

Cast iron is a rather brittle material and in most cases will not be able to withstand the sudden impact of a pressure wave well above its normal internal pressure, traveling at the speed of sound. The face of the pump casing is the largest open area of the pump. Therefore, it will experience the highest forces due the pressure change. In a typical

Table B.1 — Magnitude of pressure wave

Velocity	Delta pressure	Velocity	Delta pressure
m/s	kPa	ft/s	psi
0.3	432	1	64
0.6	863	2	127
0.9	1295	3	191
1.2	1726	4	254
1.5	2158	5	318
1.8	2589	6	381
2.1	3021	7	445
2.4	3452	8	508
2.7	3884	9	572
3.0	4316	10	635
3.4	4891	11	699
3.7	5323	12	762
4.0	5754	13	826
4.3	6186	14	889
4.6	6617	15	953

volute design, the tongue or cutwater is placed relatively close to the impeller at a given point to direct the flow leaving the impeller as efficiently as possible. This also makes the casing very rigid in this location. The face of the casing tries to bulge outward but is held firmly by the tongue. This causes extremely high stresses in this area and is usually the point where the failure begins. Experience has shown that strengthening this area has little or no effect on the casing's ability to survive water hammer.

Other materials that are more ductile may absorb the shock waves without cracking, but still risk permanent deformation and ultimate failure. The pump is not the only component that is affected by this phenomenon. Valves, sprinkler heads, and pipe fittings are also at risk of catastrophic damage. Pipe hangers and pump foundations can also be adversely affected by water hammer. Polyvinyl chloride (PVC) pipe and fittings are very susceptible to damage from water hammer.

Water hammer can be controlled through proper valve closure rates (with slow closing valves), the addition of diaphragm tanks to absorb the pressure surge, and relief valves to release the pressure.

Appendix C

Selecting and Locating Pipe Supports and Restraints

This appendix, which is not a part of this standard, discusses a number of highly technical items and calculation methods that relate to the application of pumps in piping systems. The purpose of this appendix is to provide the reader with the knowledge that there are various analytical methods, hardware, components, and references for pumping systems that can help the specifier and user minimize problems and maximize efficiency.

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This appendix is meant to give the user only a brief overview of the subject of pipe supports. There are a number of books and articles that go into this subject in considerable depth. There are also many organizations and individuals who specialize in this area who can assist the user when designing the system if required.

Selecting and locating pipe supports and restraints in a piping system, particularly when connecting to rotating equipment, such as pumps, turbines, or compressors, can be a difficult task. There are many factors that the designer must take into consideration [10]. These include system flexibility, which will change as supports and restraints are added; nozzle load limits of attached equipment; and the general layout of the structure by which the piping must be supported. All these combine to create a dynamic problem that is difficult to generalize. Within the scope of pipe support around pumps and other rotating equipment, the following design guidelines are recommended:

C.1 Restraint and stops

Restraints should be placed in a position that will force the pipe to grow in a direction away from the pump nozzle and into the piping system, where the displacement and associated expansion load can be absorbed.

Stops and restraints placed near pumps should correspond to planes of zero growth, relative to the pump. The restraint should limit pipe forces acting on the pump.

C.2 Pipe supports for vertical loads

Pipe supports for vertical loads (Figure C.1) are designed specifically to carry the weight of the pipe, along with associated vertical loads, such as insulation, process liquid, fittings, valves, etc. The supports will not restrict the thermal expansion of the piping in any direction. Rigid resting-type supports are widely used for horizontal piping runs. The support is affixed to the pipe, providing vertical support while allowing the pipe to slide laterally along the structure. These supports prove to be satisfactory in most applications where the friction load due to the weight is relatively small compared to the thermal expansion forces.

When adequate overhead space and structure is present, solid hanger supports (Figure C.2) should be used. The solid hanger will carry the vertical load of the pipe, while eliminating the frictional effects of the resting-type fixtures. In piping systems that exhibit considerable vertical movement due to thermal effects, spring-type supports should be used. Hangers should be sized so that there are minimal vertical loads on the equipment nozzles in the ambient condition.

Constant-effort spring supports (Figure C.3), when placed properly, can alleviate many problems faced when connecting to rotating equipment. Using a constant-effort device, which will support the vertical piping load exactly in hot and ambient conditions, can greatly reduce loads carried by equipment nozzles.

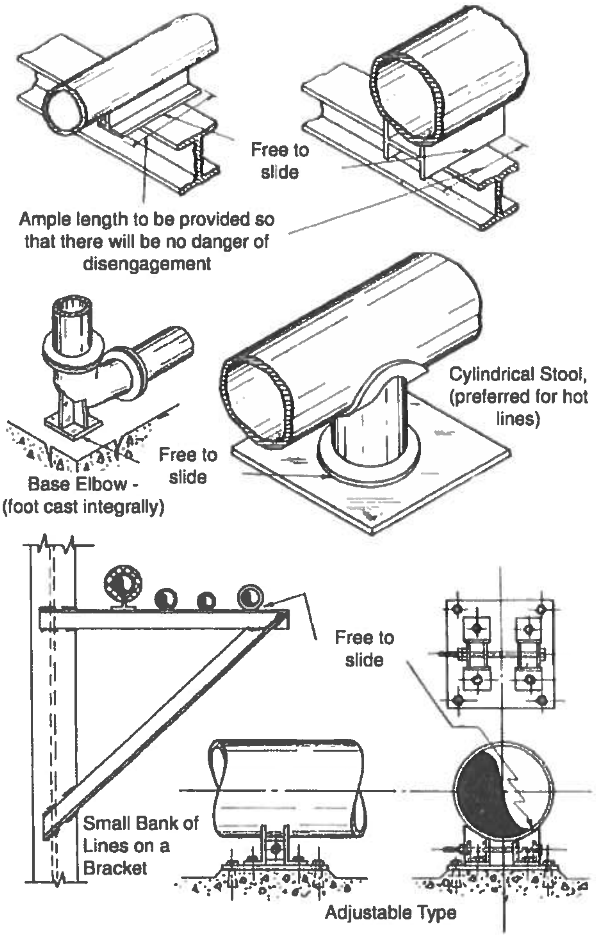

Figure C.1 — Pipe supports for vertical loads

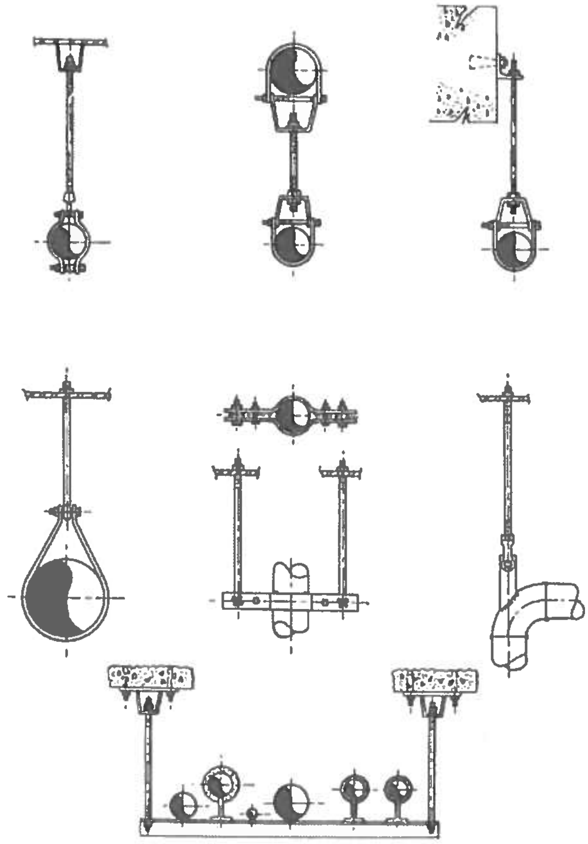

Figure C.2 — Solid pipe hanger supports

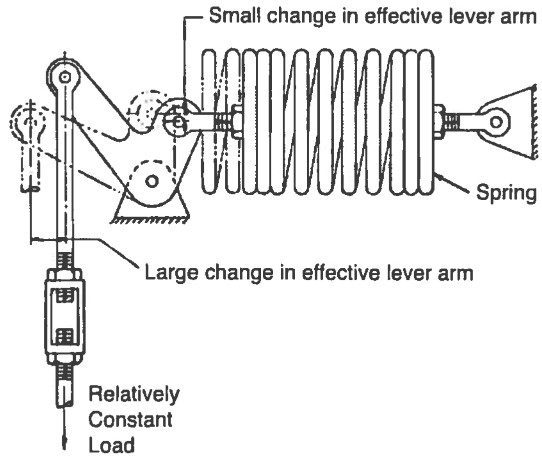

Figure C.3 — Constant-effort spring supports

C.3 Guides and restraints

Pipe restraints are used to control the direction in which a pipe system is permitted to move, due to thermal expansion. The restraint is designed to limit the movement of the pipe in the axial direction and provide no resistance to bending or rotation. Restraints can be beneficial when connecting to rotating equipment by directing pipe displacements away from the equipment connections and into more flexible areas of the pipe system that have the ability to absorb the movement. Tie rods and jointed struts can be used on horizontal or vertical pipe runs, or in transition areas at pipe elbows or vertical drops that require the movement of the pipe to be restricted in the horizontal direction.

Pipe guides (Figure C.4) are used to restrict the effects of rotation and/or bending on the pipe, while allowing unrestricted axial movement of the pipe.

C.4 True anchors

True anchors are seldom used as fixtures, which combine the characteristics of the support and restraint devices, to limit pipe movement in the three linear planes and the rotation about the three axes. They can be used to isolate vibration and displacements from different segments of the system. The devices can also be used to raise the natural frequency of the system to reduce amplitude and avoid resonance.

C.5 Spring supports

Spring supports (Figure C.5) are an important part of most pump piping systems. They are designed to support the weight load of the pipe, process liquid, and associated attached equipment, such as insulation, valves, and fittings, while at the same time absorbing the effects of thermal expansion on the pipe. Spring supports are often placed directly above or below a pump nozzle connection. The following guidelines are provided for the proper installation of spring supports:

a) Spring support height should be adjusted to allow adequate flange alignment and seating at the pump nozzle. Proper alignment should be accomplished without adjusting the spring load or removing the shipping stops from the support.

b) With the piping empty, the shipping stops will prevent the spring load from deflecting the pipe, which can cause misalignment with pumps, terminal connections, and other support fixtures. If the flange connections are made before removing the stops, the spring will exert a force at least equal to the weight of the fluid that it was designed to support. The stops should remain in place until after the hydrotest of the piping system is complete and just prior to system start-up.

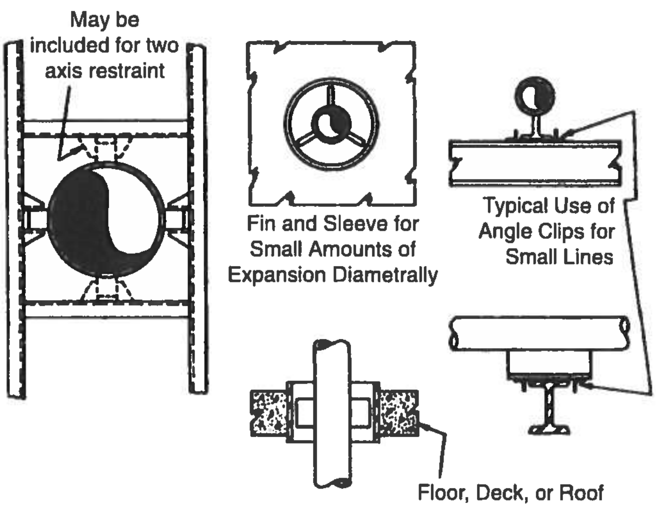

Figure C.4 — Pipe guides

c) When spring supports are used at pump attachments, they should be placed directly above, below, or adjacent to the pump nozzle.

d) Supports should be placed on the attached piping to achieve a near dead zero load on the pump nozzle. Allowances must be made, however, for angular offsets at hanger supports that are necessary when the piping is in an ambient condition.

C.6 Friction from supports

Friction should not be ignored when determining the overall flexibility of the piping in systems that connect to pumps or other rotating equipment. The analysis should be performed with and without friction included, determining the worst-case situation with regard to piping stresses and nozzle loads.

Two common methods for lowering the frictional effects within the system are low-friction bearing plates and rollers.

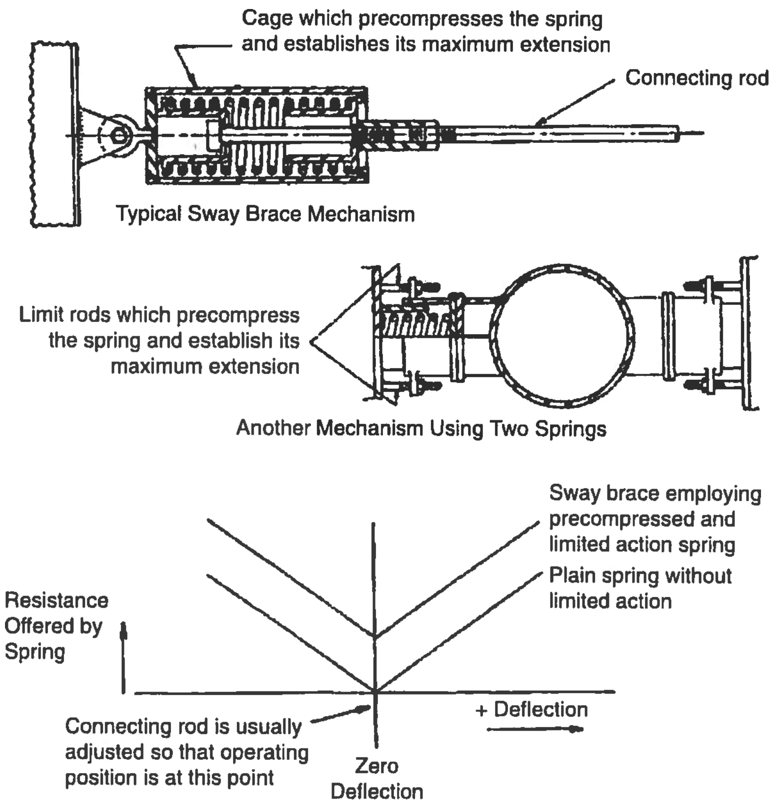

Figure C.5 — Spring supports

Appendix D

Expansion Joints and Couplings

This appendix, which is not a part of this standard, discusses a number of highly technical items and calculation methods that relate to the application of pumps in piping systems. The purpose of this appendix is to provide the reader with the knowledge that there are various analytical methods, hardware, components, and references for pumping systems that can help the specifier and user minimize problems and maximize efficiency.

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This appendix is meant to give the user only a brief overview of the subject of expansion joints and couplings. There are a number of books and articles that go into this subject in considerable depth. There are also many organizations and individuals who specialize in this area who can assist the user when designing the system if required.

D.1 Expansion joint types

Slip (axial) joints or packed joints are common in smaller-diameter applications, where misalignments and nozzle loads are not as critical. The slip joint is essentially a pair of telescoping cylinders, one inside the other. A compression sleeve is mounted at the end of the outer cylinder. Sealing between the cylinders is accomplished using either packing or an elastomer O-ring. The design is simple and reliable, but its use is limited to applications where pipe movements are in the axial direction (Figure D.1).

Elastomer joints (Figure D.2) are commonly used for low-pressure applications. Their use is widespread in hotels, institutions, and large office buildings; on condenser and circulating water lines; and as an isolation device guarding against transmission of vibrations due to pump vibration and pressure pulsations. Rubber bellows joints, due to their flexibility, provide protection from slight misalignments in piping and pump nozzle connections. When a rubber bellows is used on the inlet (suction) connection of a pump, consideration should be given to filling the internal convolutions with a softer material to minimize flow disturbance to the pump.

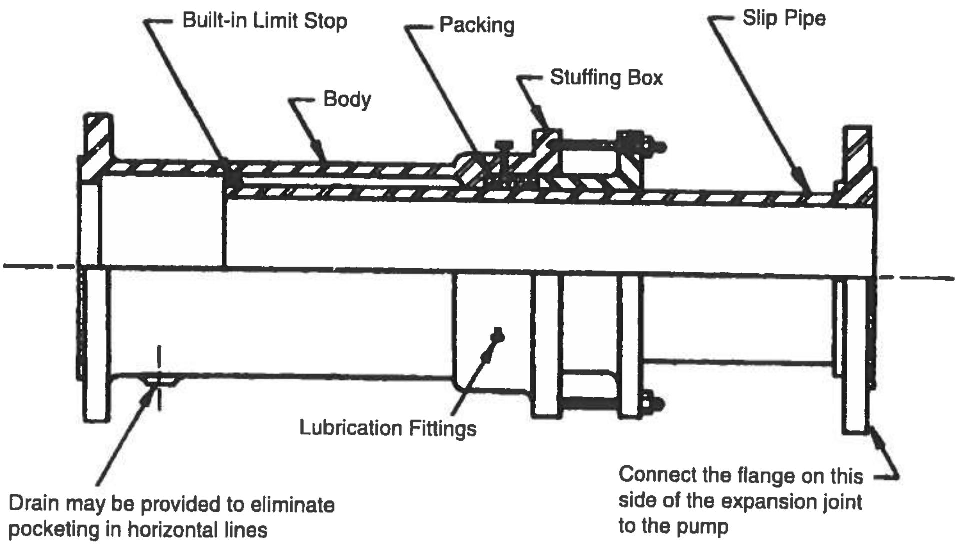

Figure D.1 — Slip/packing expansion joint

Metal bellows expansion joints are typically used for high-pressure, high-temperature applications, where nonmetallic components are not suitable (see Figure D.3).

These joints come in a wide range of standard designs and can be custom built to serve a specific task. Bellows joints are generally more versatile than packed slip joints because they combine axial movement, angular rotation cocking, and a certain amount of lateral movement offset. For increased lateral movement, two expansion joints are joined by a nipple and limit rods. These are known as *universal* or *double expansion joints*.

Caution must be used when including expansion joints in the piping design. As stated previously, the joint can be considered the weak point of the system. The piping on both sides of the expansion joint must be self-supported to operate as intended; the joint cannot withstand the weight load of the piping system. Proper pipe support fixtures must be chosen (see Section 9.6.6.4.2) to allow for the free axial movement of the joint. In pump applications where expansion joints are used in vertical pump lines, spring hangers may be required to support the weight of the piping directly over the pump nozzle. Some industries do not allow the use of flexible connections on pump suction and discharge flanges. In those cases, the flexibility to accommodate thermal growth must be built into the piping. This requires pipe strain analysis to avoid overloading either the pump or the piping.

High vibration, cyclic flow, or pump pulsations can quickly fatigue metallic bellows causing failure of the joint. External sleeves are often used to protect the fragile bellows and to provide protection for personnel and equipment should the joint fail. Internal sleeves will reduce the pressure drop across the joint and protect the bellows from flow erosion and turbulence (see Figure D.3).

Once installed, the expansion joint will require periodic inspection of the bellows assembly for corrosion and damage, but little maintenance is required to keep an expansion joint operating. If the joint is found to be defective or damaged, it should be replaced.

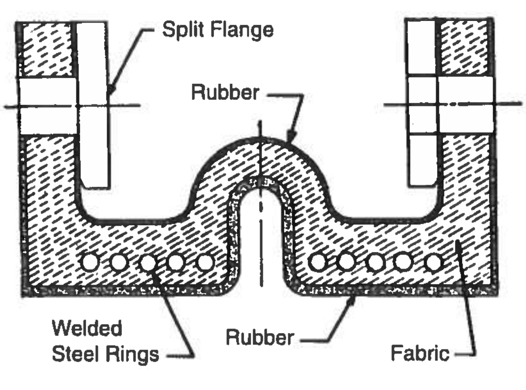

Figure D.2 — Rubber expansion joints

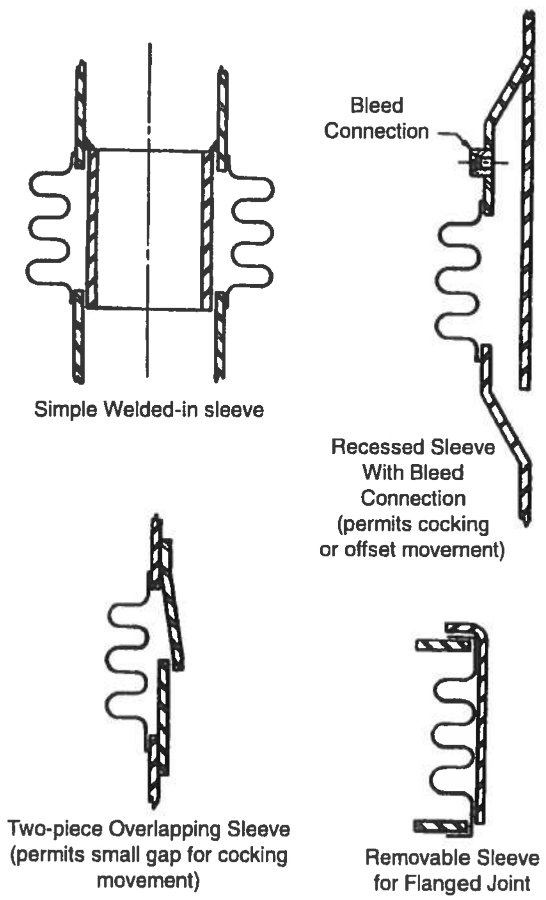

Figure D.3 — Metal bellows expansion joint

D.2 Expansion joint application

Consideration must be given to pressure effects when expansion joints are used in pump applications. When an expansion joint is mounted close to a pump nozzle, the pressure exerted by the liquid on the joint will generate a force equal to the pressure across the inside surface area of the expansion joint, times that projected area. Ideally the pump end of the bellows is anchored or restrained to absorb this force, and prevent it from acting on the pump. If the attached piping is, instead, anchored on the other side of the bellows, this force will be directed back towards the pump nozzle, where possible overloading can take place. Self-equalized expansion joints must then be used, which are equipped with tie rods on the exterior of the bellows, between the mounting flanges (Figure D.4).

Careful attention should be given to the design of the tie rods, also known as *control rods* or *thrust rods*, in piping systems capable of inducing a hydraulic pressure reaction force on the pump (most systems).

In such systems, tie rods (if properly designed and adjusted) will restrain the axial internal pressure force sufficiently so as to prevent an excessive reaction force on the pump. If an adequate design is not used, then the magnitude of the reaction force on the pump can easily exceed the pump's nozzle load capability and have harmful effects on the pump.

Standard tie rod designs having maximum allowable stress as the design criterion and that do not consider the axial deflection, or stretch, of the rods due to the pressure in the pipe, are inadequate for use near pumps without evaluating the axial deflection of the resulting design and its effect on the pump.

Designs based on a high allowable stress and that use high strength tie rods are particularly harmful in that deflection is proportional to stress and there is a high tie rod deflection corresponding to the high allowable stress.

The following requirements shall apply:

a) A piping system having expansion joints shall use tie rods to restrain the force caused by the pipe internal pressure.

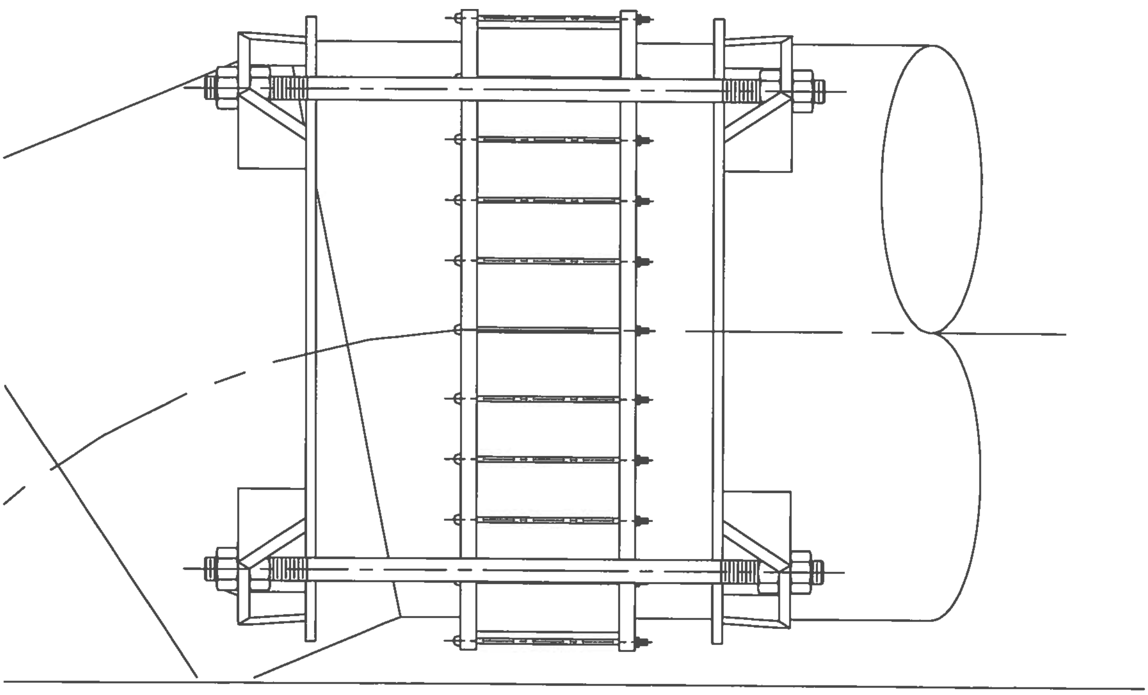

Figure D.4 — Expansion joint bolting arrangement

b) A piping system having expansion joints and tie rods in the vicinity of a pump shall be evaluated to determine if it is capable of inducing a hydraulic pressure reaction force on the pump. Most systems will induce such a force because the piping cannot freely stretch away from the pump when subjected to pressure.

c) If the piping system is capable of inducing a hydraulic pressure reaction force on the pump, the tie rod design used in the vicinity of the pump shall be evaluated for axial stiffness.

d) If the total axial stiffness (spring constant) of the tie rods, including their supporting brackets, is not equal to that of the pipe, then the stretch of the tie rods shall be evaluated. The amount that the stretch of the tie rods (including their supporting brackets) is above that of an equal length of pipe under maximum system working pressure shall not exceed 0.250 mm (0.010 in).

e) In addition, the maximum allowable nozzle loads of the pump shall not be exceeded.

f) The weight of the pipe and contents shall be supported on the opposite side of the expansion joint from the pump.

g) Tie rods shall be used with double nuts to prevent movement in either direction.

Appendix E

Specialty Piping Components and Applications

This appendix, which is not a part of this standard, discusses a number of highly technical items and calculation methods that relate to the application of pumps in piping systems. The purpose of this appendix is to provide the reader with the knowledge that there are various analytical methods, hardware, components, and references for pumping systems that can help the specifier and user minimize problems and maximize efficiency.

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E.1 Check valves and strainers

E.1.1 Check valves

Check valves are normally put in the outlet (discharge) (not inlet [suction]) line to automatically prevent reverse flow when the pump is stopped. When using a check valve in the inlet (suction) line, consideration should be given to the increased pressure drop to the inlet (suction) of the pump, the possibility of exposing the entire pump casing to outlet (discharge) pressure, and to potential water hammer shock to the piping (see Section 9.6.6.5.4). Submersible (motor-driven) turbine well pumps shall always have a check valve installed in the outlet (discharge) line (column pipe), no more than 7.6 m (25 ft) (for water at sea level) above the lowest liquid (pumping) level in the well or sump (see Section 9.6.6.5.3). This is to reduce backspin on shutdown, to reduce pump rotor upthrust on start-up, and to prevent water hammer. Check valves are sometimes used in series-parallel connections to reduce the number of valves that must be operated when changing pump conditions of service.

E.1.2 Foot valves

Foot valves are specially designed check valves used at the inlet of the suction lift line to maintain pump prime, by maintaining liquid over the first-stage impeller (see Figure E.1). Foot valves are designed to open with very little pressure differential across the valve. They should be installed in a vertical orientation (or they may not work), below the top of the waterline, and the end of the inlet (suction) line should be at least four pipe diameters below the top of the water level. This will maintain a primed condition in the inlet (suction) line. The foot valve and pipe should be sized to minimize inlet (suction) line losses that will maximize the NPSH available to the pump. Problems with leakage and failure to close may be encountered where solids are present in the liquid. Therefore, foot valves may be limited to pump installations where pump nonperformance due to foot valve failure does not place the user at high risk. It must be remembered that, unless a suction pressure relief is fitted, pumps must have the suction side designed to contain the maximum allowable working pressure of the pump, plus any hydraulic shock loading or water hammer due to sudden foot valve closing. In vertical pumps, foot valves may be used at the inlet of the bowl assemblies for

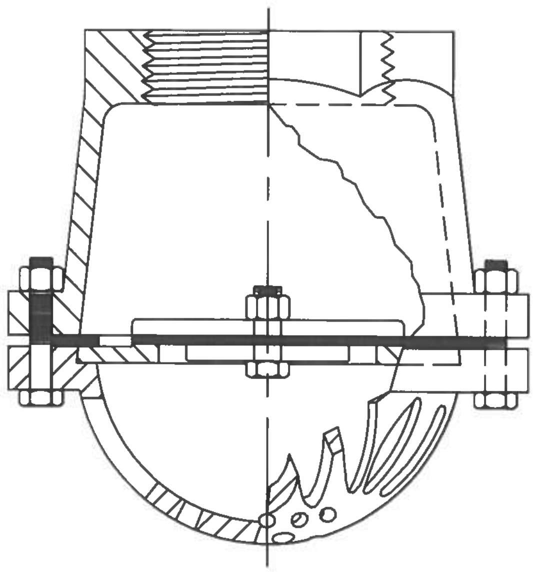

Figure E.1 — Foot valve

well pumps to keep the column pipe filled, to prevent backspin, and prevent well disturbance from rapidly draining water. This practice is very limited and occasionally used on small, less than 3.7-kW (5-hp) pumps, with less than 30-m (100-ft) settings and 345-kPa (50-psig) surface pressure. Check with the vertical turbine pump manufacturer for warranty ramifications when using a foot valve.

E.1.3 Strainers

Strainers may be installed in the pump inlet (suction) piping to keep unwanted material out of the pump. However, the pump inlet (suction) line should be thoroughly flushed before installing a suction strainer. The strainer may be installed at the suction bell in vertical turbine pumps. The strainer usually introduces only a moderate pressure drop. However, as it accumulates debris, the pressure drop will increase. It is, therefore, recommended that strainers be installed with upstream and downstream pressure taps. The pressure drop should be monitored to prevent cavitation. A rule of thumb for strainer application is that the open area be three times the area of the upstream pipe. For a cone-type strainer, the point of the strainer should face upstream (see Figure E.2). In a suction lift application or low positive suction pressure in an open system, the strainer should be sized for a minimum pressure drop. The strainer should not be installed on the pump inlet (suction) nozzle. See Section 9.6.6.3.3, Table 9.6.6.3.2, for the recommended safe distance.

A temporary strainer should be installed (in the pump inlet/ suction line) during system shakedown, prior to initial start-up, when a permanent strainer is not being installed. This is especially important and customary on critical pump services, such as nuclear, power, and refinery. However, the pressure drop must be low enough to provide the pump with adequate NPSH to preclude damaging cavitation. The strainer should remain in the system until periodic inspection shows the system to be clean. A typical temporary strainer installation is shown in Figure E.2. (Note that in some applications it is required that temporary strainers be removed as soon as they are no longer necessary.)

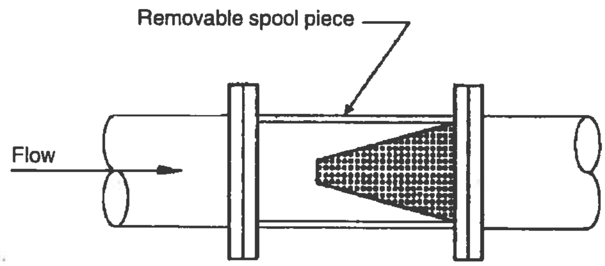

Figure E.2 — Typical temporary strainer

E.2 Devices to improve flow to the pump

In certain cases, implementing this document and, in particular, complying with piping length L_2 (see Section 9.6.6.3.3, Table 9.6.6.3.2, and Figure 9.6.6.3.3a) at the inlet of the pump, may result in large installations whose cost and overall dimensions may seem prohibitive.

It is possible to reduce length L_2 (Figure 9.6.6.3.3a) by fitting a baffle (flow straightener) before or after the inlet convergent pipe. Other devices, such as vaned elbows, also improve inlet flow to the pump. When the suction piping includes a second perpendicular elbow near the pump (see Figure E.3), the flow straightner (vertical splitter) should be positioned in the straight run of pipe leading to the pump, at least five pipe diameters away from the pump.

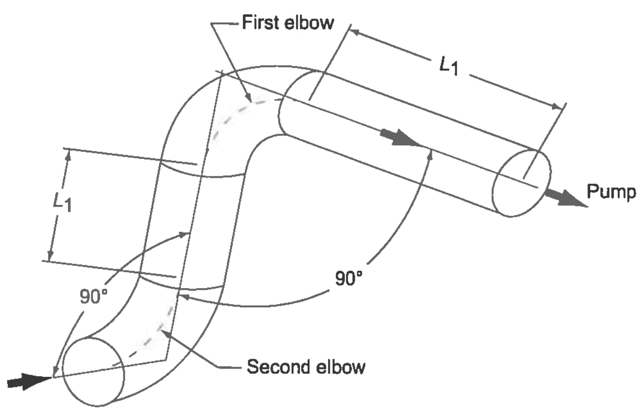

Figure E.3 — Second upstream perpendicular elbow

In cases where an inlet (suction) strainer is required and the inlet pipe length is insufficient to ensure proper flow to the pump inlet, a combination fitting with both a flow strainer and straightening vanes can be used.

Flow improvement devices and the resulting allowable L_2 length shall only be used with mutual consent of the pump supplier and installer, or with applicable and acceptable flow analysis tools.

E.3 Piping for suction lift applications

Self-priming systems require special piping considerations for proper operation, as listed below.

a) The inlet (suction) piping shall be the same size as the pump inlet (suction) flange, unless otherwise approved by the pump supplier. Horizontal runs should be sloped upward to the pump.

b) The static lift must not exceed the pump rating. Depending on the manufacturer, static lift may be defined as: suction lift rating, maximum lift or lift capability, or static priming lift.

c) The static lift and total length of inlet (suction) piping should be minimized to keep priming time to a minimum. Excessive priming time will result in an increased fluid temperature that can cause liquid in the priming chamber to vaporize before prime is achieved.

d) All connections in the inlet (suction) piping shall be leak-free to avoid air being drawn in (see Section 9.6.6.3), which will extend and possibly compromise the priming of the pump.

e) A priming bypass line to allow evacuation of the inlet pipe air volume is recommended to facilitate priming. It is mandatory if a check valve is installed in the pump discharge line (Figure E.4).

f) The inlet (suction) piping should be designed such that no high points are created, where air or vapor can be trapped or accumulate, which can prevent priming.

g) The effects of dissolved gases and altitude should be carefully considered.

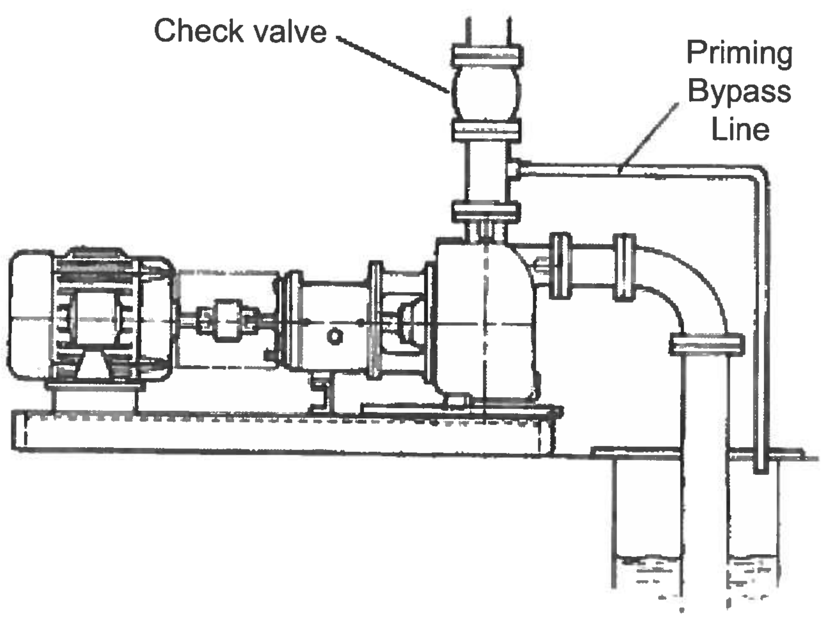

Figure E.4 — Self-priming bypass

E.4 Solids/slurry

Systems intended to carry solids-bearing liquids need special attention. In addition to general piping design guidelines, care is needed to avoid excess energy consumption, settling/blockage, and wear. Factors to consider include pumpage velocity, piping arrangement, and piping materials.

E.4.1 General

Solids/slurry characteristics are beyond the scope of this standard. Refer to ANSI/HI 12.1–12.6 *Centrifugal Slurry Pumps for Nomenclature, Definitions, Applications, and Operation.*

E.4.2 Piping arrangement

Piping should be arranged to avoid sudden changes in direction and areas where solids can accumulate, which could result in rapid wear and blockages. General guidelines include the following:

a) Avoid low spots.

b) Use long-radius elbows wherever turns are needed.

c) If piping inclines are used on heavy slurries, keep the angle slightly less than the angle that would allow gravity to move the solids when the pump is not operating.

d) Provide means to flush the piping in sections where blockage may occur.

e) Keep valves to a minimum.

f) Maintain accessibility to potential wear areas (joints, elbows, and geometry changes).

E.4.3 Materials

Slurries can be very abrasive and corrosive; piping materials must be chosen properly to extend life. The pipe material must be resistant to corrosion by the liquid in the slurry and resistant to erosion by the solids. Slurries are usually handled by carbon steel, or rubber-lined or duplex stainless-steel pipe. The choice is typically governed by particle size and carrier characteristics.

Rubber-lined pipe is often a good choice for small particles like sand. Metal pipes are required if there are larger solids present. In addition, the liquid carrier must also be compatible with the pipe material.

E.5 Air release valves

Air release valves are an essential component in the design of system piping. These valves are hydromechanical devices that vent pockets of air as they accumulate in a system. In general, as the liquid is being pumped, entrained air will bubble up to the high points in the system piping. If the pipeline slopes upward, the velocity of the liquid will move the air bubbles to stagnate at a high point. In contrast, if the pipeline is fairly flat and the inside surface of the pipe is very rough, or the pipeline slopes downward, then the fluid velocity may not be sufficient to keep the air bubbles moving.

As a consequence, it is possible for a pocket of air to collect at these high points, and to gradually reduce the effective liquid flow area, which may create a throttling effect similar to a partially closed valve. Additionally, sudden compression of these air pockets can set up severe shock waves when released with the potential for serious equipment damage.

When pumping into a pressurized system, an automatic air release valve may be required.

For vertical wet pit pumps on water service, it is usually appropriate to use both an air release valve and a vacuum valve. The valves should be located on the high point of the pump outlet (discharge) nozzle and between the pump outlet (discharge) and the outlet discharge check valve. It is important to size and adjust the air release valve to evacuate most of the air on start-up. A small volume of air is desirable to cushion the water as it slams against the check valve on pump start-up. A vacuum valve is usually desirable on vertical turbine wet pit/well pumps to release the vacuum accumulated during pump shutdown, when the water in the column pipe returns by gravity to the wet well. If the vacuum in the column is not released to atmospheric pressure when the pump is shut down, it may result in a damaging water slam against the check valve.

On initial filling of piping systems or after a shutdown that results in draining of the system, caution is required to prevent possible damage to air release valves. During filling of the pipeline with liquid there can be a sudden change in momentum forces on the air release valves when the fluid flowing in the pipe changes from all low-density gas to the higher density liquid. When this interface passes a high point with an air release valve, the valve can see the same sudden change in momentum. The resulting forces may exceed the strength of the air release valve housing.

For horizontal pumps that are not self-venting (trap air at the top of the volute), a valve should be located at the highest point on the volute case.

Additionally, air release valves may be installed on pipeline high points, changes in grade, and at periodic intervals on long horizontal runs that lack a defined high point.

Appendix F

Discharge Pressure Pulsation and Acoustic Resonance

This appendix, which is not a part of this standard, discusses a number of highly technical items and calculation methods that relate to the application of pumps in piping systems. The purpose of this appendix is to provide the reader with the knowledge that there are various analytical methods, hardware, components, and references for pumping systems that can help the specifier and user minimize problems and maximize efficiency.

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F.1 Discharge pressure pulsation/acoustic resonance

F.1.1 General

In addition to producing steady pressure to move fluid, all pumps generate pressure pulses or fluctuations. In the majority of cases with rotodynamic pumps, these pulsations are inconsequential because they are very small in magnitude.

In certain installations, pressure pulsations of significant magnitude can be amplified 5 to 20 times by acoustical resonance in the piping system. These amplified pulses can in turn cause pump vibration and fatigue failure of internal pump components and piping supports. For these pumping applications, the pump system designer should separate the dominant pump-pressure pulsation frequency and determine if the outlet (discharge) piping arrangement has corresponding resonant frequencies. The piping arrangement should be altered to eliminate sympathetic conditions.

The purpose of this section is to inform pump system designers of this phenomenon and to provide basic calculation methods.

F.1.2 Outlet (discharge) piping acoustics

Outlet piping systems have acoustical resonance just as pipe organs have acoustical resonance. If the pipe length, or the distance from the pump to a reflection point, results in multiple half-wave resonance equivalent to the vane passing frequency, the potential exists for the pressure pulsations to be reflected back to the pump and thereby amplified. Outlet (discharge) piping resonance may be calculated using basic wave theory, and reflection points can be listed based on actual field experience.

The half-wave resonance frequency of a pipe may be calculated as follows:

$$\text{Frequency, } f = \frac{ka}{2L}$$

Where:

k = harmonic multiples (1,2,3...n)

a = acoustical velocity of sound in the discharge pipe – m/s (ft/s)

L = distance from the pump's last volute vane (cutwater) to any reflection point in the piping – m (ft)

NOTE: Typically, only the first and second harmonic multiples need be considered.

The acoustical velocity of sound within the pipe may be calculated as follows:

$$\text{Velocity, } a = C_{us}\left(g \times \frac{E}{\gamma}\right)^{0.5} \times \left[\frac{1}{\left(1 + \frac{2RE}{eE_1}\right)}\right]^{0.5}$$

Where:

a = acoustical velocity of sound – m/s (ft/s)

C_{us} = US customary unit correction factor – 12.0

g = Acceleration due to gravity – 9.81 m/s^2 (32.2 ft/s^2)

E = fluid bulk modulus of the liquid (see below)

γ = fluid specific weight – 9.8 kN/m^3 at 15 °C (62.4 lb$_f$/ft^3 at 60 °F) for water

R = inner radius of the pipe – mm (in)

e = thickness of the pipe wall – mm (in)

E_1 = elastic modulus of the wall material (see below)

The bulk/elastic modulus (E_1) for typical materials is

Carbon steel = 21×10^7 kPa (30×10^6 psi)

Cast iron = 10×10^7 kPa (15×10^6 psi)

Prestressed concrete = 1.0×10^7 kPa (1.5×10^6 psi)

Polyethylene = 0.12×10^7 kPa (0.18×10^6 psi)

The bulk/elastic modulus (E) of a typical liquid is

Water at ordinary conditions = 0.20×10^7 kPa (0.30×10^6 psi)

For rigid pipe and ambient water, the acoustical velocity is 1440 m/s (4720 ft/s); with 100-mm (4-in) schedule 40 carbon steel pipe, the velocity is 1330 m/s (4365 ft/s); while for polyethylene plastic 100-mm (4-in) schedule 40 pipe, the value drops to 266 m/s (872 ft/s).

Reflection points are discontinuities in the discharge piping that can reflect the pressure wave back to the source. Typical reflection points include plate orifices, reducers, or increasers with 50% or more area change, tees at headers, and partially closed gate or check valves. Long-radius elbows do not reflect acoustical waves, but transmit them similarly to straight pipe. Short-radius elbows and vaned elbows, on the other hand, may reflect acoustical waves.

Appendix G

Index

This appendix is not part of this standard, but is presented to help the user with factors referenced in the standard.

Note: an f. indicates a figure, and a t. indicates a table.

