ANSI/HI 2.3-2013

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American National Standard for

Rotodynamic Vertical Pumps of Radial, Mixed, and Axial Flow Types

for Design and Application

Hydraulic Institute
6 Campus Drive
First Floor North
Parsippany, New Jersey
07054-4406
www.Pumps.org

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Sponsor

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American National Standards Institute, Inc.

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Foreword (Not part of Standard)

Scope

The purpose and aims of the Institute are to promote the continued growth of pump knowledge for the interest of pump users and pump 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.

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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.

2) Use of Hydraulic Institute Standards is completely voluntary. Existence of Hydraulic Institute Standards does not in any respect preclude a member from manufacturing or selling products not conforming to the Standards.

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 Technical Director of the Hydraulic Institute. The inquiry will then be directed to the appropriate technical committee for provision of a suitable answer.

If a dispute arises regarding contents of an Institute publication or an answer provided by the Institute to a question such as indicated above, then the point in question shall be sent in writing to the Technical Director of the Hydraulic Institute, who shall initiate the Appeals Process.

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 in accordance with the *ANSI Essential Requirements.*

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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. Since 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 to be applied be stated in reference to this standard. If no such statement is provided, metric units shall govern.

Consensus for this standard was achieved by use of the canvass method

The following organizations, recognized as having an interest in the standardization of centrifugal 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 & Consulting, LLC
A.W. Chesterton Company
Black & Veatch Corp.
Brown and Caldwell
Colfax Fluid Handling
ekwestrel corp
GIW Industries
Healy Engineering, Inc.
J.A.S. Solutions Ltd.
John Anspach Consulting
Kemet Inc.
Las Vegas Valley Water District
Patterson Pump Company
Pentair
Pump D^3, LLC
Weir Floway, Inc.
Weir Minerals Hazleton, Inc.
Weir Minerals North America
Xylem Inc.
Zan Kugler P.E., LLC

Committee List

Chair – Fred Walker, Weir Floway, Inc.
Vice-Chair – Michael Coussens, Peerless Pump Company

Committee Members	Company
Paul Behnke	ITT - Industrial Process
Daniel Boldt	Weir Floway, Inc.
James Bonifas	Nidec Motor Corporation
Randal Ferman	ekwestrel corp
Ralph Gabriel	John Crane Inc.
Al Iseppon	Pentair
Aleksander Roudnev	Weir Minerals North America
Paul Ruzicka	Xylem Inc. - Applied Water Systems
Joseph Salah	Sulzer Process Pumps (US) Inc.
Arnold Sdano	Fairbanks Nijhuis
Ernest Sturtz	CDM Smith
Bruce Ticknor III	National Pump Company

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2.3 Design and application

The purpose of this standard is to provide a guide for the design and application of rotodynamic vertical pumps for various services. No attempt has been made to cover all phases of vertical pump design and application, but an endeavor has been made to point out some of the principal features of this type of pump and the precautions that should be taken in its use.

Rotodynamic pumps are kinetic machines in which energy is continuously imparted to the pumped liquid by means of a rotating impeller, propeller, or rotor. The most common types of rotodynamic pumps are radial, mixed, and axial flow, both in horizontal and vertical arrangements.

Vertical pumps offer flexibility of design that is not usually available with other pump types.

- The depth of the pump setting can be selected so that the net positive suction head available (NPSHA) exceeds the net positive suction head required (NPSHR) at all times.

- The pumping element is normally submerged (in a wet pit or can), which eliminates the need for priming devices, enhancing unattended reliable service.

- Minimum floor space is required.

- Many vertical pumps have characteristically steep head versus rate-of-flow curves. A steep head curve characteristic represents less rate-of-flow change with respect to head.

- In the range of intermediate specific speed (n_S [N_S]) designs, 50 - 100 (2500 - 5000), the maximum pump input power usually coincides with the recommended operating range and will not cause driver overload. Typical pump head and power characteristic curves are found in ANSI/HI 2.1-2.2 *Rotodynamic Vertical Pumps for Nomenclature and Definitions*.

- It is often possible to change the staging on the pump, i.e., adding to (or subtracting from) existing equipment or changing impellers in the pump.

2.3.1 Scope

This standard is for design and application of vertical turbine, mixed flow, axial flow vertical diffuser, submersible motor deepwell and short-set pumps, types VS0, VS1, VS2, VS3, VS6, VS7, and VS8 (Figure 2.3.1a) that are driven by vertical electric motors or horizontal engines with right-angle gears. Vertical overhung impeller pumps, types VS4 and VS5 (Figure 2.3.1b) are included in Appendix B. Excluded from the scope of this document are vertical in-line volute pumps and centrifugal volute pumps mounted vertically, such as sewage pumps.

2.3.1.1 Preferred units for pump applications

Preferred terms, units, and symbols to be used in the technology of pump applications are shown in Table 2.3.1.1.

2.3.1.2 Specific speed and suction specific speed

The user is cautioned to check carefully the basis of calculation of specific speed and suction specific speed before making any comparisons because there are subtle but significant differences in methods used throughout industry and in related textbooks and literature.

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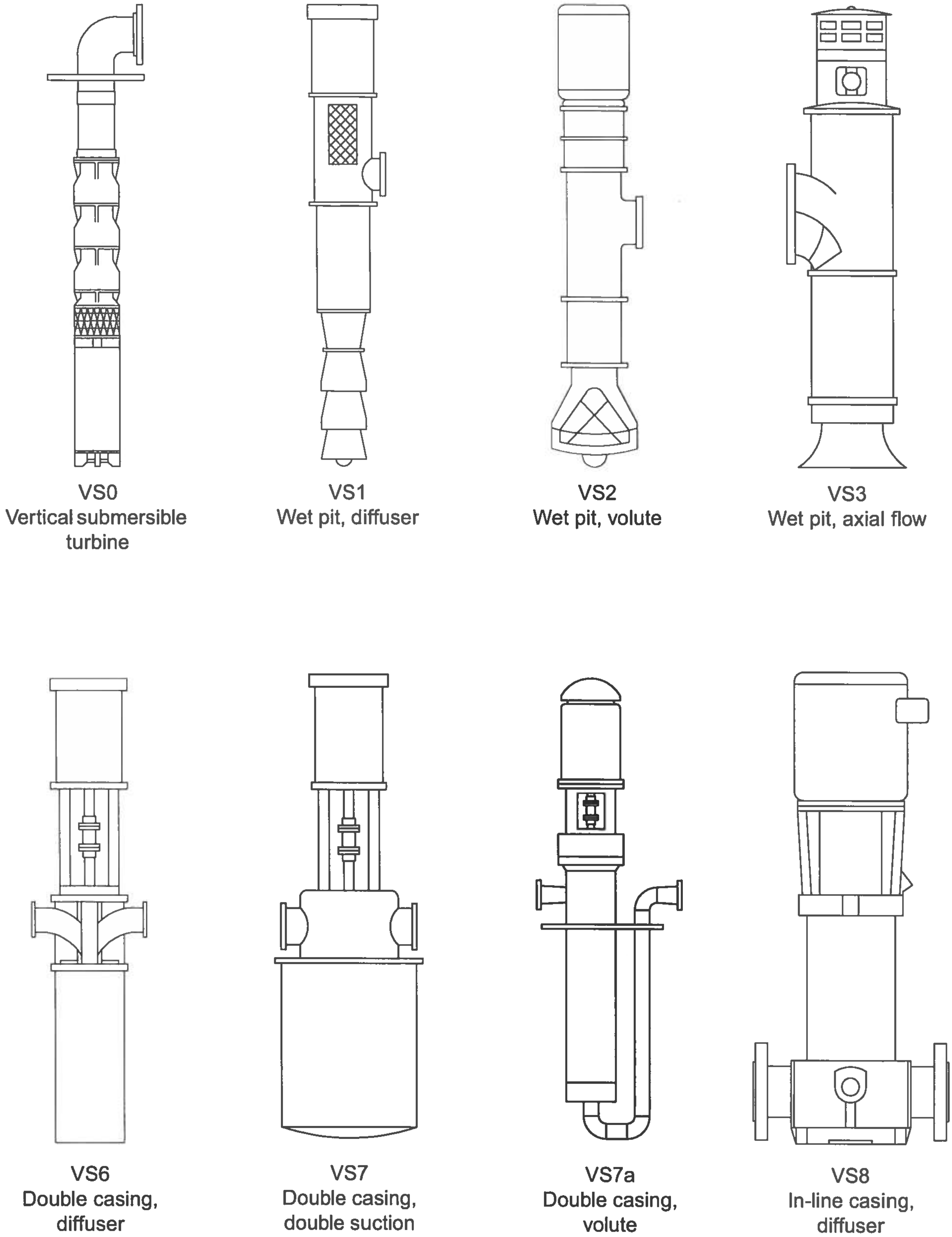

Figure 2.3.1a — Included rotodynamic vertical pump types

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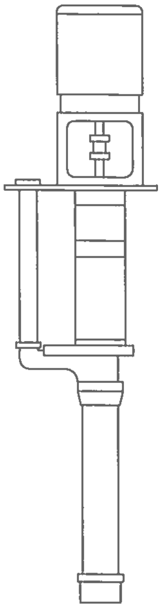

VS4
Vertical sump, line shaft, and

VS5
Vertical sump, cantilever

Figure 2.3.1b — Rotodynamic vertical pump types included in Appendix B

US customary units

When calculating the value for specific speed and suction specific speed, the unit of measurement used for rate of flow is defined in US gallons per minute (gpm).

Specific Speed: An index of pump performance at the pump's best efficiency point (BEP) rate of flow, with the maximum diameter impeller, and at a given rotative speed. Specific speed is expressed by the following equation:

$$n_s = \frac{n(Q)^{0.5}}{(H)^{0.75}}$$

Where:

n_s = specific speed

n = rotative speed, in revolutions per minute (rpm)

Q = total pump flow rate, in cubic meters per second (m^3/s) (US gallons per minute [gpm])

H = head per stage, in meters (m) (feet [ft])

NOTE: Specific speed derived using cubic meters per second and meters multiplied by a factor 51.6, is equal to specific speed derived using US gallons per minute and feet. The usual symbol for specific speed in US units is N_s.

An alternative definition for specific speed is sometimes used based on flow rate per impeller eye rather than total flow rate. In a double suction impeller pump, when this alternative method is used, the resultant value of specific speed is less by a multiplying factor of $1/(2)^{0.5}$, i.e., 0.707 times less.

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Table 2.3.1.2 — Principal symbols

Symbol	Term	Metric Unit	Abbreviation	US Customary Unit	Abbreviation	Conversion Factor[a]
A	Area	square millimeter	mm^2	square inch	in^2	645.2
Bar	Pressure	bar	bar	pound/square inch	psi	0.0689
BEP	Best efficiency point	cubic meter/hour	m^3/h	US gallon/minute	gpm	0.2271
D	Diameter	millimeter	mm	inch	in	25.4
δ (delta)	Deflection	millimeter	mm	inch	in	25.4
Δ (delta)	Difference	dimensionless[b]	—	dimensionless[b]	—	—
η (eta)	Efficiency	percent	%	percent	%	1
F	Force	newton	N	pound (force)	lbf	4.448
g	Gravitational acceleration	meter/second squared	m/s^2	foot/second squared	ft/s^2	0.3048
h	Head	meter	m	foot	ft	0.3048
H	Total head	meter	m	foot	ft	0.3048
K	Thrust factor	newton/meter	N/m	pound/foot	lb/ft	14.59
l	Static lift	meter	m	foot	ft	0.3048
n	Speed	revolution/minute	rpm	revolution/minute	rpm	1
NPSHA	Net positive suction head available	meter	m	foot	ft	0.3048
NPSHR	Net positive suction head required	meter	m	foot	ft	0.3048
NPSH3	Net positive suction head required for 3% head reduction at first stage	meter	m	foot	ft	0.3048
n_s (N_s)	Specific speed $n_s = n(Q)^{0.5}/(H)^{0.75}$ (refer to specific speed and suction specific speed in Section 2.3.1.2)	Index number	—	Index number	—	0.0194
ν (nu)	Kinematic viscosity	millimeter squared/second	mm^2/s	foot squared/second	ft^2/s	92,900
π	pi = 3.1416	dimensionless	—	dimensionless	—	1
p	Pressure	kilopascal	kPa	pound/square inch	psi	6.895
P	Power	kilowatt	kW	horsepower	hp	0.7457
Q	Rate of flow (Capacity)	cubic meter/second	m^3/s	US gallon/minute	gpm	0.0000631
Q	Rate of flow (Capacity)	cubic meter/hour	m^3/h	US gallon/minute	gpm	0.2271
RM	Linear model ratio	dimensionless	—	dimensionless	—	1
ρ (rho)	Density	kilogram/cubic meter	kg/m^3	pound mass/cubic foot	lbm/ft^3	16.02
S (N_{SS})	Suction specific speed = $n(Q)^{0.5}/(NPSH3)^{0.75}$ (refer to specific speed and suction specific speed in Section 2.3.1.2)	Index number	—	Index number	—	0.0194
s	Specific gravity	dimensionless	—	dimensionless	—	1
t	Temperature	degree Celsius	°C	degree Fahrenheit	°F	(°F – 32) × $^5/_9$
τ (tau)	Torque	newton-meter	N•m	pound-foot	lb•ft	1.356
U	Residual unbalance	gram-millimeter	g-mm	ounce-inch	oz-in	720
v	Velocity	meter/second	m/s	foot/second	ft/s	0.3048
x	Exponent	none	none	none	none	1

[a] Conversion factor × US customary units = metric units (except temperature).
[b] Δ is a dimensionless symbol used to indicate a difference. This term takes on the units of the measured or calculated quantity associated with the difference.

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Suction specific speed: An index of pump suction operating characteristics determined at the BEP rate of flow for the first-stage impeller at the maximum diameter. Suction specific speed is an indicator of the net positive suction head required for a 3% drop in head (*NPSH3*) at a given rate of flow (*Q*) and rotative speed (*n*) and is expressed by the following equation:

$$S = \frac{n(Q)^{0.5}}{(NPSH3)^{0.75}}$$

Where:

S = suction specific speed

n = rotative speed, in rpm

Q = flow rate per impeller eye, in m^3/s (gpm)

= total flow rate for single suction impellers

= one half total flow rate for double suction impellers

NPSH3 = net positive suction head required in meters (feet) that will cause the total head (or first-stage head of multistage pumps) to be reduced by 3%. The required NPSH (NPSHR) qualified by this criterion will be referred to as *NPSH3*.

NOTE: Suction specific speed derived using cubic meters per second and meters, multiplied by a factor of 51.6, is equal to suction specific speed derived using US gallons per minute and feet. The US customary symbol N_{ss} is sometimes used to designate suction specific speed.

2.3.1.3 Introduction to pump classifications

Rotodynamic pumps may be classified by such methods as impeller or casing configuration, end application, specific speed, or mechanical configuration. The method used within this standard (as indicated in Appendix A, Figures A.1 – A.3) is based primarily on commonly distinctive mechanical configurations. Commonly used pump types are classified as overhung (Type OH), between bearings (Type BB), or vertically suspended (Type VS).

ANSI/HI Standards (for design and application) have historically been subdivided into

rotodynamic centrifugal pumps (ANSI/HI 1.3)
and
rotodynamic vertical pumps (ANSI/HI 2.3)

with a demarcation between the two categories being determined by the arrangement of the hydraulic configuration (impeller or casing). On either side of this line of demarcation are pump types that can be clearly identified to fit into each of the defined categories.

There are also several pump types or arrangements that are not so clearly defined.

Appendix B in ANSI/HI 1.3 provides an identification and introduction into such arrangements, and discusses any design and application considerations that may be considered relevant to these specific configurations.

2.3.2 Impeller types

An enclosed impeller, Figure 2.3.2a, has the vanes enclosed between a front shroud and back shroud. It is commonly used for all pump types in the medium to high head per stage range.

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A semi-open impeller, Figure 2.3.2b, has the vanes attached only on the back shroud, with the exposed vanes on the inlet-facing side running in close proximity to a matching case wall, liner, or cone. Impellers of this type are commonly used over a wide range of specific speeds, particularly for services where impeller-to-seat clearance may be adjusted and where moderate amounts of small solids are present in the pumped fluid.

An open axial flow impeller (Figure 2.3.2c) has a single inlet with the flow entering and discharging axially (or nearly axially). Impellers of this type are sometimes called *propellers* and do not have shrouds. Impellers of this type are typically used for low-head, single-stage applications.

A double suction impeller (Figure 2.3.2d) discharges radially into a casing designed to conduct the liquid axially upwards to a vertical discharge conduit, or to the next pumping stage in series in a multistage arrangement. This double suction impeller has both an upper and lower inlet eye and hydraulic axial thrust is inherently balanced, with theoretically zero net hydraulic axial thrust transmitted to the shaft. Enhanced suction performance – reduced NPSHR – is provided by the combined performance of two impeller inlet eyes operating in parallel.

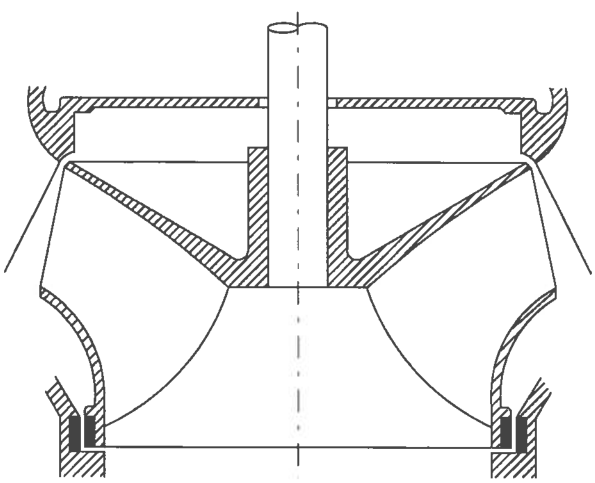

Figure 2.3.2a — Enclosed impeller

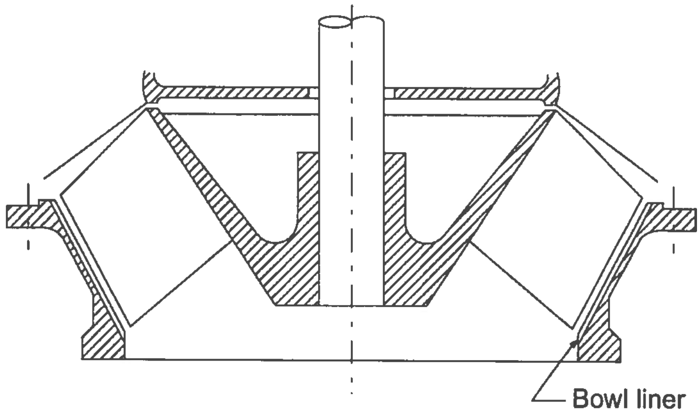

Figure 2.3.2b — Semi-open impeller

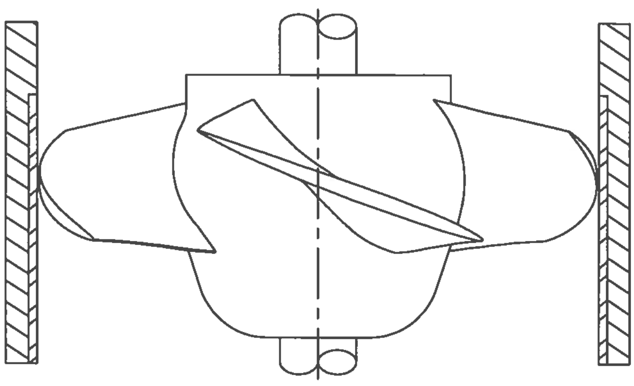

Figure 2.3.2c — Axial flow impeller (propeller)

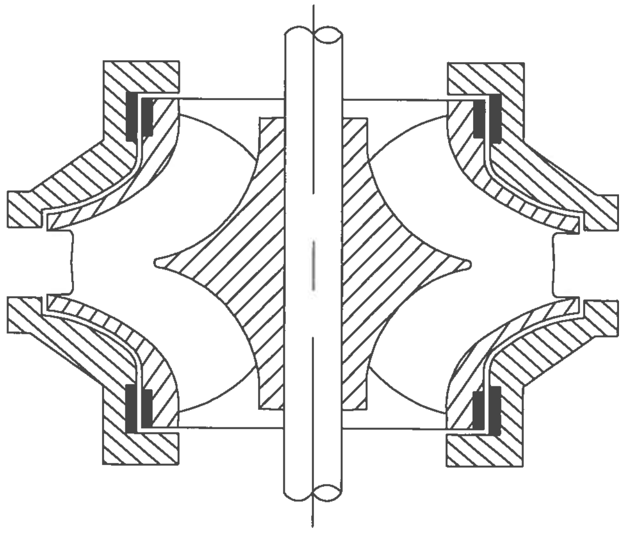

Figure 2.3.2d — Double suction impeller

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The double suction impeller and the inducer first-stage (Figure 2.3.2e) designs are typically used where the NPSHA is low, such as in a suction receiver (can), where the NPSHA is provided only by the liquid elevation above the impeller. The superior NPSH performance of a double suction impeller or inducer might also enable a higher operating rpm.

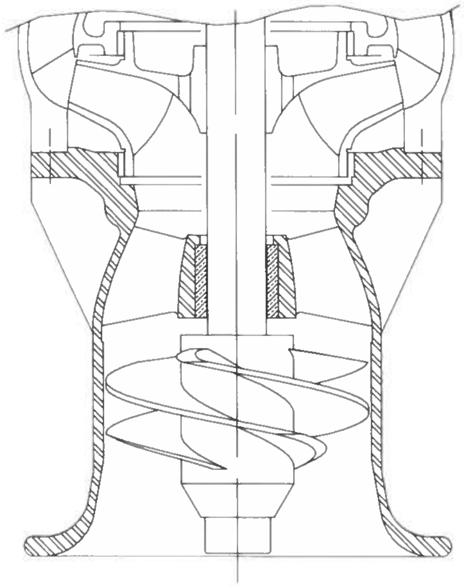

Figure 2.3.2e — Inducer

2.3.3 Mechanical features

2.3.3.1 Radial thrust

Radial thrust is a lateral force, in a direction perpendicular to the shaft, acting against the rotor. Radial thrust generated in a vertically suspended vertical turbine pump (VTP) bowl assembly is considered very low. Pressure forces around the impeller periphery are equalized by the multivane bowl diffusers. In theory this eliminates pump rotor loading on the sleeve-type bowl bearing bushings. Thus the multivane diffusers provide a broad flow range of hydraulic stability. This is generally true for specific speed pumps less than 120 (6000) n_s with a continually rising head–capacity curve.

Radial rotor thrust forces can develop later in a bowl assembly life. In general, these forces occur from impeller hydraulic and mechanical imbalances due to loss of material from erosion, corrosion, cavitation, mechanical problems, etc.

Ideally a vertical pump's rotor assembly is suspended, hanging plumb from the thrust bearing, and in this condition the lower portions of the rotor do not rest on the sleeve bearing bushings. When the pump mounting is tilted from plumb vertical, gravity will pull the rotor (shafting) to rest against the sleeve bearing bushings. As a result, additional sleeve bearing bushing wear may occur from a portion of the rotor weight acting laterally against the sleeve bearing bushings, especially during frequent operational cycles of start-up and shut-down, or when the pumped liquid contains abrasive solids. However, inclined vertical pumps are occasionally the best solution for an application, given appropriate bearing and lubrication options are selected.

Vertical pump line-shaft bearing bushings are of the sleeve type and typically experience very little radial loading. Optimum line-shaft bearing bushing life can be expected with proper radial alignment, lubrication, bearing spacing, and sufficient shaft straightness.

Vertical submersible turbine pumps (VS0) and vertical multistage in-line pumps (VS8) have similar considerations. In the typical installation, the rotating shaft is centered vertically in the bearing bushings. In cases where the pump is mounted horizontally, the weight of the rotor becomes the major source of radial thrust on the pump bearing bushings.

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2.3.3.2 Axial thrust

2.3.3.2.1 Terminology

A_{slv} — Area of sleeve O.D. minus I.D., in mm^2 (in^2)

A_{db} — Net area back ring area minus shaft area, in mm^2 (in^2)

A_s — Area of shaft exposed on lower end of bowl shaft, in mm^2 (in^2)

A_{df} — Net impeller area front ring area minus shaft area, in mm^2 (in^2)

B — Number of bowl assembly stages

C — Experimental thrust coefficient – dimensionless (Figure 2.3.3.2.5)

D_{rb} — Impeller back ring diameter, in mm (in)

D_{rf} — Impeller eye, front ring diameter, in mm (in)

D_s — Shaft diameter, in mm (in)

D_{slv} — Shaft sleeve O.D., in mm (in)

F_{afr} — Net bowl assembly axial thrust at any flow rate, in N (lb)

$F_{a\,\mathrm{BEP}}$ — Net bowl assembly axial thrust at BEP, in N (lb)

F_n — Net pump assembly axial thrust, in N (lb)

F_s — Pump shaft axial thrust due to suction pressure on bottom end of bowl shaft, in N (lb)

F_{slv} — Axial thrust from shaft sleeve, in N (lb)

H_{Stg} — Single-stage head at BEP, in m (ft)

p_{slv} — Pressure differential across sleeve in packing box or mechanical seal, in kPa (psi)

p_s — Suction pressure on exposed lower end of bowl shaft, in kPa (psi)

T — Thrust per stage from theoretical calculation at BEP, in N (lb)

W — Dead weight of one stage rotating assembly, shaft and impeller, in N (lb)

W_r — Dead weight of all rotating components above the bowl assembly, in N (lb). This includes line-shafting and line-shaft couplings, shaft through driver, if applicable, and motor to pump shaft coupling, if applicable.

2.3.3.2.2 Introduction

Axial thrust (F_n,) is the net force acting through the rotor on the thrust bearing. Components of axial thrust include: 1) dynamic forces resulting from pressure and momentum acting on the rotating impellers and other rotor surfaces and 2) static forces, i.e, rotor weight. Net axial thrust can be positive (downthrust) or negative (upthrust). Axial downthrust is normal in a vertically suspended bowl assembly and keeps the pump shaft tensioned.

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The equations for calculating the theoretical axial thrust (*T*) per stage are provided in Sections 2.3.3.2.2 to 2.3.3.2.4. However, due to the complex and not easily analyzed nature of the forces present, refer to the individual manufacturer for their actual thrust values.

Usually the pump manufacturer has thrust data available over the bowl assembly's entire flow range. These data can be generated from actual bowl assembly axial thrust tests. It is recognized that thrust values for similarly sized bowl assemblies will differ among manufacturers due to differences in hydraulic design.

The net axial downthrust force is carried by the pump shaft. The shaft will stretch, i.e., elongate, under this load. Before the pump starts up, any stretch that occurs is due to rotor weight, the sum of the static forces. The thrust load will increase after the pump starts up due to the addition of the dynamic forces. To prevent the impeller from bottoming up against the bowl and damaging the pump when the shaft stretches under running load, the impeller axial position in the bowl must be set before the pump starts. Shaft elongation is maximum at shut-off flow conditions due not only to the highest head, but also due to increased value of the thrust factor, *K*.

On very long, deep settings, shaft stretch may exceed the axial clearance available within the bowl assembly. This requires special machining for increased clearance with some reduction in efficiency. Increasing the diameter of the column shaft will reduce the axial stretch.

The allowable axial adjustment, the maximum axial clearance, with axial flow and semi-open impeller pumps is typically very small. These pumps require tight clearances to prevent large losses and reduction of pump performance.

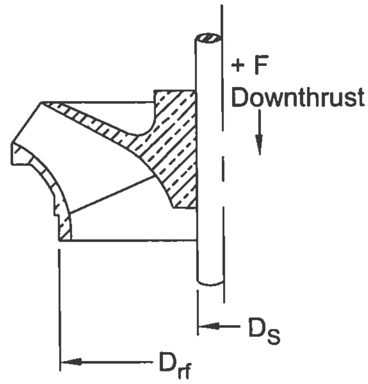

Figure 2.3.3.2.3a — Enclosed impeller plain top shroud

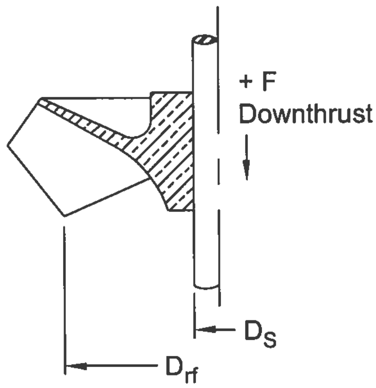

Figure 2.3.3.2.3b — Semi-open impeller

2.3.3.2.3 Dynamic forces

The dynamic forces creating thrust on a vertical turbine pump enclosed impeller (Figure 2.3.3.2.3a) are due to the difference in pressure distributions on the upper and lower shrouds along with the force from the change in momentum of the flow through the impeller.

The semi-open impeller (Figure 2.3.3.2.3b) has only an upper shroud. The difference in pressure distributions along both the backside and the vaned side of the shroud is typically greater than between upper and lower shrouds of an enclosed impeller. Semi-open impeller axial thrust is higher than that of the enclosed impeller.

The axial flow pump impeller (propeller) has no upper or lower shroud; vanes are attached directly to the hub. The axial thrust generated is primarily from dynamic forces created by interaction of the propeller vanes with liquid.

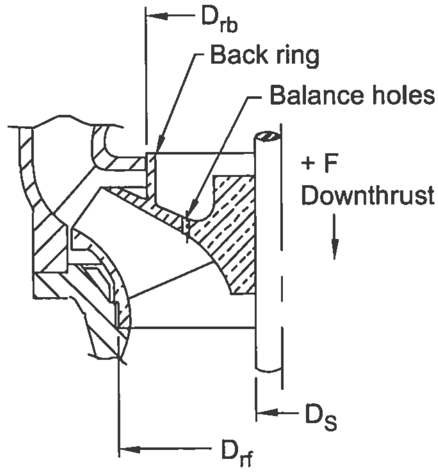

Figure 2.3.3.2.3c — Enclosed impeller with back ring and balance holes

Impeller back ring with balance holes configuration (Figure 2.3.3.2.3c) reduces the axial thrust. Back rings may be cast integrally into impellers with a top shroud. They are used when pump total axial thrust requires reduction. The flow through balance holes in the impeller hub shroud, combined with the leakage past the balance ring, reduces efficiency. The exact efficiency reduction depends on the individual design and pump size and specific speed. The effect of increased leakage through clearances due to wear of the back ring arrangement may be an increase in downthrust and should be considered in sizing the thrust bearing.

Vertical pumps may have a first stage whose design differs from the subsequent stages to reduce the NPSHR of the pump. Thrust contributions of any unique first stage must be considered in the final calculation of axial thrust.

2.3.3.2.3.1 Shaft upthrust forces (negative value)

Short-time transient upthrust during start-up may occur with little static discharge head, such as when filling the pump column or a conduit.

Upthrust force may also occur in the following situations:

- From operating at rates of flow beyond BEP with low-head systems.
- From enclosed suction can pumps where suction pressure exists on the exposed lower end of the bowl shaft (Figure 2.3.3.2.3.1a). Also, considerations may be required for static (nonoperating) condition pump shaft uplift.
- From shaft sleeves through packing boxes and pressure breakdown bushings. Usually the forces are small except when very high pump discharge pressure exists and shaft sleeve cross-sectional areas are large (Figures 2.3.3.2.3.1b and 2.3.3.2.3.1c).
- From mechanical seal shaft sleeves. When the seal housing pressure is high, significant upward force on the seal sleeve can be imposed on the sleeve cross-sectional area. This force is transmitted to the pump shaft from the secured shaft sleeve (Figure 2.3.3.2.3.1b).

2.3.3.2.4 Static forces

The static component of axial thrust is the sum of the dead weights of impellers, bowl shaft, line shaft, line-shaft couplings, shaft through driver (if applicable), and driver to pump shaft coupling (solid-shaft driver) or adjusting nut (hollow-shaft driver).

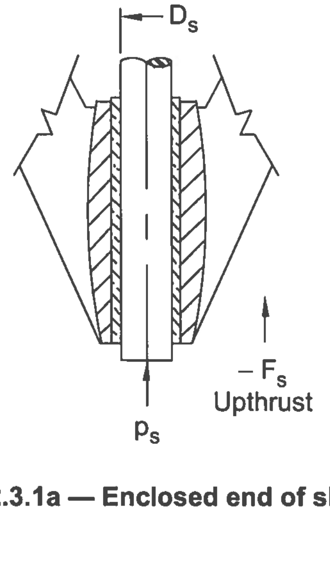

Figure 2.3.3.2.3.1a — Enclosed end of shaft at suction

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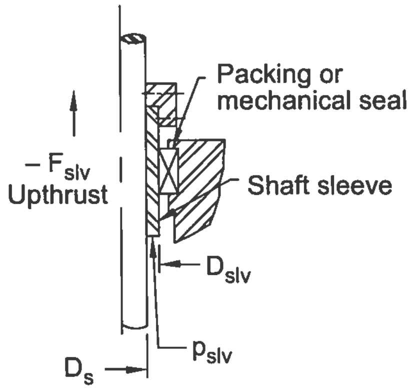

Figure 2.3.3.2.3.1b — Shaft sleeve through packing or mechanical seal

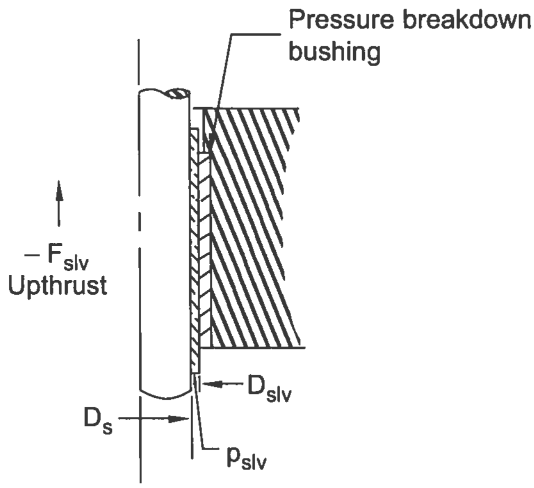

Figure 2.3.3.2.3.1c — Shaft sleeve through pressure breakdown bushing

2.3.3.2.5 Estimated pump axial thrust at BEP (experimental coefficient method)

It may be helpful to estimate the pump's axial shaft thrust when manufacturer's actual thrust factors are not available. Figure 2.3.3.2.5 provides the experimental coefficient *C*, which includes the impeller flow momentum change. This coefficient along with the impeller eye area dimensions are required to estimate the pump thrust per the following calculations. For enclosed impellers, the solid line for values of *C* represents an average from a number of tests on rotodynamic vertical pumps from 33 (1700) through 230 (12,000) specific speeds.[1] For semi-open impellers, the dashed line is from Stepanoff.[2] However, a pump manufacturer's specific design may provide a slightly different *C* value. When using the graph, the value *C* obtained is to be used as an estimated value only at pump BEP.

a) Axial thrust load (for impellers with no back ring) only at pump BEP.

Net pump assembly axial thrust -
$F_n = F_{a\ \mathrm{BEP}} - F_{slv} + W_r - F_S$

Net bowl assembly axial thrust -
$F_{a\ \mathrm{BEP}} = (T + W) \times \mathrm{B}$

Thrust per stage at BEP -

Metric units: $T = \dfrac{H_{stg} \times \rho \times g \times C \times A_{df}}{1 \times 10^6}$

US customary units: $T = \dfrac{H_{stg} \times \rho \times C \times A_{df}}{144}$

Impeller area calculation, front ring area minus shaft area -
$A_{df} = 0.785\ (D_{rf}^2 - D_s^2)$

---

[1] J. L. Dicmas, *Vertical Turbine, Mixed Flow & Propeller Pumps*, McGraw-Hill Book Co., New York, NY, 1987, pp 83-86.
[2] A.J. Stepanoff, *Centrifugal & Axial Flow Pumps*, 2nd Ed, John Wiley & Sons, New York, 1957 - Fig 7.26.

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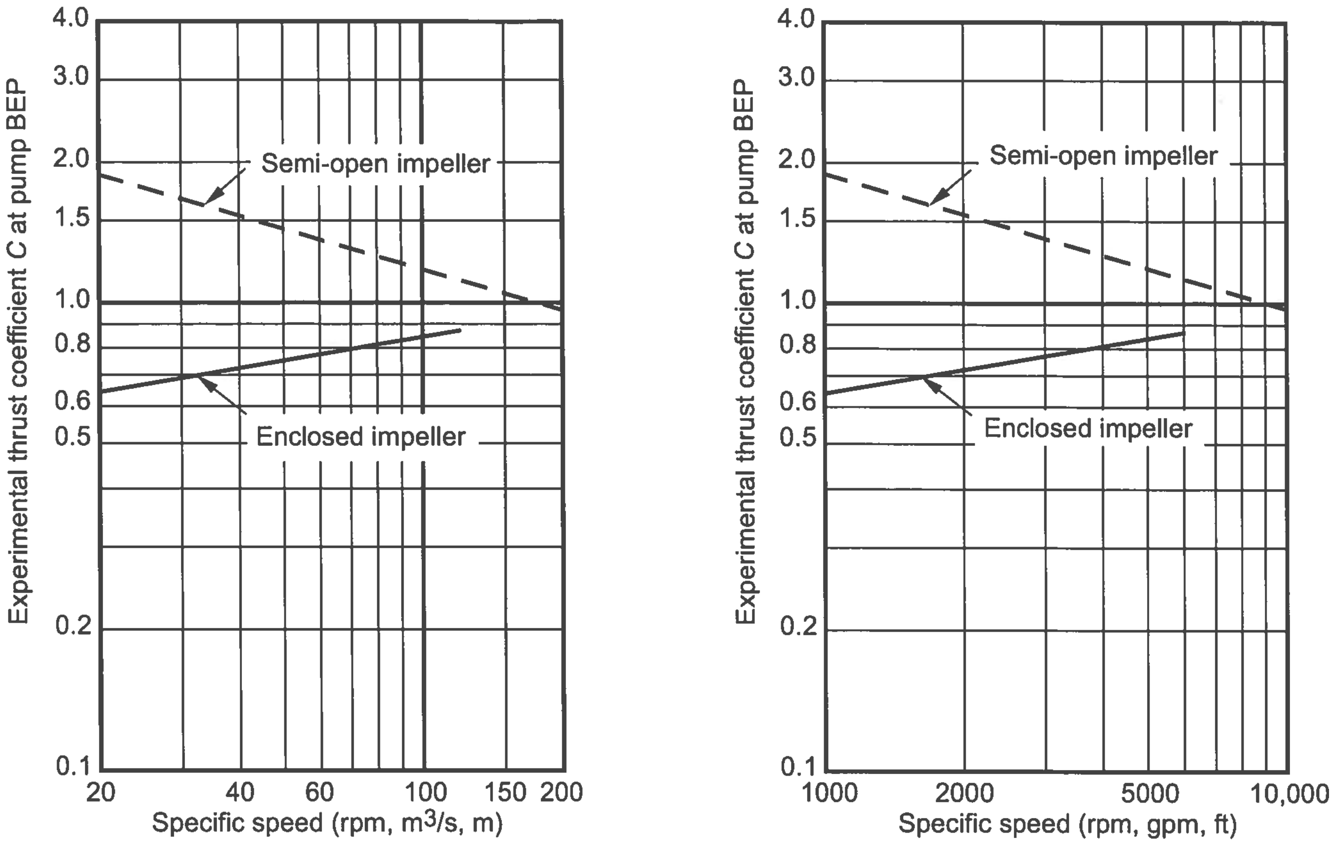

Figure 2.3.3.2.5 — Experimental thrust coefficient C[1, 2]

Thrust on bowl shaft end (negative value) -

Metric units: $F_s = \dfrac{p_s \times A_s}{1000}$

US customary units: $F_s = p_s \times A_s$

Thrust on pump shaft sleeves (negative value) -

Metric units: $F_{slv} = \dfrac{p_{slv} \times A_{slv}}{1000}$

US customary units: $F_{slv} = p_{slv} \times A_{slv}$

Sample calculation:

Let H_{stg} = 30.48 m (100 ft)

C = 1.0

[1] J. L. Dicmas, *Vertical Turbine, Mixed Flow & Propeller Pumps*, McGraw-Hill Book Co., New York, NY, 1987, pp 83-86.
[2] A.J. Stepanoff, *Centrifugal & Axial Flow Pumps*, 2nd Ed, John Wiley & Sons, New York, 1957 - Fig 7.26.

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ρ = 1000 kg/m^3 (62.4 lbm/ft^3)

A_{df} = 25,806 mm^2 (40 in^2)

g = 9.81 m/s^2

In metric units:

$$T = \frac{30.48 \times 1000 \times 9.81 \times 1.0 \times 25{,}806}{1 \times 10^6} = 7716 \text{ N}$$

In US customary units:

$$T = \frac{100 \times 62.4 \times 1.0 \times 40}{144} = 1733 \text{ lbf}$$

Note that 1 N = 0.2248 lbf

7716 N × (0.2248 lbf/N) ~= 1733 lbf

b) Axial thrust load (for impellers with back ring) only at pump BEP.

Net pump assembly axial thrust -
$F_n = F_{a\ \mathrm{BEP}} - F_{slv} + W_r - F_s$

Net bowl assembly axial thrust -
$F_{a\ \mathrm{BEP}} = (T + W) \times B$

Thrust per stage at BEP – (The pressure behind the back ring with impeller with balance holes that have a net area three times the diametrical ring clearance is approximately 35 kPa [5 psi] above the suction pressure at four-pole speed.)

Metric units: $$T = \frac{H_{stg} \times \rho \times g \times C \times (A_{df} - A_{db})}{1 \times 10^6} + \frac{35 \times A_{db}}{1000}$$

US customary units: $$T = \frac{H_{stg} \times \rho \times C \times (A_{df} - A_{db})}{144} + 5 \times A_{db}$$

Impeller area calculation, front ring area minus shaft area -
$A_{df} = 0.785\ (D_{rf}^2 - D_s^2)$

Impeller area calculation, back ring area minus shaft area -
$A_{db} = 0.785\ (D_{rb}^2 - D_s^2)$

Thrust on bowl shaft end (negative value) -

Metric units: $$F_s = \frac{p_s \times A_s}{1000}$$

US customary units: $F_s = p_s \times A_s$

Thrust on pump shaft sleeves (negative value) -

Metric units: $$F_{slv} = \frac{p_{slv} \times A_{slv}}{1000}$$

US customary units: $F_{slv} = p_{slv} \times A_{slv}$

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2.3.3.2.6 Estimating axial thrust (manufacturer's thrust factor method)

Thrust calculations for thrust bearing sizing and shaft elongation can be made more accurately by using the pump manufacturer's thrust factor. This assumes the pump will operate in an operating range around BEP. To calculate the pump's thrust, insert the manufacturer's published thrust factor in the following equations:

Net pump assembly axial thrust -

$$F_n = F_{afr} - F_{slv} + W_r - F_s$$

Net bowl assembly axial thrust -

$$F_{afr} = (\mathrm{K} \times H_{Stg} \times s) \times B + (W \times B)$$

Thrust on bowl shaft end (negative value) -

Metric units: $F_s = \dfrac{p_s \times A_s}{1000}$

US customary units: $F_s = p_s \times A_s$

Thrust on pump shaft sleeves (negative value) -

Metric units: $F_{slv} = \dfrac{p_{slv} \times A_{slv}}{1000}$

US customary units: $F_{slv} = p_{slv} \times A_{slv}$

2.3.3.2.7 Estimating axial thrust at rates of flow other than BEP

Knowledge of thrust at flow rates other than pump BEP is sometimes necessary. Specific applications require vertical pumps to operate continuously at flow rates near shutoff or near pump runout. Special consideration may be necessary to limit line-shaft elongation to within the impeller endplay limits. Also, vertical motor or pump thrust bearings may be selected to contain pump thrust, both up and down.

When evaluating a vertical pump's thrust versus flow characteristics, it may be necessary to determine the following:

- Flow rate at which pump's thrust may be negative (upthrust), usually at runout flows
- Magnitude of upthrust
- Magnitude of downthrust at or near zero flow (pump shutoff)

Figure 2.3.3.2.7, Axial thrust versus rate of flow curves, shows the thrust factor percentage versus percent of BEP rate of flow for different impeller designs. Refer to pump manufacturers for pump thrust factor curves for other impeller designs. See Section 2.3.3.2.3 for axial thrust equation.

2.3.3.3 Shafting

The Hydraulic Institute recognizes and adopts the ANSI/AWWA E103-07 *AWWA Standard for Shafted Pumps* document for pump shafting power transmission design. (See current edition for formulae and design details.) This is a nationally recognized VTP industry standard. The design stress safety margins have been successfully used since the 1960s. The standard states the pump shafting maximum combined shear stress shall not exceed 30% of the material yield tensile strength (YTS) nor be more than 18% of the ultimate tensile strength (UTS). Also it states a threaded shaft coupling maximum combined shear stress shall not exceed 20% of the material YTS nor be more than 12% of the UTS.

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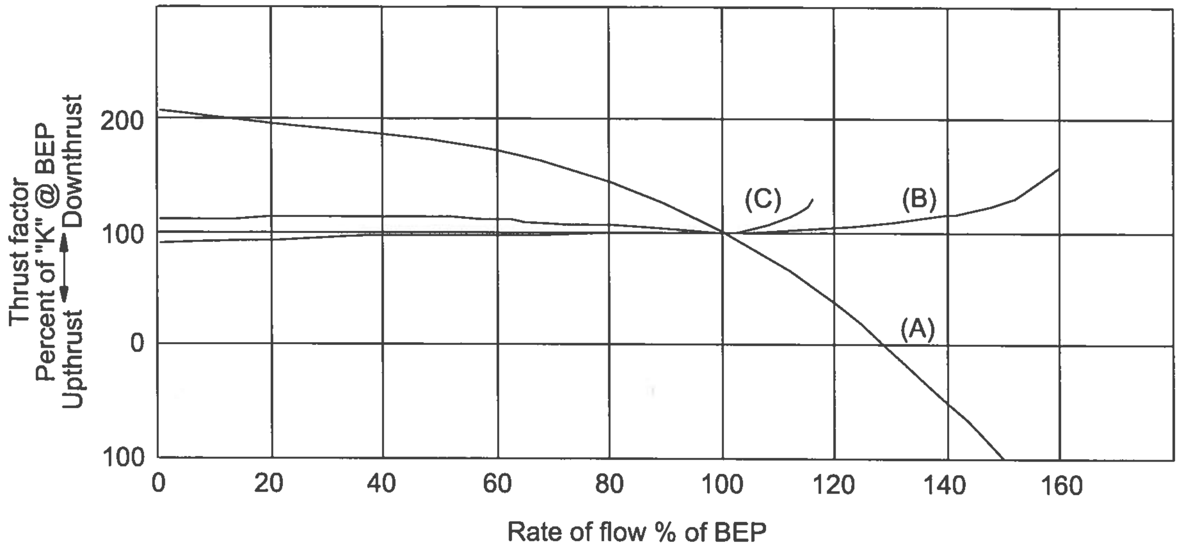

A) Example - Low n_s vertical turbine, enclosed impeller.
B) Example - Low n_s vertical turbine, semi-open impeller.
C) Example - High n_s mixed flow pump.

Figure 2.3.3.2.7 — Axial thrust versus rate of flow curves

The formulae for determining combined shaft shear stress are as follows:

Metric units:

$$S_{cs} = (1 \times 10^3) \times \sqrt{\left(\frac{2 \times F_A}{\pi \times D^2}\right)^2 + \left(\frac{4.86 \times 10^7 \times P}{n \times D^3}\right)^2}$$

US customary units:

$$S_{cs} = \sqrt{\left(\frac{2 \times F_A}{\pi \times D^2}\right)^2 + \left(\frac{321{,}000 \times P}{n \times D^3}\right)^2}$$

Where:

S_{cs} = combined shear stress, in kPa (psi)

F_A = axial thrust acting through the shafting, in N (lb)

D = shaft diameter at the root of the threads or minimum diameter at any undercut, in mm (in)

P = power transmitted by the shaft, in kW (hp)

n = rotational speed of pump, in rpm

The line shafts shall be of a material and size that will transmit the torque from the driver to the impellers and support the maximum thrust load with a proper factor of safety.

The threaded ends of line shafts should be connected with a shaft coupling that has a factor of safety greater than that of the shaft. Threads at these joints shall be such that they tighten when operating.

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On larger-diameter shafts, threaded connections can be difficult to assemble or disassemble. Alternatively, a coupling connection built to accommodate keys to transmit torque and split thrust rings or other means to transmit thrust should be used.

Shaft sleeves of suitable metal or special coating are sometimes provided at bearings and shaft seals.

2.3.3.4 Open line shaft and enclosed line shaft

An open line-shaft-type deepwell pump and an enclosed line-shaft-type deepwell pump are shown in ANSI/HI 2.1–2.2 *Rotodynamic Vertical Pumps for Nomenclature and Definitions*. Examples of these pumps are also shown herein; see Figures 2.3.3.4a and 2.3.3.4b.

The open line-shaft pump is often referred to as a *product-lubricated* or *water-lubricated pump*. The lubrication for an enclosed line-shaft pump may be oil, grease, filtered pump discharge water, or clean water from an external source. Representative cutaway sections of open and enclosed line-shaft construction for vertical turbine pumps are shown in Figures 2.3.3.4c and 2.3.3.4d.

The open line-shaft bearing bushings are subject to abrasive wear when the pumped liquid contains sand or other suspended solids. The open line-shaft type does not have a possible contamination problem, which can exist with the enclosed line-shaft type when drip oil or packed grease is used. Other examples of product-lubricated pump constructions can be found in ANSI/HI 2.1–2.2. Pump types VS0 and VS8 are all product lubricated.

The enclosed line-shaft configuration isolates the line-shaft bearings from the pumped product using an inner enclosing tube (inner column) assembly that encloses the shafting and bearing system. This option is desirable when the pumped liquid contains solids or abrasives that can rapidly wear out the bearings or the shaft journals. With enclosed line shaft, clean bearing lubricant can be injected or fed down through the enclosing tube assembly to the line-shaft bearings. The lubrication for an enclosed line-shaft pump may be oil, grease, filtered pump discharge water, or clean water from an external source. A bypass port in the top of the bowl assembly, not necessarily used on lower head or single-stage applications, prevents the building up of pressure in the shaft-enclosing tubes.

Mineral oil contamination of wells is a possible problem with enclosed line shaft, therefore, biodegradable oils are frequently used. Selecting a proper biodegradable oil is important because many of these oils promote molds and bacteria growth.

There are numerous variations of the open and enclosed line-shaft constructions, especially in the short-setting pumps and custom engineered-to-order verticals. A variety of bearing constructions and lubrication systems are applied to suit different applications.

2.3.3.5 Types of bearing bushings and spacing

The open line-shaft-type vertical turbine pump for water service is normally built with grooved synthetic rubber line-shaft and bowl bearing bushings. The enclosed line-shaft type is normally built with bronze line shaft and bowl bearing bushings. Other materials, such as carbon, PTFE, tungsten carbide, or composites can be used for water or other pumped liquids.

Open line-shaft bearing bushings are product lubricated. Bronze bearings may be used for short-set open line shafts in clean water applications. Enclosed line-shaft bearing bushings of all material types could be gravity-fed water lubricated, or bearing bushings of bronze could be oil lubricated. Some bowl assemblies are built with tandem bearing bushings having both bronze and rubber running surfaces. Rubber bearing bushings outlast bronze bearing bushings while operating in sandy water. However, with rubber bearings, the shaft or sleeve abrasive wear may govern useful life. Bowl assemblies expected to pump abrasive water are often fitted with rubber bearings throughout. The suction case or bell bearing bushing, when provided, may be packed with grease during assembly and safeguarded at the top by a protecting sand collar fitted to the shaft and enshrouding the upper end of the bearing bushing.

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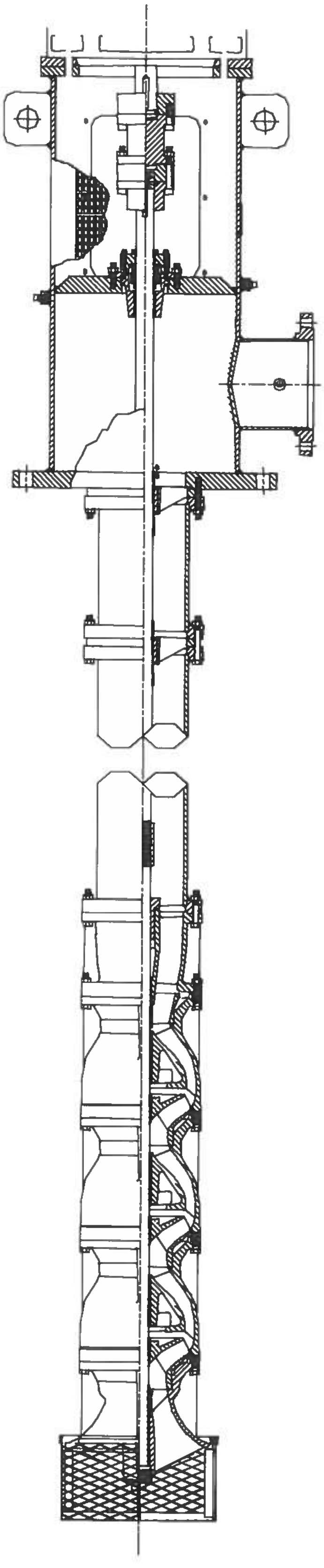

Figure 2.3.3.4a — Open line-shaft, product-lubricated, vertical turbine pump

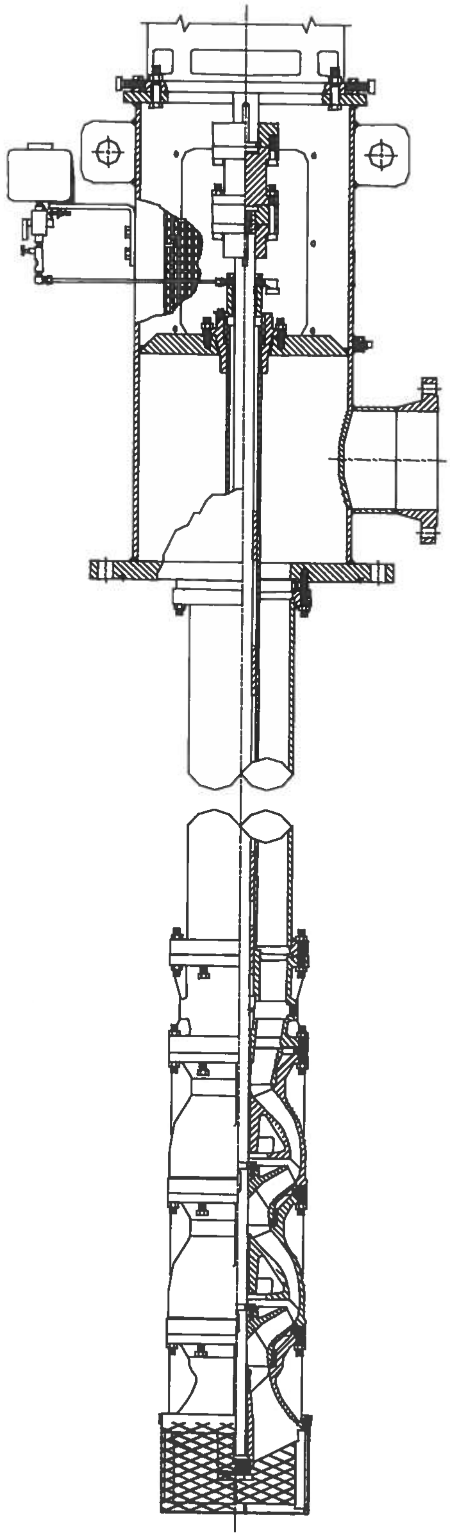

Figure 2.3.3.4b — Enclosed line-shaft, oil-lubricated, vertical turbine pump

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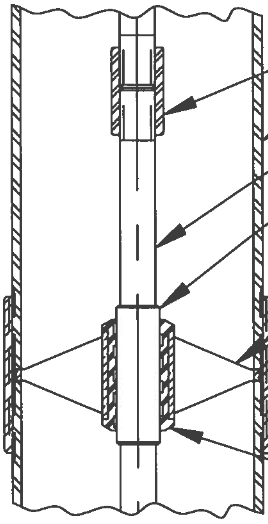

Figure 2.3.3.4c — Open line shaft

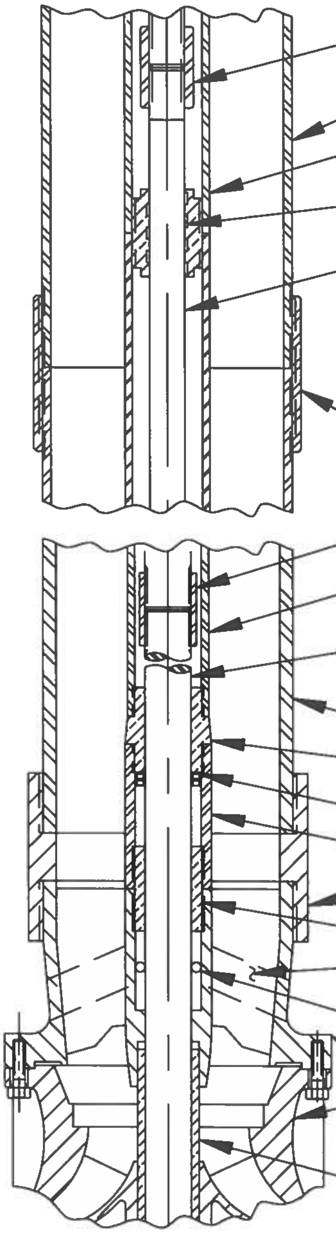

Figure 2.3.3.4d — Enclosed line shaft

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It is important that the lateral mode of shaft vibrations be considered when selecting shaft diameter and bearing spacing for both drive- and head shafts. The natural frequency of the shaft supported between two bearing bushings is calculated. It is often based on the assumption made for calculating the natural frequency of uniform beams. Usually, first and second critical frequency is checked. The lateral natural frequency, F_n, in rpm, is calculated per the following equations.

Metric units:

$$F_n = 0.762 \times \frac{N}{L} \times 83.034 \times \sqrt{\frac{g}{w_L}} \times \left[1.4504 \times 10^{-4} \times E \times I \times \left(7.9797 \times 10^{-2} \times \frac{N}{L}\right)^2 + 0.2249 \times F\right]^{0.5}$$

US customary units: $$F_n = 30 \times \frac{N}{L} \times \sqrt{\frac{g}{w_L}} \times \left[E \times I \times \left(3.1416 \times \frac{N}{L}\right)^2 + F\right]^{0.5}$$

Where:

N = calculated critical frequency mode number: $N = 1$ for first mode; $N = 2$ for second mode

L = shaft length between bearing bushing supports, in m (in)

g = acceleration due to gravity, in m/s^2 (in/s^2)

w_L = shaft weight per unit length, in N/m (lb/in)

E = modulus of elasticity of shaft material, in N/m^2 (lb/in^2)

I = moment of inertia of shaft's cross section, in mm^4 (in^4), $I = 3.1416 \times \frac{d^4}{64}$, d = shaft diameter, in mm (in)

F = axial force on shaft, in N (lb), equal to pump thrust plus weight of rotational components below the shaft location

Using the above equation for enclosed line-shaft natural frequency analysis gives uncertain results because of the enclosing-tube effect.

Most line-shaft bearing bushings are spaced at intervals not exceeding 3.0 m (10 ft). The critical frequency calculations for shaft sizes between 25 mm (1 in) and 40 mm (1.50 in) with 3.0-m (10-ft) spacing comes very close to the range of first critical. However, with good manufacturing practices in preparation of the pump drive-train parts, it has been found by practical experience that pumps of line shaft, water-lubricated rubber bearing bushing construction at 3.0-m (10-ft) spacing operate successfully at 1800 rpm without excessive vibrations. At this same rotational speed, metal bearing bushing open line-shaft construction spacing is typically between 1.5 and 3.0 m (5 and 10 ft). Both enclosed and open line-shaft construction allow increased spacing with lower rpm. At 3000 or 3600 rpm, the bearing spacing has the smallest values. The manufacturer's suggested bearing spacing is to be used.

2.3.3.6 Lubrication systems

One of the most important requirements for a reliable pumping system is adequate lubrication of the bearings under all operating conditions.

Vertical pumps with open line shaft may require prelubrication of any bearing bushing not submerged prior to pump start-up. It is recommended the open line-shaft type should include provisions for bearing prelubrication. This prevents the bearing bushings from burning or seizing before the pumped water reaches them, although there are materials that can withstand momentary dry-start conditions. Postlubrication for spin-down conditions may need to be considered. Consult the pump manufacturer for the recommended prelubrication rate of flow and time required before start-up, and if applicable, postlubrication rate of flow and time for the rotational coast-down.

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Vertical pumps with enclosed line shaft must have provisions for lubrication from an external source. This applies to water, grease, or oil lubrication. A manual or automatic system may be used, providing a compatible liquid is from an external source. An automatic system normally incorporates a solenoid valve in the lubrication line and a time-delay relay in the control equipment.

If the pumped water is corrosive, such as seawater, or laden with abrasives, a pressurized method of lubrication could be used. Filtered water from an external source is delivered to the enclosed line shaft, intermediate bowl, and suction bearings where the flushing fluid pressure must exceed the maximum pressure developed by the pump. Some other enclosed line-shaft pump designs use pressurized grease lubrication.

For some applications, special holes are provided in the pump bowl shaft to conduct lubrication from the upper shaft-enclosing tube to the bowl bearing bushings and the tail bearing bushing, or piping may be used to accomplish the same purpose.

2.3.3.7 Shaft seals

The fluid pumped in most vertically suspended pumps must be sealed around a rotating shaft. This can be accomplished with either a mechanical seal or packing within the discharge head. A means of sealing in the lubricants and sealing out moisture and contaminants must be provided for the bearing housing. This section describes the various types of rotating shaft seals available and their typical applications. Selecting the correct type and materials is based on service desired, leakage tolerable, liquid properties, and temperature. A seal manufacturer with proven application experience should be consulted for critical or special services.

2.3.3.7.1 Typical schematics

Diagrammatic classifications of the more popular types of mechanical seals available are shown in Figures 2.3.3.7.1a and 2.3.3.7.1b. The pump manufacturer will supply mechanical seal details for the specified service conditions.

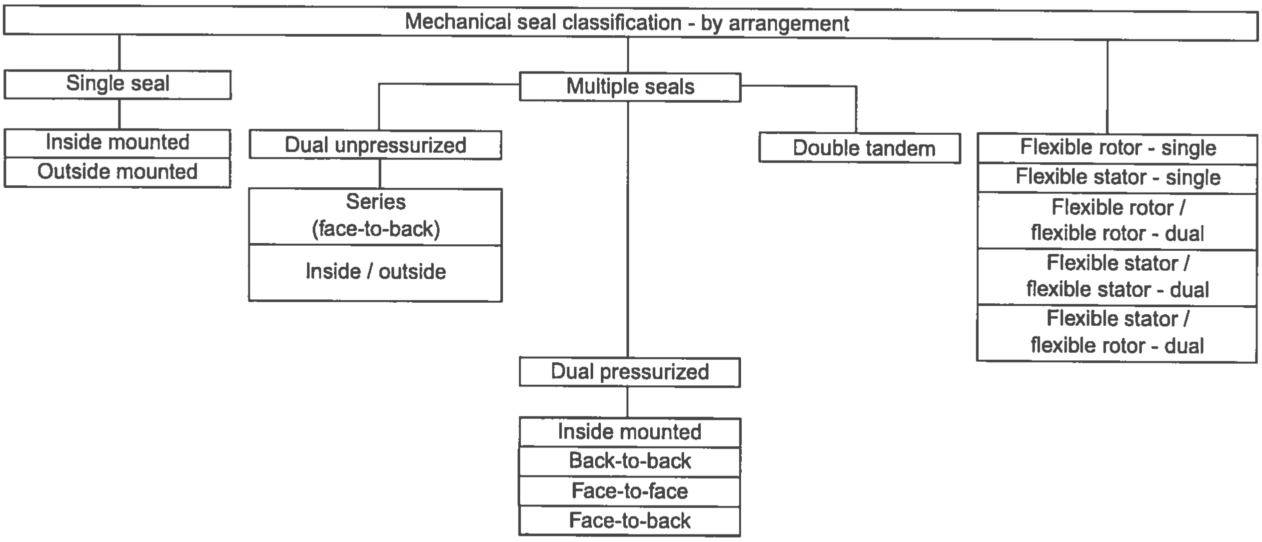

Figure 2.3.3.7.1a — Mechanical seal classification by arrangement

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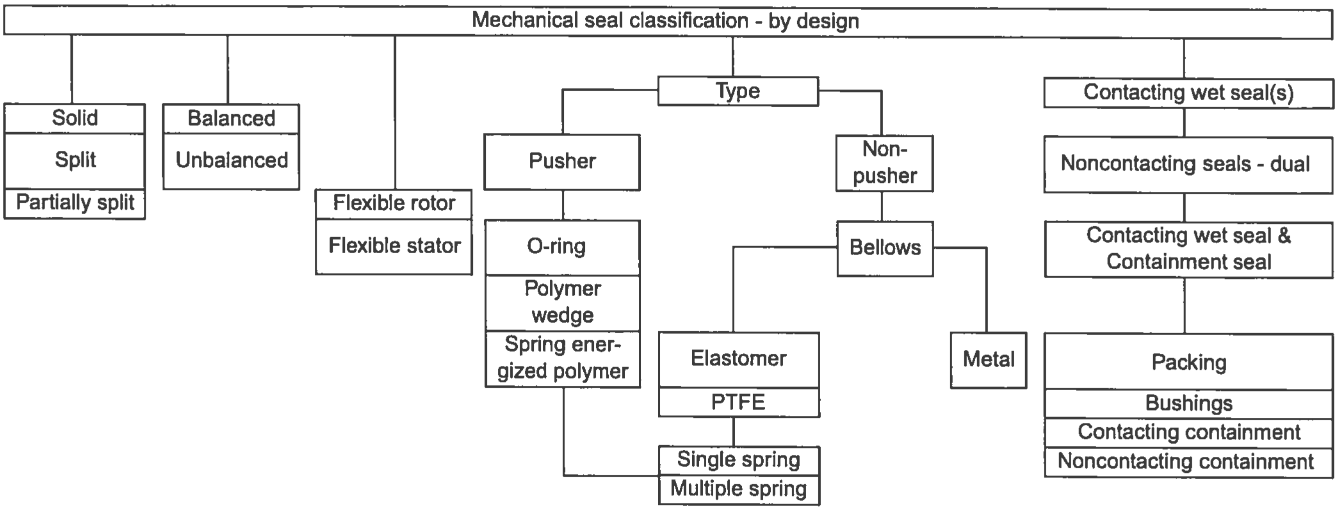

Figure 2.3.3.7.1b — Mechanical seal classification by design

2.3.3.7.2 Application of mechanical seals

Mechanical seals can be applied to virtually any service and are used in many applications where packed stuffing boxes are not suitable. Some of their primary characteristics follow:

a) Very low leakage and longer life than packing.

b) No periodic adjustment as with packing.

c) Capable of sealing at higher pressures and shaft speeds than packing.

d) For all but the simplest low-pressure cooling water pumps, the mechanical seal will have a higher initial cost and will require a much more complete disclosure of the liquid handled. (Provide the pump manufacturer with complete information on the liquid being handled, including the liquid description, suction and discharge pressure, temperature, quantity and type of solids, abrasives, etc.)

e) The piping to seal chambers equipped with mechanical seals is arranged in many ways depending on the type of pump and the application conditions. The application rules for the choice of the flush or circulation piping required are complex, and a user is urged to discuss this point thoroughly with both the pump manufacturer and possibly the mechanical seal manufacturer. More detailed information on flush plans is available in API Standard 682.

The user or specifier should work closely with the pump or seal manufacturer during development of the purchase specifications. Some important points follow:

a) The pump or seal manufacturer can best advise if the mechanical seal specified is the best selection based on the manufacturer's broad knowledge of pump and mechanical seal application history.

b) The pump manufacturer can best advise if the specified seal will fit within the stuffing box or seal chamber.

c) The pump or seal manufacturer is usually in a position to recommend the proper metallurgy of the seal for compatibility with the product pumped. With the many mechanical seal types available, it is recommended that full disclosure of operating conditions be provided to the manufacturer for assistance with the seal construction specifications.

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2.3.3.7.3 Seal chambers

The selection of seal chamber arrangement is based on requirements to maintain a proper environment for the mechanical seal. Factors to consider are the standby, start-up, normal operation, and shut-down conditions; venting of gases and vapors; supply of liquids for flushing, quenching, cooling, bypass, recirculation, and barrier fluid; and provision for drainage.

An elementary arrangement is shown in Figure 2.3.3.7.3a with an external mechanical seal.

Where the seal can operate immersed in the pumped liquid without isolation from suspended abrasives and there is an adequate margin of pressure to prevent vaporization in the seal chamber, an internal seal arrangement such as shown in Figure 2.3.3.7.3b may be used.

For higher pressures, temperatures, and more advanced seal support requirements, use of a special seal such as shown in Figure 2.3.3.7.3c may be indicated. There are many variations on this arrangement, including jacketing for water cooling.

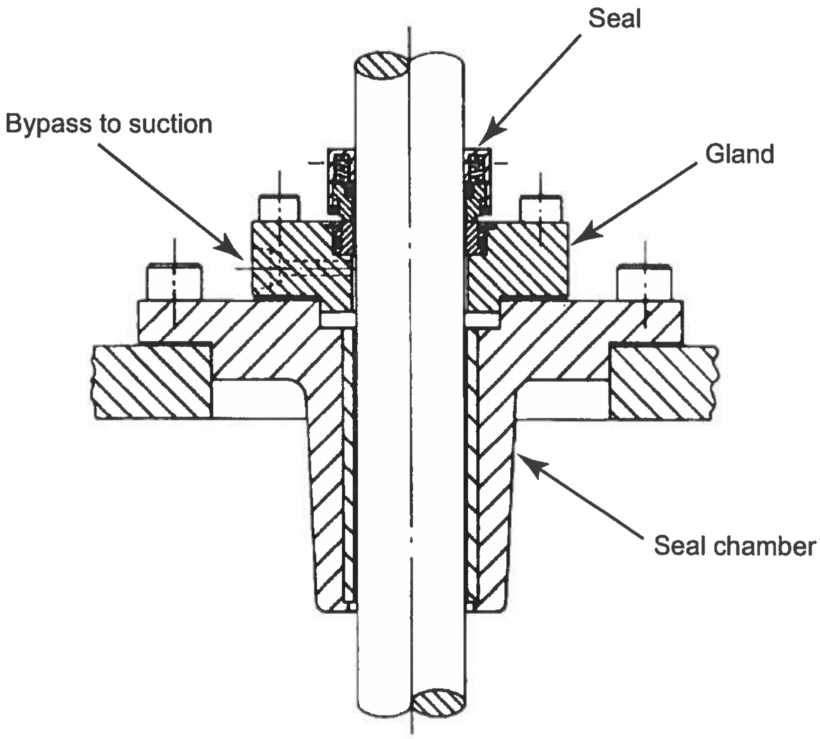

Figure 2.3.3.7.3a — External seal

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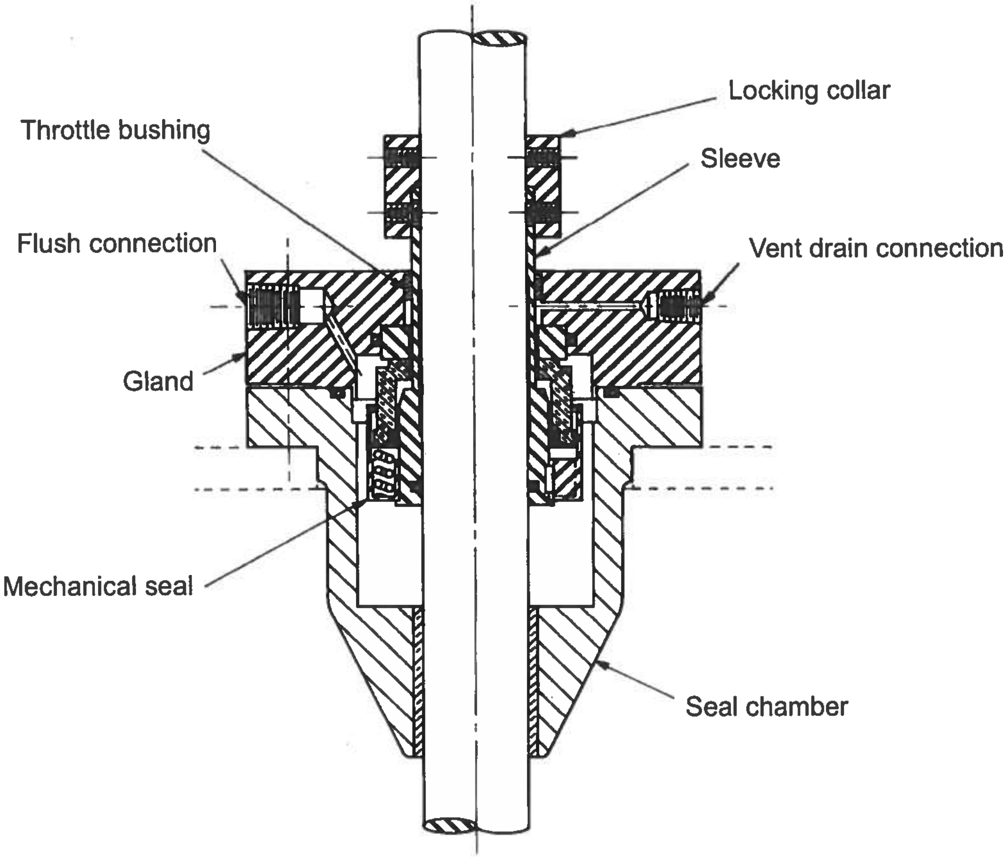

Figure 2.3.3.7.3b — Internal seal arrangement

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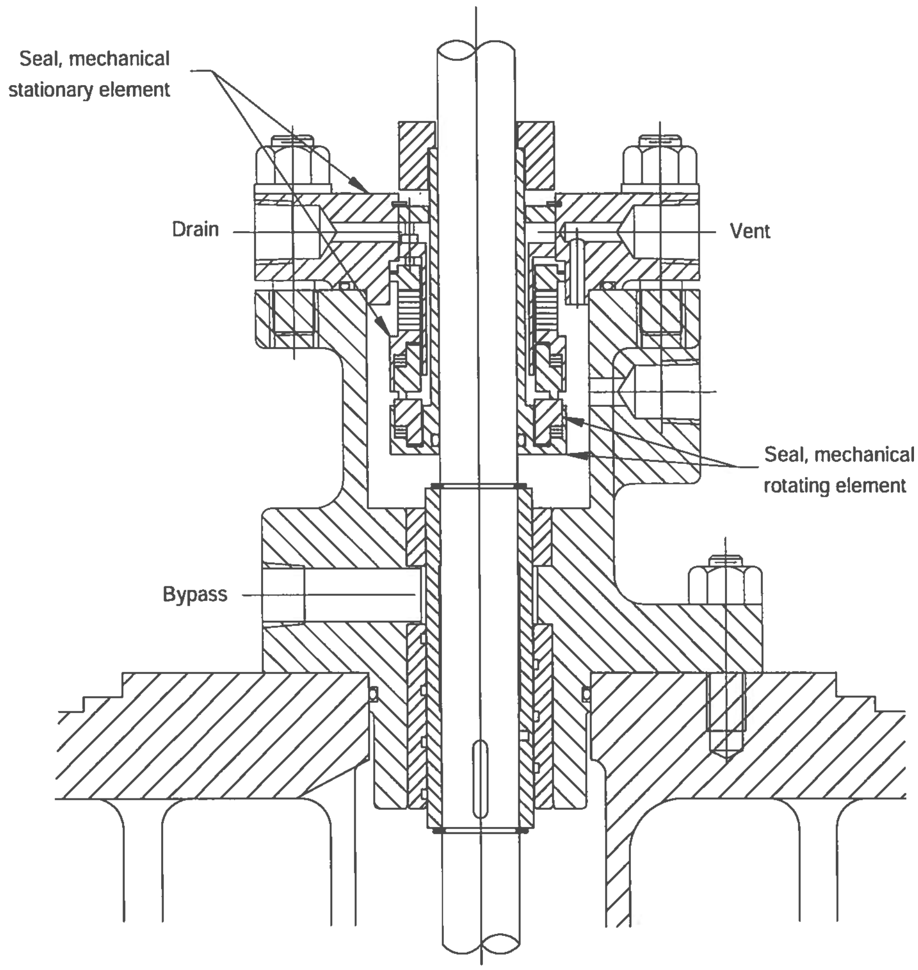

Figure 2.3.3.7.3c — Special seal arrangement

2.3.3.7.4 Packed stuffing boxes

The two most common packed stuffing box configurations for vertical pumps are those with and without lantern rings. Both arrangements have a bushing below the packing. The two figures show these two constructions:

The construction in Figure 2.3.3.7.4a is used where the pump discharge pressure is not high, and where pumped fluid is clean and its leakage to atmosphere is acceptable.

The type of stuffing box in Figure 2.3.3.7.4b is used to provide water injection to the shaft-enclosing tube (inner column). A provision for grease injection to the lantern ring near the lower end of the packing ring stack is optional.

Other special-purpose packed stuffing boxes provide for cooling, throat bushings, quench glands, and other special features for the particular application conditions. Packed stuffing boxes normally are limited to moderate pressures and temperatures and require a slight leakage for packing lubrication and cooling. Care is required in adjusting the packing gland to avoid shaft sleeve and packing damage. The number of packing rings in the stuffing box, together with the size and type of packing vary by manufacturer. In most cases, it is recommended that specifications leave open the exact details about the number of rings or the size or type of packing and allow the pump manufacturer to make recommendations based on application experience.

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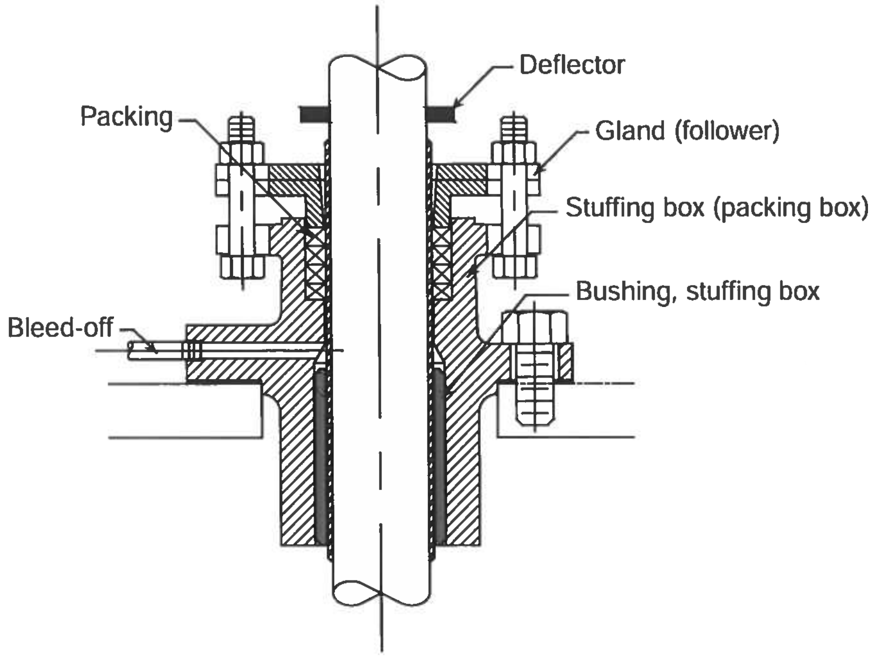

Figure 2.3.3.7.4a — Stuffing box for low to intermediate pressure service

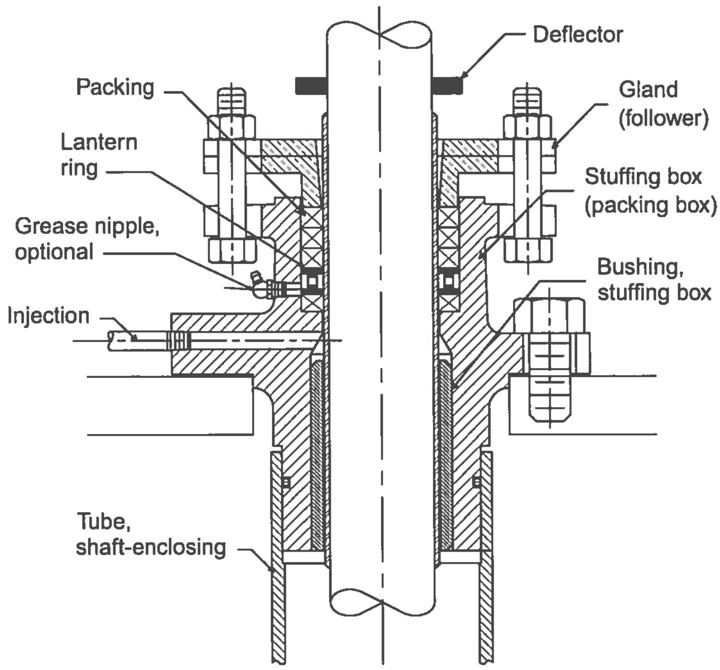

Figure 2.3.3.7.4b — Stuffing box with injection

2.3.3.7.5 Other means of sealing the pumped medium

Alternatives to mechanical seals or packed seal chambers are available. For example, limited-leakage throttle bushings are used for some applications.

A throttle bushing is a device that relies on the close clearance between the shaft or sleeve and the bushing to provide leakage resistance. The leakage is drained back to the suction source. This is essentially a zero maintenance seal, but leakage volume may be excessive, depending on the application conditions.

2.3.3.7.6 Bearing housing closures and isolators

Closures are typically bearing isolators or lip-type oil seals on or near a pump bearing to retain oil or grease and to exclude moisture and foreign material from the bearing space.

Bearing isolators are composed of both a stationary and a rotating component, which act in concert to retain lubricant and exclude contaminants from the bearing housing. The rotor turns with the shaft and, together with the stator, form a difficult path for contaminants. Contaminants that manage to get between the rotor and stator are collected and expelled through an expulsion port by a combination of centrifugal force and gravity drain.

For lip-type seals, an accurate and smooth shaft surface and concentric bore for the closure are requirements for optimum sealing.

2.3.3.8 Nozzle loading

For vertical turbine short set pumps, see ANSI/HI 9.6.2 *Assessment of Applied Nozzle Loads*. It is very important that the discharge piping (and suction piping for VS6, VS7, and VS8 units) be properly anchored to transfer forces and moments to the foundation. The system piping should not exert excessive forces and moments on the pump discharge head. The piping shall not exert forces or moments on the pump discharge nozzles any greater than allowed by ANSI/HI 9.6.2.

For other materials and higher temperatures, the methodology for revising the nozzle loads is presented in ANSI/HI 9.6.2.

2.3.3.9 System piping and foundation

The foundation should be designed to carry the wet weight of the pump, the sole plate, the weight of the driver, the flange nozzle loading, and absorb vibrations generated during operation. The pump supporting foundation shall be a minimum of five times the mass of the entire pump assembly.

The system piping should not exert excessive forces and moments on the pump discharge head. The piping must be fully supported by piping supports and anchors. The piping shall not exert forces or moments on the pump discharge nozzles any greater than those allowed by ANSI/HI 9.6.2. For additional information on bfoundations and foundation bolting, refer to ANSI/HI 2.4 *Rotodynamic Vertical Pumps for Manuals Describing Installation, Operation, and Maintenance*; for suction and discharge piping, refer to ANSI/HI 9.6.6 *Rotodynamic Pumps for Pump Piping*.

2.3.3.10 Structural natural frequencies

Vertical pump types VS1, VS2, VS3, VS6, and VS7 are generally flexible structures above the mounting plate. The above-base structure of a vertical pump is occasionally found to have a lateral mode natural frequency located at running speed. If the pump is operated in a resonant condition (running speed coincident with a structural natural frequency), then excessive vibration can occur. If the pump has a variable-speed drive (VSD), then the probability of an operating speed being located at a structural natural frequency is greatly increased.

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To help prevent the natural frequencies of the upper pump structure from being resonant at operating speed, pump manufacturers may conduct a structural dynamic analysis. For information regarding dynamic analysis, see ANSI/HI 9.6.8 *Dynamics of Pumping Machinery.*

2.3.3.11 Motor/Driver interface

2.3.3.11.1 Vertical pump motor mounting

Vertical pump motors are normally supplied with a rabbet fit mounting base conforming to industry standard dimensions and drilling. It is customary for the vertical pump manufacturer to supply the motor in order to ensure proper mating of the base flanges, proper coupling of the shaft extensions, specification of the axial thrust requirements, and specification of the start-up torque requirements.

Alignment of motor shaft to pump shaft can be affected by the motor base register eccentricity and the mounting base face runout. Care must be exercised in the procurement of the motor to ensure proper motor-to-pump interface requirements are met. Usually the pump manufacturer takes responsibility for procurement of the motor to ensure the various design interface issues are covered.

The National Electrical Manufacturers Association's (NEMA) Standard MG 1 and the International Electrotechnical Commission's (IEC) Standard 72-1 define motor base mating dimensions for vertical pump motors.

2.3.3.11.2 Thrust bearings

Vertical pump bowl assembly units are not typically designed to balance the unequal pressure forces acting on the impellers. Although this maximizes efficiency, it also leaves a high resultant axial force, which must be supported. The magnitude of this axial force depends on the pump design and size, as well as the condition of service, such as rate of flow and pressure. The pump manufacturer normally supplies the value of this axial thrust, which must be supported by the thrust bearings. These thrust bearings may be in the driver or gear, or in an integral thrust bearing (also called the *thrust pot*) installed between the pump discharge head and driver, or a separate thrust bearing is installed above or below the bowl assembly. Regardless of the location of the thrust bearing, with nonvertical shaft orientations, the manufacturer should be consulted to ensure satisfactory loading and life of the thrust bearing.

Typical practice is to require the motor bearings to handle momentary upthrust capabilities (approximately 30% of normal downthrust bearing rating). When pump is equipped with an axial thrust bearing it should be designed to handle momentary upthrust. Thrust bearing construction normally uses angular contact ball, spherical roller, or tilting pad hydrodynamic-type bearings. Life expectancy, maintenance, and design considerations normally dictate the type, size, and location of the thrust bearing used. Typical life expectancy for antifriction bearings is normally based on 17,500 hours (L_{10}) at rated condition. The pump manufacturer should be consulted for its recommendations. For more information, see ANSI/HI 1.3 *Rotodynamic Centrifugal Pumps for Design and Application*.

2.3.3.11.3 Anti-reverse rotation devices

Reverse rotation of vertical pumps may occur if the pump is subjected to flow reversal, on stand-by, shut-down, or on start-up when the motor leads are connected improperly. Damage may occur due to reverse overspeed (see Section 2.3.4.6) or unscrewing of threaded line-shaft couplings. Anti-reverse rotation devices (such as nonreverse ratchets) are used to protect the pump and the driver from potentially damaging reverse rotation conditions.

However, anti-reverse rotation devices will not relieve the stresses due to flow reversal. The torque load developed under reverse flow, locked rotor conditions needs to be checked in order to properly size rotor components and the anti-reverse rotation device. Such devices may also be used to reduce the pump starting torque under reverse flow conditions (see Section 2.3.4.4).

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2.3.3.11.4 Pump-to-driver shafting

Vertical pump drivers are either of solid-shaft or hollow-shaft construction; see Figures 2.3.3.11.4a and 2.3.3.11.4b.

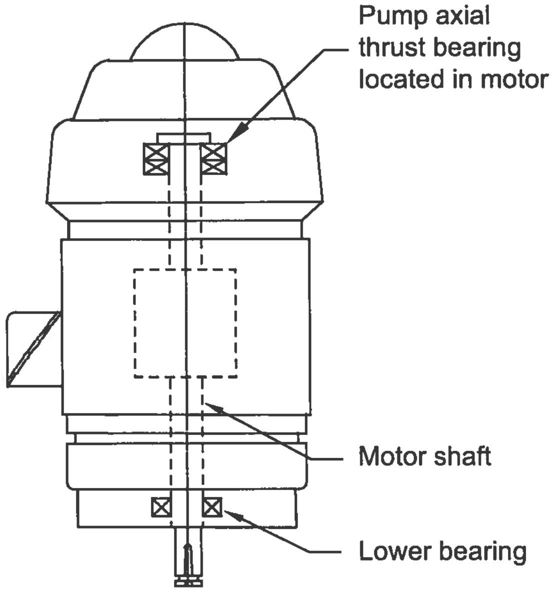

Figure 2.3.3.11.4a — Solid-shaft motor

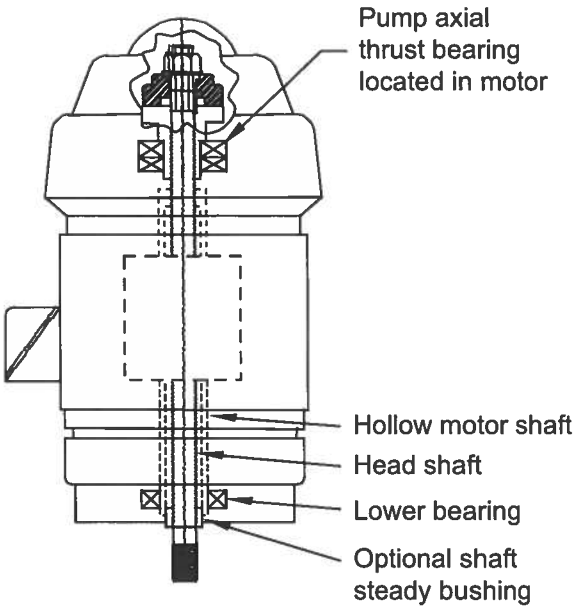

Figure 2.3.3.11.4b — Hollow-shaft motor

2.3.3.11.4.1 Solid-shaft configuration

Solid-shaft drivers are connected to the pump through either a rigid or a flexible coupling. Rigid couplings must transmit torsional and axial loads, maintain shaft alignment, and permit adjustment of the shaft as required to set the desired axial running clearance for the impellers. Pumps of VS8 type with tight rotor end float clearances may require motors specified with limited rotor movement from the pump axial thrust.

The driver lower bearing (usually rolling element type) in conjunction with the rigid coupling provides pump shaft support for mechanical seals. When precision manufactured, this coupling is not normally dynamic balanced. The rigid flanged-type coupling is capable of transmitting both axial up- and downthrust. This style coupling is preferred when pump shaft axial upthrust occurs. Flexible couplings are normally only used with solid-shaft drivers when a separate thrust bearing is provided in or above the pump discharge head and below the flexible coupling. The flexible coupling then only transmits the torque, with the separate thrust bearing providing the axial and radial shaft support and allowing for the required axial adjustment of the rotating element.

2.3.3.11.4.2 Hollow-shaft configuration

In the hollow-shaft configuration there is no external shaft extension. The top pump shaft, called the *head shaft*, extends through the driver shaft (also known as the *quill*), which is hollow, and is coupled at the top with a key and adjusting nut arrangement (see Figure 2.3.3.11.4.2). The coupling is located within the motor, under the motor's drip cover. This coupling permits the shaft to be adjusted to compensate for the tolerance stack-up of the pump rotor and casing components and to provide the desired axial running clearance for the impellers. This clearance is normally specified by the manufacturer and is determined by mechanical and efficiency considerations, as well as thermal and pressure elongation expectations of the column and shafting.

This hollow-shaft arrangement provides optimum access to the head shaft adjusting nut where impeller lift is sensitive, such as on deepwell pumps. A single-piece head shaft is typically used on outside deepwell pump installations where headroom clearance for disassembly is not limited.

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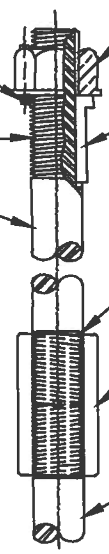

Figure 2.3.3.11.4.2 — Head shaft coupling, rigid style, for hollow-shaft motor

Hollow-shaft motors are coupled to the pump head shaft with either a self-release or bolted coupling.

A self-release coupling is a device that can be used to keep a pump's line shaft from unscrewing, due to torque reversal, and protect the driver from damage. Should a torque reversal occur, such as might take place if motor leads were wired incorrectly, the driver coupling will lift and disengage the pump shaft. Self-release couplings should not be used on motors subject to upthrust. Bolted couplings rigidly connect the pump line shaft to the motor.

Bolted couplings will handle upthrust but will not protect a motor or line shaft in case of torque reversal. Lock screws usually secure the threaded adjusting nut to the driver rotor flange (clutch). The lock screws usually provide limited pump shaft upthrust protection.

A bottom steady (quill) bushing option is normally offered with hollow-shaft drivers to provide added shaft support. This bottom bushing is often recommended with two-piece head shafts, long one-piece head shafts, or to solve head shaft vibration problems.

2.3.3.11.5 Couplings

The application of couplings for the vertical close-coupled or deepwell pump can be unique. There are several common shaft and bearing configurations, each with their own type of coupling requirements.

If the driver shaft and the pump shaft are both supported separately in what can be called a *two-bearing system*, then a full-flex coupling is needed. It can be elastomeric or metallic. Pumps with thrust pots (thrust bearing assemblies) mounted between the shaft seal and the driver require a full-flex coupling. The coupling choice then becomes similar to that of the horizontal centrifugal pump. In these cases the selection of the coupling beyond torque, speed, duty cycle, and misalignment, is one of recognizing the environment. The environment includes the usual temperature, moisture, indoor or outdoor, type of industry, and local preferences. The coupling is not used to transfer thrust loads from driver to pump or vice versa. Misalignment comes from a tolerance stack up between the pump column, mounting plate, driver support, and driver base. A NEMA Type C face or Type D flange motor may be used. It should be noted that these pumps often have a semi-open impeller and need to have shaft axial adjustment to maintain clearance and efficiency. The coupling should be capable of axial movement through its shaft mounting, see Figure 2.3.3.11.5a, or meshing of teeth as in a gear coupling. Spacer couplings, see Figure 2.3.3.11.5b, are used for seal removal in some cases.

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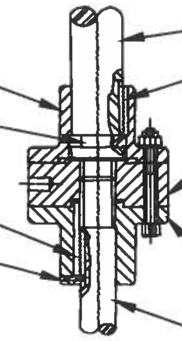

Figure 2.3.3.11.5a — Flanged adjustable coupling, rigid style

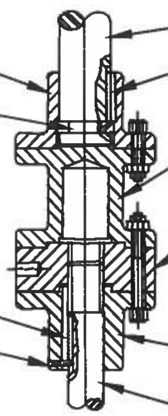

Figure 2.3.3.11.5b — Flanged adjustable spacer coupling, rigid style

If the pump rotor is suspended or (in the case of VS8 pump types) supported by the motor bearings, a rigid coupling is used, such as shown in Figures 2.3.3.11.5a and b, and Figure 2.3.3.11.4.2. The rigid coupling can be of the adjustable type. The semi-open impeller version would use the rigid adjustable coupling that can be adjusted for elongation of the shaft and deflection of the bearings to set impeller clearance off of the bowl seat. This coupling will also be able to carry thrust loads. These couplings are built as industrial units, high-speed units, and API 610 units. Also in this rigid adjustable category are the hollow-shaft motor units with a one-way drive coupling and impeller adjusting nut at the top of the motor.

When vertical pumps are used with a right-angle gearbox there can be couplings on both the horizontal input shaft and the vertical output shaft. The gearbox is treated like the vertical motor in many applications with respect to the shafting connection options as described above. The coupling that connects the driver to the gearbox is treated in a similar manner to that of a horizontal pump with driver. As in those applications, diesel engine drives require consideration of the torsional vibration, and *U*-joints can add to the bearing loads.

2.3.4 Pump performance, selection criteria

2.3.4.1 Pumping system requirements

A pumping system is composed of the piping, valves, vessels, flow measuring equipment, and any other conduits through which the liquid is flowing.

For successful pump operation, the pump and the system components must be properly matched to each other.

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The requirements and the characteristics of the system must be determined before the pump can be selected. Modifications to the system may be needed to achieve overall compatibility. Consideration should be given to changes in the system over a period of time if operating conditions change.

In open pit and well installations, the variation or drawdown of water level must be considered as these affect submergence of the pumping element.

2.3.4.2 Pump versus system curves

A typical simple system curve and pump curve are shown in Figure 2.3.4.2. Note that the pump always operates at the intersection point of the pump curve and system curves.

With more complicated systems, the static head will vary as the suction and discharge liquid levels, or pressures, change. Friction head will be affected by changes in pipe condition. Similarly, the pump curve will change if the pumps are operated at variable speed, or several pumps are operated simultaneously.

Pump discharge piping with siphon and no valve, typical of low-head pump installations, will require higher start-up head than duty head, and probably higher start-up power than duty power.

All these changes will generate new intersection points of the pump and system curves. A complete plot of these curves is a very useful tool for the system designer to determine the total pump operating range.

It should be noted that most manufacturers' ratings curves are based on the bowl assembly performance. The pump column losses and discharge elbow loss (see Section 2.3.4.20) should be included in the pump performance curve.

2.3.4.3 System pressure limitation

The system must be capable of withstanding the pressure at the operating conditions as well as at other transient conditions that may be reasonably expected. If the system is equipped with a discharge shut-off valve, then the piping should be designed for pump shut-off pressure or protected with a pressure-relief valve of adequate capacity.

The possibility of pressure surges in the system must also be considered, as discussed in more detail in Section 2.3.4.5.

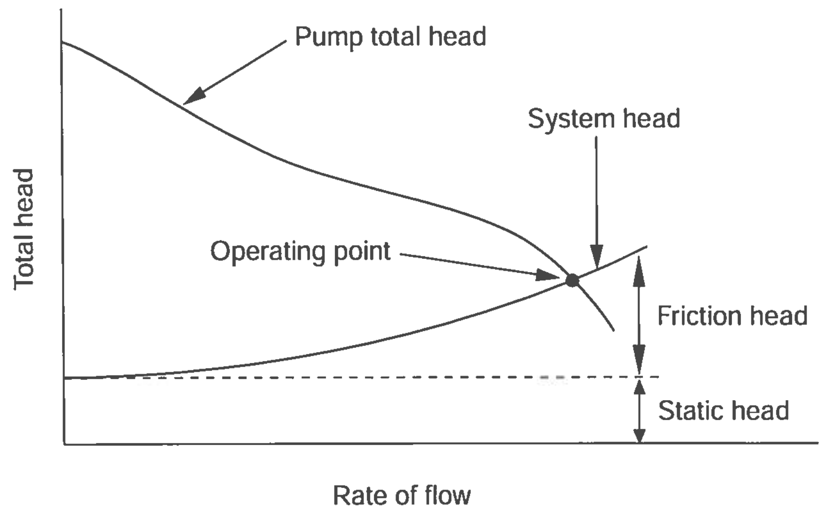

Figure 2.3.4.2 — Pump performance versus system curve

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2.3.4.4 Reverse runaway speed

A sudden power and/or check valve failure during pump operation against a static head will result in a flow reversal, and the pump will operate in the opposite direction of rotation from that of normal pump operation.

If the pump is driven by a prime mover offering little resistance while running backwards, the reverse speed may approach its maximum consistent with zero torque. This speed is called *runaway speed*. The runaway speed may exceed that of the corresponding normal pump operation. This excess speed may impose high mechanical stresses on the rotating parts both of the pump and the prime mover and, therefore, knowledge of this speed during initial design work is essential to safeguard the equipment from possible damage. Refer to ANSI/HI 2.4 *Rotodynamic Vertical Pumps for Manuals Describing Installation, Operation, and Maintenance* and Section 2.3.3.11.3 Anti-reverse rotation devices, in this standard.

2.3.4.5 Water (hydraulic) hammer analysis

Water hammer is an increase in pressure due to rapid changes in the velocity of a liquid flowing through a pipeline. This dynamic pressure change is the result of the transformation of the kinetic energy of the moving mass of liquid into pressure energy. When the velocity is changed by closing a valve, starting or stopping a pump, or by some other means, the magnitude of the pressure produced is frequently much greater than the static pressure on the line, and may cause rupture or damage to the pump, piping, or fittings. This applies to both horizontal and vertical pump installations.

The head due to water hammer in excess of normal static head is a function of the rate of change of velocity, the size and length of the pipe, and the velocity of pressure wave along the pipe. The magnitude of water hammer can be calculated with a fair degree of accuracy by an engineer thoroughly experienced in this work, provided all of the factors influencing the potential water hammer condition are known. It is recommended that specialized engineering services be engaged for such calculations, since few pump users or pump manufacturers have the knowledge and experience necessary for this work.

Water hammer may be controlled by regulating valve closure time, reducing pump acceleration or deceleration, or by application of relief valves, surge chambers, and other means.

The installation of an air-vacuum release valve may be necessary when a vertical pump will have to drive air out of the column when starting. When the pump is shut down, the air-vacuum release valve admits air and allows the water in the column to flow backward through the pump. Vertical pumps whose length from discharge centerline to sump liquid level is greater than atmospheric pressure, expressed in terms of head or liquid column height, typically 10 m (34 ft), can produce a vacuum vapor or air pocket in the column on shut-down, regardless of whether the discharge valve is a slow-opening gate valve or a check valve. To prevent potentially damaging water hammer surge and reaction forces, the pump should not be started with vacuum in the column. The formation of vacuum on shutdown can be reduced in VS0 style pumps by the proper placement of check valves in the drop pipe (the downward piping extension attached to the suction stage of the pump). If the pump discharge is open to the atmosphere, then an air-vacuum release valve is not necessary.

2.3.4.6 Start-up and shut-down analysis

During start-up, the driver must provide adequate accelerating torque (the additional torque required above normal operating torque) to ensure a successful start in a reasonable amount of time. Because torque is directly related to power, the shape of the pump input power curve becomes very important. If maximum input power occurs in the normal operating range, a driver sized for the operating range is normally capable of starting the pump. However, if maximum power occurs at shutoff, which is common for high specific speed pumps, the starting requirements become critical for proper driver selection. In a system with one pump or when starting the first of several units in parallel, the unit can be started with the discharge valves open, or partially open, to avoid operation at shutoff. For high specific speed pumps operating in parallel, there are several options when starting the second and succeeding units. Possible extreme starting procedures range from starting against an open discharge valve at full reverse speed to starting against a closed discharge valve. Either condition requires attention to selection of driver and

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pump shafting. Normally, an in-between starting procedure is selected, where the discharge valve is synchronized to start opening prior to, at the same time as, or after the driver is started. The problem then becomes selecting the optimum procedure and time interval for valve opening.

Some of the parameters to consider are:

- The required valve opening time.
- The length and diameter of the piping system to determine the effect of the water in the system.
- The rotational inertia or WK^2 or WR^2, in newton-meters squared (pound-feet squared) of the pump and driver.
- The four quadrant performance characteristics (Karman–Knapp diagram) of the pump.
- The speed–torque, speed–time, speed–current, and safe time–current characteristics of the motor (if motor driven).
- The system head curve, the number of pumps in parallel, and the available starting voltage.
- The resulting analysis will not only ensure that the driver is capable of starting the pump, but can result in a lower initial capital investment and higher operating efficiency.
- When vertical pumps are started with discharge valves closed, a provision should be made to rapidly vent the column and the pump head to make sure the line-shaft bearing bushings are lubricated and air is not compressed and then suddenly allowed to expand.
- Avoidance of water hammer is the primary concern during the shut-down of a pump, especially in installations with long piping. Gradual closing of the discharge valve is one way to eliminate or reduce water hammer. A mathematical system analysis may be required in some installations to compute the severity of the water hammer.
- The possibility of the pump running in reverse direction after a shut-down must also be considered. Where the system makes it a likely occurrence, the driver and the pump must be designed for the maximum reverse rotation speed. Provisions may be made in the driver (anti-reverse rotation device) or in the system (check valves, rapidly closing shut-off valves, siphon breakers) to prevent reverse rotation.

2.3.4.7 Pump and motor speed torque curves

A plot of speed versus torque requirements during the starting phase of a pump is sometimes checked against the speed versus torque curve of the driving motor. The driver may need to be capable of supplying more torque at each speed than that based on corrected values from full speed, BEP conditions in order to bring the pump up to rated speed. This is usually easily achievable with standard induction motors, but under certain conditions, such as high specific speed pumps or reduced voltage starting, a motor with high starting torque may be required. Synchronous motors typically have less starting torque than an induction motor, and speed versus torque characteristics should be reviewed.

When rotodynamic pumps in the low to medium specific speed range (under n_s = 68 [3500]) are started with the discharge valve closed, the procedure used to calculate the minimum torque requirements at various speeds under this condition is as follows:

Determine the maximum pump power input required at rated speed under shut-off conditions. Convert this power to torque.

Torque of the pump varies as the square of its speed.

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At zero speed, the torque would theoretically be zero, but the driver must overcome mechanical seal (or gland packing) friction, rotating element inertia, and bearing friction in order to start the shaft turning. This requires a torque at zero speed of from 2 to 15% of the maximum torque at rated speed.

Speed–torque requirements for starting conditions other than closed discharge vary depending on the percentage of static head to total head; the volumetric content of the discharge line; the condition of the discharge line, that is, full, partly full, or empty; and conditions that may change during the starting period, such as the opening or closing of bypass valves. Each of these conditions determines a different torque requirement at any specified speed, which should be obtained from the pump manufacturer when necessary.

Any procedure for transmitting the requirements for pump starting and running torque to the driver supplier needs to be understood and accepted by both the pump supplier and the driver manufacturer. Pump torque requirements are often indicated at two values: OPEN VALVE and CLOSED VALVE. The "OPEN VALVE" condition is the *rated operating point* for the pump. The "CLOSED VALVE" is the value at minimum flow. In most applications pumps are started against a physically closed valve or a check valve, which may require lower torque, but have constraints on operating time at this condition.

NOTE: Speed–torque curves will vary depending on pump specific speed (n_S), however, for most radial flow machines the following speed–torque curve data are representative.

- The starting torque will vary depending on the timing of valve opening and the system head curve. For large pumps and special starting, requirements for the driver need to be reviewed on an individual basis by an engineer.
- For high viscosities (> 500 cP) the breakaway torque becomes more significant, and as the viscosity increases so does the breakaway torque.
- Rotodynamic vertical pump performance test acceptance tolerances are typically unilateral, resulting in values of head and flow above the specified design point, which is accompanied by increased input power. This can result in as much as an additional 10% increase in the required driver power and torque.

The following curves, in Figures 2.3.4.7a and 2.3.4.7b, show how this may be completed prior to ordering all drivers.

Curves may be completed using the examples and the WR^2 or WK^2 values supplied by the pump manufacturer. Limiting conditions may also apply. Reduced-voltage starting requires specific confirmation that any selected motor will be capable of accelerating the pump up to operating speed in an acceptable time frame. Pumps with high viscosity fluids or limited motor inrush limits require review.

Small pumps (<10 kW [13 hp]) with high suction pressures or dual mechanical seals may have higher starting torques than shown. Due to additional drag and breakaway losses caused by the mechanical seals, small drivers must be sized to meet starting requirements.

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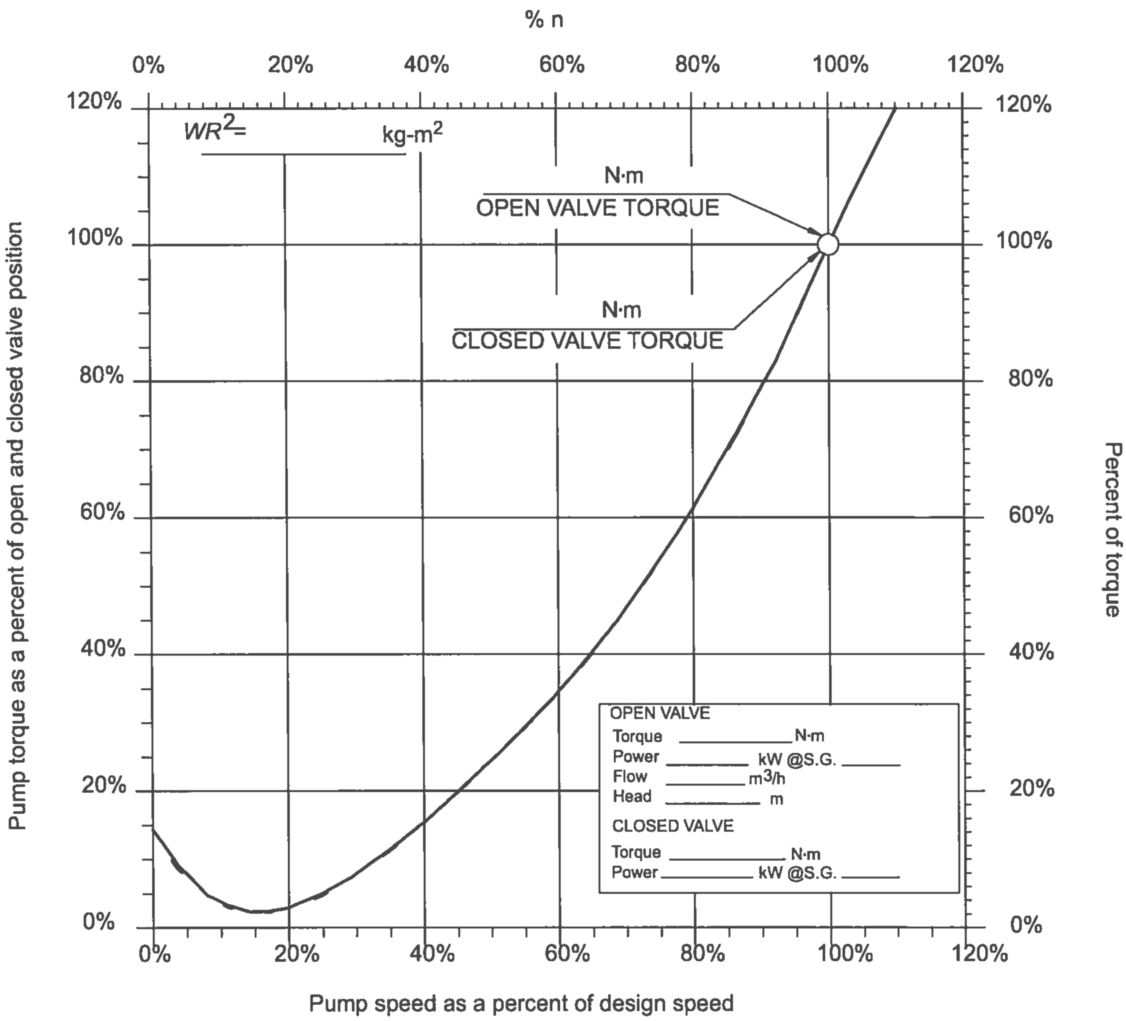

Figure 2.3.4.7a — Torque versus speed - metric units

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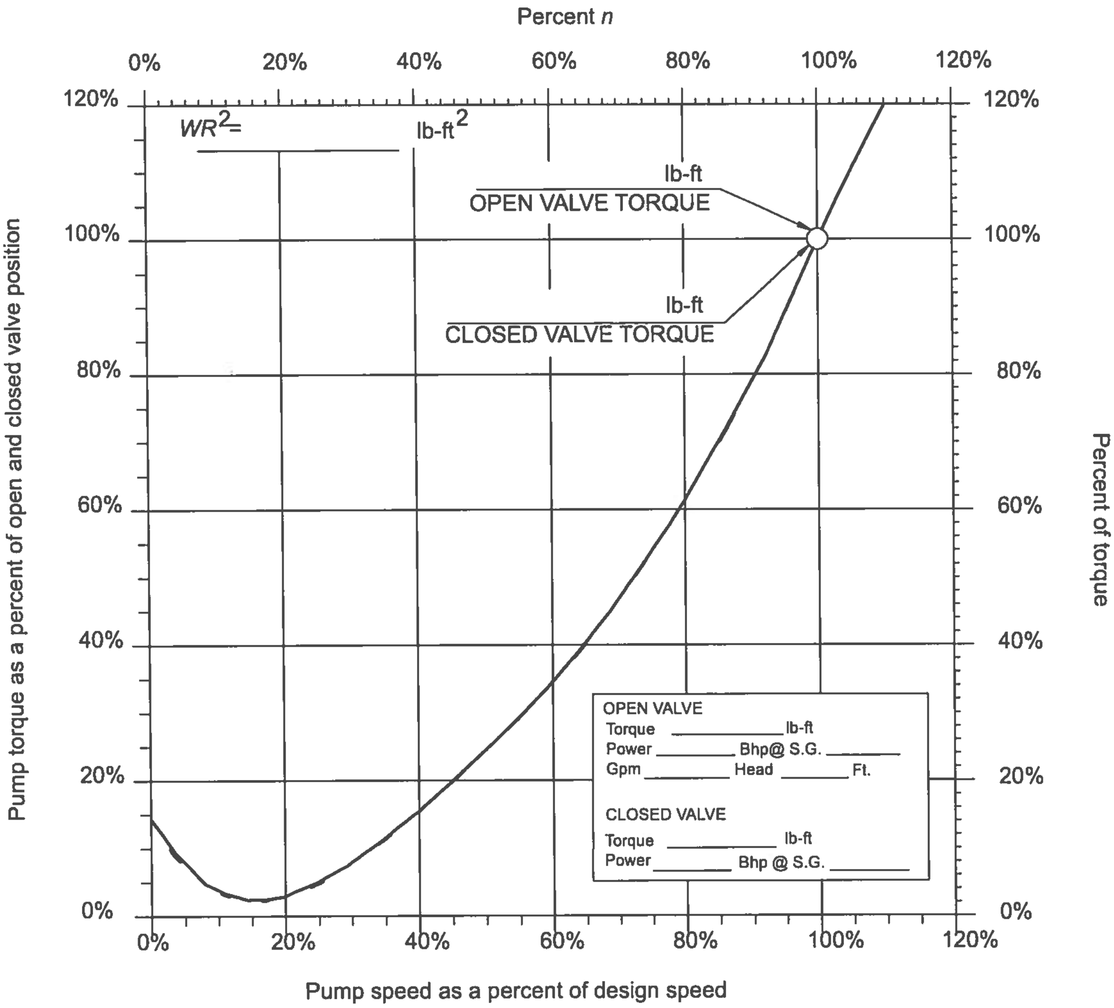

Figure 2.3.4.7b — Torque versus speed - US customary units

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2.3.4.8 Predicting pump performance after speed of rotation or impeller diameter change

A characteristic of rotodynamic (horizontal and vertical) pumps is that it is possible to determine the change in rate of flow, head, and power at any point on the characteristic curves by calculation when there is a change in the speed of rotation. The performance will vary based on the following equations known as *affinity rules*:

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2}$$

$$\frac{H_1}{H_2} = \left[\frac{n_1}{n_2}\right]^2$$

$$\frac{P_1}{P_2} = \left[\frac{n_1}{n_2}\right]^3$$

Where:

Q_1	=	rate of flow at original speed, in m^3/h (gpm)
H_1	=	total head at original speed, in m (ft)
P_1	=	power at original speed, in kW (hp)
n_1	=	original pump speed
Q_2	=	rate of flow at desired speed, in m^3/h (gpm)
H_2	=	total head at desired speed, in m (ft)
P_2	=	power at desired speed, in kW (hp)
n_2	=	desired pump speed, in rpm

This can be most useful for predicting pump performance when applying variable-speed drive controls.

The power relationship is based on criteria that the pump efficiency stays constant with change in speed. However, if the speed of rotation is substantially reduced from original, the relative power loss in bearings and stuffing-box friction may be increased. The hydraulic friction losses may also be relatively increased when the Reynolds number for the water passages is reduced.

These same affinity rules apply for changes in diameter of the pump impeller. For diameter change, substitute in the above equations D_1 for n_1 and D_2 for n_2, where:

D_1	=	original impeller diameter
D_2	=	reduced impeller diameter

NOTE: With respect to impeller diameter change, application of the affinity rules has limitations and does not always accurately reflect the behavior of a pump when the impeller is trimmed. In the specific speed range of 25 [1200] and higher, exponents for diameter ratios in the above equations for flow and head may be different than 1 and 2, respectively, for a given manufacturer's pump design. The affinity rules are not generally useful for high specific speeds above 150 [8000].

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Normally the D diameter is the largest outside diameter (front shroud) of the impeller vanes, but if the impeller exit is machined at an angle, then D shall be the root mean square (RMS) average value between the diameters, or $D = \sqrt{\frac{D_F^2 + D_B^2}{2}}$ (see Figure 2.3.4.8).

If the thickness of the impeller vanes at the outside diameter has been reduced by filing or grinding, the new diameter impeller shall be similarly treated.

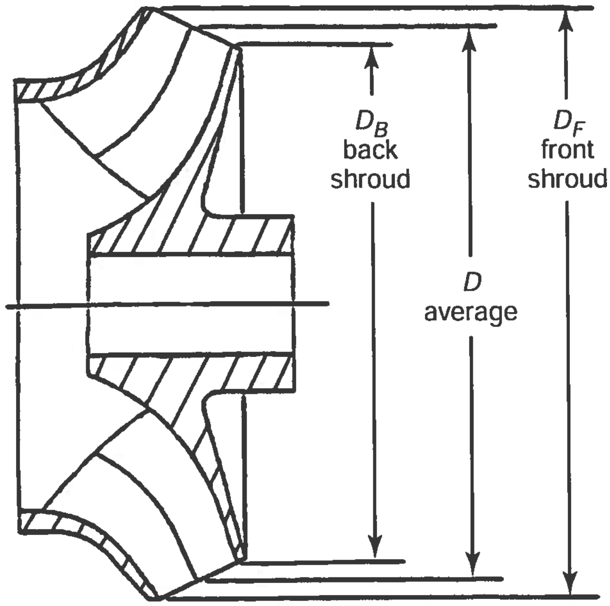

Figure 2.3.4.8 — Impeller with angled outside diameter

Example (metric units) D_1 = 300 mm D_2 = 290 mm

Q_1 = 500 m^3/h H_1 = 100 m P_1 = 45 kW

Q_2 = ? H_2 = ? P_2 = ?

$$\frac{Q_1}{Q_2} = \frac{300}{290} \qquad \frac{H_1}{H_2} = \left[\frac{300}{290}\right]^2 \qquad \frac{P_1}{P_2} = \left[\frac{300}{290}\right]^3$$

$$Q_2 = \frac{500}{1.034} \qquad H_2 = \frac{100}{1.070} \qquad P_2 = \frac{45}{1.107}$$

$$Q_2 = 484 \text{ m}^3\text{/h} \qquad H_2 = 93.5 \text{ m} \qquad P_2 = 40.7 \text{ kW}$$

Example (US customary units) D_1 = 12 in D_2 = 11.4 in

Q_1 = 2000 gpm H_1 = 103 ft P_1 = 63 hp

Q_2 = ? H_2 = ? P_2 = ?

$$\frac{Q_1}{Q_2} = \frac{12.0}{11.4} \qquad \frac{H_1}{H_2} = \left[\frac{12.0}{11.4}\right]^2 \qquad \frac{P_1}{P_2} = \left[\frac{12.0}{11.4}\right]^3$$

$$Q_2 = \frac{2000}{1.053} \qquad H_2 = \frac{103}{1.108} \qquad P_2 = \frac{63}{1.166}$$

$$Q_2 = 1900 \text{ gpm} \qquad H_2 = 93 \text{ ft} \qquad P_2 = 54 \text{ hp}$$

The above equations are not recommended without consulting the pump manufacturer when changing impeller diameters by more than 5%.

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2.3.4.9 Determining operating range, series and parallel operation

Pumps operating in series produce head that is additive at the rate of flow at which they would run individually. Two pumps, each capable of 1000 m^3/h at 50 m of head, when connected in series, could deliver 1000 m^3/h at 100 m of head. Series operation is therefore used where higher pressures are required than the pressures that an individual pump can supply. See Figure 2.3.4.9.

Pumps operating in parallel produce a rate of flow that is additive at the head at which they would run individually. Two pumps, each capable of 600 m^3/h at 35 m, when connected in parallel could deliver 1200 m^3/h at 35 m of head (see Figure 2.3.4.9b).

Pumps used in parallel services should have similar pump curves to ensure the system demand is shared equally.

In all cases it is the system curve that will determine the final operation point. Two pumps operating in parallel will not automatically deliver twice the flow of one pump operating independently.

2.3.4.10 Range of operation

To determine the operating range, one must define the minimum and maximum rate of flow at which the pump must operate continuously. To find these points, determine the intersection of the pump head curves with the system head curves. Figure 2.3.4.10 is an example of two pumps in parallel. The maximum flow always occurs when the pumps are operating at high water level. This means that the pump could operate continuously anywhere between minimum and maximum flow, depending on the suction water level and the number of pumps in operation. However, the possible operating range could exceed the preferred or allowable operating region of one or both pumps as defined by ANSI/HI 9.6.3 *Rotodynamic Centrifugal and Vertical Pumps - Guideline for Allowable Operating Region*. The pump manufacturer can make a recommendation for proper operating region.

2.3.4.11 Continuous, intermittent, and cyclic service

Pumps are designed to operate continuously, intermittently, or in cyclic duty. Each application requires that the pump and driver be carefully selected for the service specified. If the operating conditions are known, then the pump can be designed to meet those conditions. In addition, the driver must also be carefully selected. For example, an electric motor may be limited to only a few starts per hour to prevent overheating.

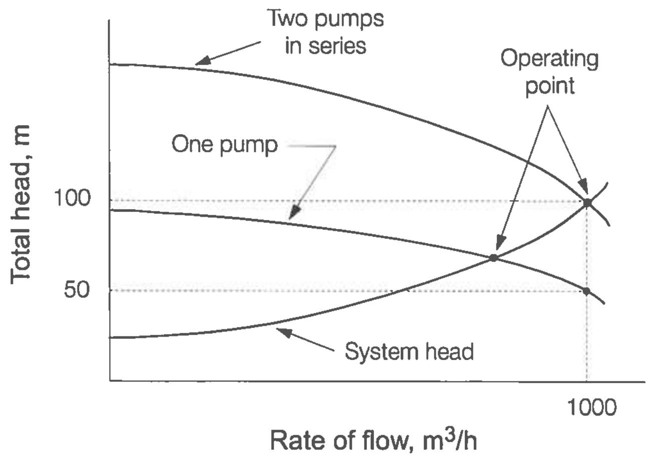

Figure 2.3.4.9a — Series operation

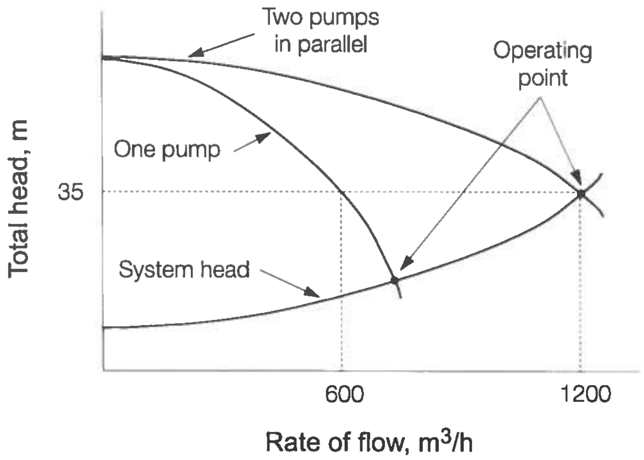

Figure 2.3.4.9b — Parallel operation

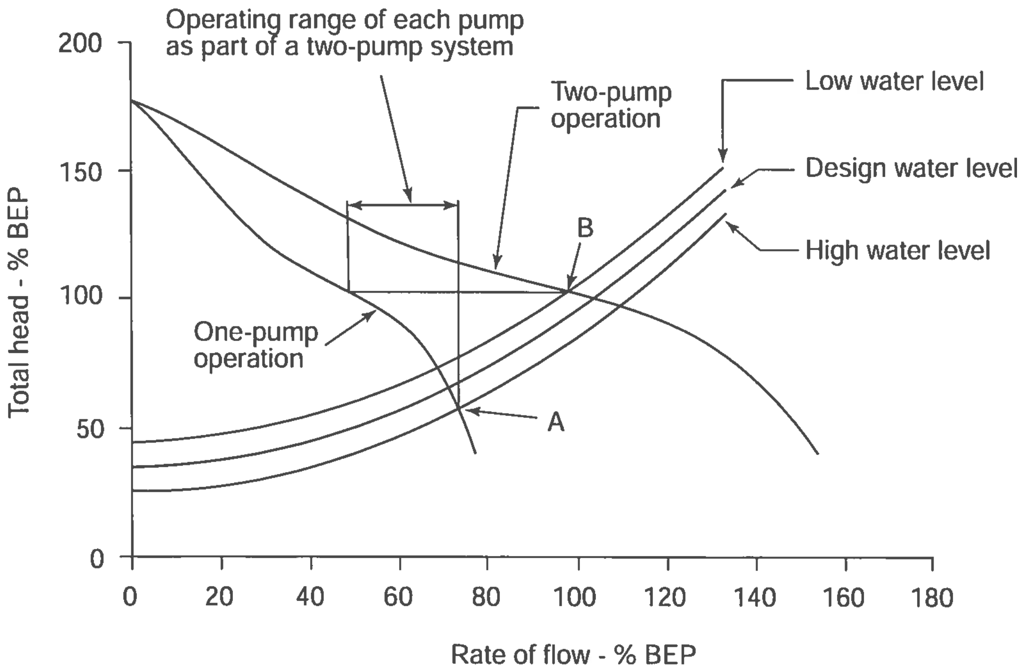

Figure 2.3.4.10 — Operating range

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2.3.4.12 Operation away from BEP flow

A rotodynamic pump is designed for optimum performance at one specific rate of flow at a given speed. This is referred to as the *best efficiency point* (BEP). From an energy consumption standpoint, it is best to operate pumps at BEP flow. However, this is not practical in many systems. Therefore, knowledge of where the pump operates on the curve should establish the feasibility of such operation.

2.3.4.12.1 Minimum and maximum flow limitations

All rotodynamic vertical pumps have limitations on the minimum and maximum flow at which they should be operated continuously or for an extended period of time. Therefore, attention must be given to the operating range selection to avoid problems at off-peak flows. Some causes of problems at reduced rates of flow are as follows:

Temperature rise: Absorption of the input power into the pumped liquid raises the liquid temperature. In general, the temperature rise across the pump should be limited to 8 °C (15 °F) and with a safe margin against flashing.

Suction recirculation: Circulatory flow in the impeller eye at low flow operation can cause localized pitting and mechanical damage. The recirculation onset depends on the impeller inlet design. Continuous operation with suction recirculation should be avoided in high-energy pumps.

Discharge recirculation: Circulatory flow in the discharge area of impellers can cause large forces on impeller shrouds, resulting in random axial unbalance of forces and high thrust. Mechanical vibration and bearing failures can occur. The problem is most severe in high-energy pumps. Typically, vertical turbine pumps with heads greater than 75 m (250 ft) per stage and/or more than 225 kW (300 hp) per stage are considered high energy.

2.3.4.12.2 Operation below the best efficiency point flow

The energy level can be an important consideration for minimum continuous flow, since the destructive forces are greater at high energy levels. Flow limitations as high as 70% of BEP flow may be required in specialized pump applications of high energy levels. Conversely, for normal energy levels, the required flow for continuous operation may be as low as 20% of BEP flow. Consult the pump manufacturer for recommended minimum flow requirements on any specific application.

Net positive suction head: In some designs, the NPSHR by the impeller increases at low flows, and noise, impeller pitting, and other symptoms of cavitation can occur. The pump manufacturer's performance curve should be checked for NPSHR.

2.3.4.12.3 Operation above the best efficiency point flow

When a pump operates at flow beyond the BEP flow, commonly called a *runout* condition, problems can occur due to one or more of the following causes:

Net positive suction head: NPSHR by the impeller increases with flow. Noise, impeller pitting, and other symptoms of cavitation can occur if the NPSHA is inadequate. Consult the pump manufacturer's curve to determine the appropriate NPSHR.

Vortexing: For a given intake structure or sump design, the tendency to vortex increases with flow. A sump design should be evaluated for adequacy at flows greater than the rated flow in order to avoid surface and submerged vortices. Vortices can affect hydraulic performance and create unsteady bearing loads, causing increased wear and vibration.

Flow separation: Cavitation pitting due to flow separation can occur at flows higher than BEP flow.

Vibration: Rotodynamic vertical pumps operate best at the BEP flow condition. Increased vibration is to be expected at rates of flow greater than BEP.

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Upthrust: Some vertical pump designs experience upthrust beyond the BEP flow condition, to the right of the BEP condition, or when starting against low system resistance. Care should be taken to determine the upthrust capability of the pump and that there is adequate upthrust capacity.

During start-up and shut-down, most pumps must operate at shutoff or against a totally open nonpressurized system. From the standpoint of excessive vibration and cavitation, these conditions should be limited to as short a period as possible.

2.3.4.13 Intake design and submergence

Among the most important factors affecting the operation of a vertical wet pit pump are the suction conditions. Insufficient submergence or inadequate NPSHA usually causes a serious reduction in rate of flow and efficiency and often leads to serious trouble from vibration and cavitation.

For a successful installation, special emphasis should be placed on proper sump and can design, along with appropriate materials of construction. The function of the intake structure, whether it is an open channel, a fully wetted tunnel, a sump, or a tank is to supply an evenly distributed flow to the pump suction. An uneven distribution of flow, characterized by strong local currents, can result in formation of surface or submerged vortices and with certain low values of submergence, may introduce air into the pump, causing a reduction of rate of flow, an increase in vibration, and additional noise. Uneven flow distribution can also increase or decrease the power consumption with a change in total developed head.

Submergence is a term used to relate the setting of an immersed pump to the surface level of the suction liquid. The submergence of a pump is the vertical distance from the suction water level surface to the lip of the suction bell. It is a linear dimension partially describing a system and is not a substitute for the characteristic term *NPSHA*. The suction bell must be well below the water surface and the intake or sump must be of a functionally correct design. This is true for all specific speed designs. Minimum submergence is often specified by the pump manufacturer to help prevent vortices. Please refer to ANSI/HI 9.8 *Pump Intake Design* for recommendations.

On large pumping stations, it is strongly recommended to perform model sump tests for assurance of sump design. Higher n_S vertical pumps and larger sizes are more sensitive to off-design conditions. Refer to ANSI/HI 9.8 for recommendations on sump design. These recommendations are general in nature; the pump manufacturer should be contacted for specific applications.

Pump plugging or clogging with debris is a common problem in irrigation and raw water systems. Large pumps taking water from sumps, lakes, rivers, etc., should have trash racks designed into the sump intake structure. Smaller pumps may get by with suction strainers.

2.3.4.14 Suction piping and cans

The suction piping should be arranged to avoid any accumulation of vapor. Provisions should be made at the highest point on the suction side of the pump for venting of vapors to prevent vapor lock. Additional information on suction piping and intake design may be found in ANSI/HI 9.6.6 *Rotodynamic Pumps for Pump Piping* and ANSI/HI 9.8.

Vertical can or barrel pumps (as shown in Figure 2.3.4.14) come in a variety of sizes and shapes. Many can pump arrangements are conceived by the engineer or contractor, either based on very general knowledge or simply fitted into the piping, without properly considering flow patterns at the can inlet or within the can. To avoid potential problems, such as cavitation, submerged vortexes, surges, or other flow instabilities, it is recommended that reference be made to ANSI/HI 9.8 for the intake piping and can design.

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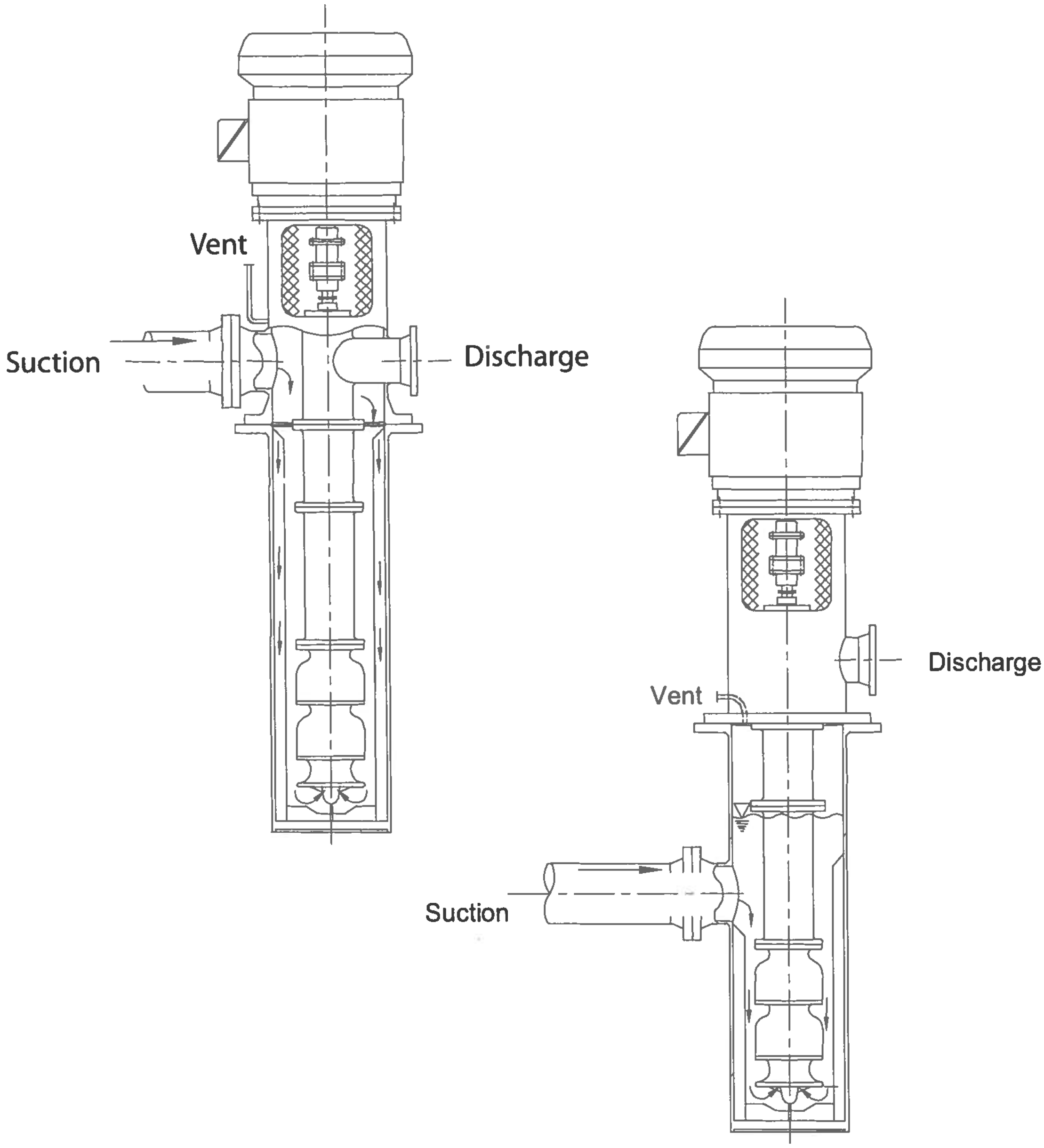

Figure 2.3.4.14 — Suction cans (barrels) with vents

2.3.4.15 Net positive suction head available (NPSHA)

Net positive suction head available (NPSHA) is the total suction head in meters (feet) of liquid absolute corrected to datum less the vapor pressure of the liquid in meters (feet). Therefore, NPSHA is the pressure or head available above vapor pressure to move and accelerate the fluid into the impeller inlet:

$$NPSHA = h_{sa} - h_{vp}$$

Where:

h_{sa} = total suction head, in m (ft) absolute

h_{vp} = vapor pressure of pumped liquid, in m (ft)

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or: $h_{sa} = h_a + Z_s - h_f$

Then:

$$NPSHA = h_a + Z_s - h_f - h_{vp}$$

Where:

h_a = absolute pressure on the surface of the liquid where the pump takes suction, in m (ft) of liquid. In an open system, h_a equals atmospheric pressure.

Z_s = static elevation of the liquid above the datum point of the pump, in m (ft). If the liquid level is below the pump datum, Z_s is a negative value.

h_f = friction and entrance head losses in the suction piping, in m (ft). If suction piping is not used, $h_f = 0$.

When the absolute vapor pressure is expressed in psia, the following formula may be used:

Metric units: $$NPSHA = \frac{1000}{\rho g}(p_a - p_{vp}) + Z_s - h_f$$

US customary units: $$NPSHA = \frac{144}{\gamma}(p_a - p_{vp}) + Z_s - h_f$$

Where:

p_a = absolute pressure, in kPa (psia). In an open system, p_a equals atmospheric pressure.

p_{vp} = vapor pressure, in kPa (psia)

ρ = density, in kg/m^3 (lbm/ft^3)

g = gravity constant, in m/s^2 (ft/s^2)

γ = specific weight of liquid at the pumping temperature, in lb/ft^3

If a pump takes its suction from a source where the absolute pressure on the surface of the liquid (p_a) is equivalent to the vapor pressure (p_{vp}), then the NPSHA is the difference in elevation between the liquid level and the datum, minus the entrance and friction losses in the suction piping:

$$NPSHA = Z_s - h_f$$

The formulae shown above are commonly used for determining the NPSHA for proposed installations and for measuring the NPSHA in existing installations without suction piping. The formula commonly used for measuring the NPSHA in existing installations with suction piping is as follows:

$$NPSHA = h_{atm} + h_g + \frac{v^2}{2g} + Z_s - h_{vp}$$

Where:

h_{atm} = atmospheric pressure, in m (ft) absolute

h_g = gauge head at the suction of the pump, in m (ft) of liquid. h_g is a negative value if it is below atmospheric pressure.

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$\frac{v^2}{2g}$ = velocity head at the point of measurement of h_g (this is necessary because gauge readings do not include the velocity head)

Z_s = distance between suction datum and suction gauge

2.3.4.16 NPSHA corrections for temperature and elevation

NPSHA is a function of the absolute pressure and the vapor pressure. In an open system, the absolute pressure is in turn a function of the elevation, and the vapor pressure varies with the temperature. The following are some examples of NPSHA calculations for open systems:

Example (metric units): Applications with water in an open system at sea level with a pumping temperature of 20 °C are common. Given ρ = 1000 kg/m^3 and g = 9.8 m/s^2.

p_a = 100 kPa

p_{vp} = 4.1 kPa

Z_s = 3 m

h_f = 0 m

We find:

$$NPSHA = \frac{1000}{\rho\, g} \times (p_a - p_{vp}) + Z_s - h_f$$

$$= \frac{1000}{1000 \times 9.8} \times (100 - 4.1) + 3.0 - 0$$

$$= 12.8 \text{ m}$$

To find the NPSHA for water of 82 °C temperature (p_{vp} = 51.8 kPa and ρ = 970 kg/m^3), proceed as follows:

$$NPSHA = \frac{1000}{\rho\, g} \times (p_a - p_{vp}) + Z_s - h_f$$

$$= \frac{1000}{970 \times 9.8} \times (100 - 51.8) + 3.0 - 0$$

$$= 8.1 \text{ m}$$

To find the NPSHA for 82 °C water at 1525 m elevation (p_a = 84.5 kPa), proceed as follows:

$$NPSHA = \frac{1000}{\rho\, g} \times (p_a - p_{vp}) + Z_s - h_f$$

$$= \frac{1000}{970 \times 9.8} \times (84.5 - 51.8) + 3.0 - 0$$

$$= 6.4 \text{ m}$$

The correction for elevation is approximately 1 m per 1000 m of elevation.

Example (US customary units): Applications with water in an open system at sea level with a pumping temperature of 85 °F are common. Given γ = 62.4 lb/ft^3.

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p_a = 14.7 psi

p_{vp} = 0.6 psi

Z_s = 10.0 ft

h_f = 0

We find:

$$NPSHA = \frac{144}{\gamma}(p_a - p_{vp}) + Z_s - h_f$$

$$= \frac{144}{62.4}(14.7 - 0.6) + 10.0 - 0.0$$

$$= 42.5 \text{ ft}$$

To find the NPSHA for water of 180 °F temperature (p_{vp} = 7.51 psia and γ = 60.53 lb/ft^3), proceed as follows:

$$NPSHA = \frac{144}{\gamma}(p_a - p_{vp}) + Z_s - h_f$$

$$= \frac{144}{60.53}(14.7 - 7.51) + 10.0 - 0.0$$

$$= 27.1 \text{ ft}$$

The correction for elevation is approximately 1 ft per 1000 ft of elevation. To find the NPSHA for 180 °F water at 5000 ft elevation (p_a = 12.25 psi), proceed as follows:

$$NPSHA = \frac{144}{\gamma}(p_a - p_{vp}) + Z_s - h_f$$

$$= \frac{144}{60.53}(12.25 - 7.51) + 10.0 - 0.0$$

$$= 21.3 \text{ ft}$$

2.3.4.17 NPSH margin considerations

Every system must be designed such that the NPSHA is equal to, or exceeds, the NPSHR by the pump throughout the range of operation. If the pump is allowed to operate with net positive suction head equal to NPSH3, the first-stage bowl head will drop by 3% from the published performance curve for a given speed, and increased noise, vibration, and damage associated with cavitation may occur. That is why pumping systems must be designed such that the NPSHA exceeds the NPSH required by the pump by a certain value, called *NPSH margin*. The amount of margin required varies depending on the pump design, the application, and the materials of construction. There is a tendency in most pumping systems for NPSHA to decrease with increasing flow rate; equally there is a tendency in most pump designs for NPSH3 to increase with increasing flow rate. Therefore adequate NPSH margins should be checked and confirmed for all anticipated flow rates considering manufacturers' recommendations and application experience. Refer to ANSI/HI 9.6.1 *Rotodynamic Pumps - Guideline for NPSH Margins* for detailed discussion of margins.

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2.3.4.18 NPSH requirements for pumps handling hydrocarbon liquids and water at elevated temperatures

The NPSH requirements of rotodynamic vertical pumps are normally determined on the basis of handling water at or near normal room temperatures. Operating experience in the field has indicated, and a limited number of carefully controlled laboratory tests have confirmed, that pumps handling certain hydrocarbon liquids, or water at significantly higher than room temperatures, will operate satisfactorily with less NPSHA than would be required for cold water (20 °C [68 °F]).

The consistency of results obtained on tests conducted with both water and hydrocarbon liquids suggests that the NPSHR of a rotodynamic vertical pump may be reduced when handling any liquid having relatively high vapor pressure at pumping temperature. However, available data are limited to the liquids for which temperature and vapor pressure relationships are shown on Figures 2.3.4.18a and b, thus application of these charts to liquids other than hydrocarbons and water is not recommended except where it is on an experimental basis.

Until specific experience has been gained with operation of pumps under conditions where Figures 2.3.4.18a and 2.3.4.18b apply, NPSH reduction should be limited to 50% of the NPSH required by the pump for cold water. When using these charts for high-temperature liquids, and particularly with water, due consideration must be given to the susceptibility of the suction system to transient changes in temperature and absolute pressure, which might necessitate provision of a margin of safety of NPSH far exceeding the reduction otherwise available for steady state operation. Because of the absence of available data demonstrating NPSH reduction greater than 3 m (10 ft), the chart has been limited to that extent and extrapolation beyond that limit is not recommended.

Figures 2.3.4.18a and 2.3.4.18b are based on pumps handling liquids without entrained or dissolved gas. When entrained or dissolved gases are present in a liquid, pump performance may be adversely affected, even with normal NPSH available, and would suffer further with reduction in NPSH available. Where dissolved gases are present, and where the absolute pressure at the pump inlet would be low enough to release such noncondensables from solution, the NPSHA may have to be increased above that required for cold water to avoid deterioration of pump performance due to such release. For hydrocarbon mixtures, vapor pressure may vary significantly with temperature, and specific vapor pressure determinations should be made for actual pumping temperatures.

To use these charts, enter Figures 2.3.4.18a and 2.3.4.18b at the bottom of the chart with the pumping temperature and proceed vertically upward to the true vapor pressure. From this point, follow along or parallel to the sloping lines to the right side of the chart, where the NPSH reduction may be read on the scale provided. If this value is greater than one half of the NPSHR on cold water, deduct one half of the cold water NPSHR to obtain the corrected NPSH required. If the value read on the chart is less than one half of the cold water NPSHR, deduct this chart value from the cold water NPSHR to obtain the corrected NPSHR.

Example (metric units): A pump that has been selected for a given rate of flow and head requires a minimum of 5 m NPSHR to pump that rate of flow when handling cold water. In this case, the pump is to handle propane at 12.8 °C, which has a vapor pressure of 690 kPa. Following the procedure indicated above, the chart yields an NPSH reduction of 2.9 m, which is greater than one half of the cold water NPSHR (required). The corrected value of NPSHR (required) is therefore one half the cold water NPSHR (required), or 2.5 m.

Example (metric units): The pump in the example above has also been selected for another application: to handle propane at -10 °C, where it has a vapor pressure of 345 kPa. In this case, the chart shows an NPSH reduction of approximately 1.83 m, which is less than one half the cold water NPSHR. The corrected value of NPSHR is therefore 5 m less 1.83 m, or 3.17 m.

Example (US customary units): A pump that has been selected for a given rate of flow and head requires a minimum of 16 ft NPSHR to pump that rate of flow when handling cold water. In this case, the pump is to handle propane at 55 °F, which has a vapor pressure of 100 psia. Following the procedure indicated above, the chart yields an NPSH reduction of 9.5 ft, which is greater than one half of the cold water NPSHR (required). The corrected value of NPSHR (required) is therefore one half the cold water NPSHR (required), or 8 ft.

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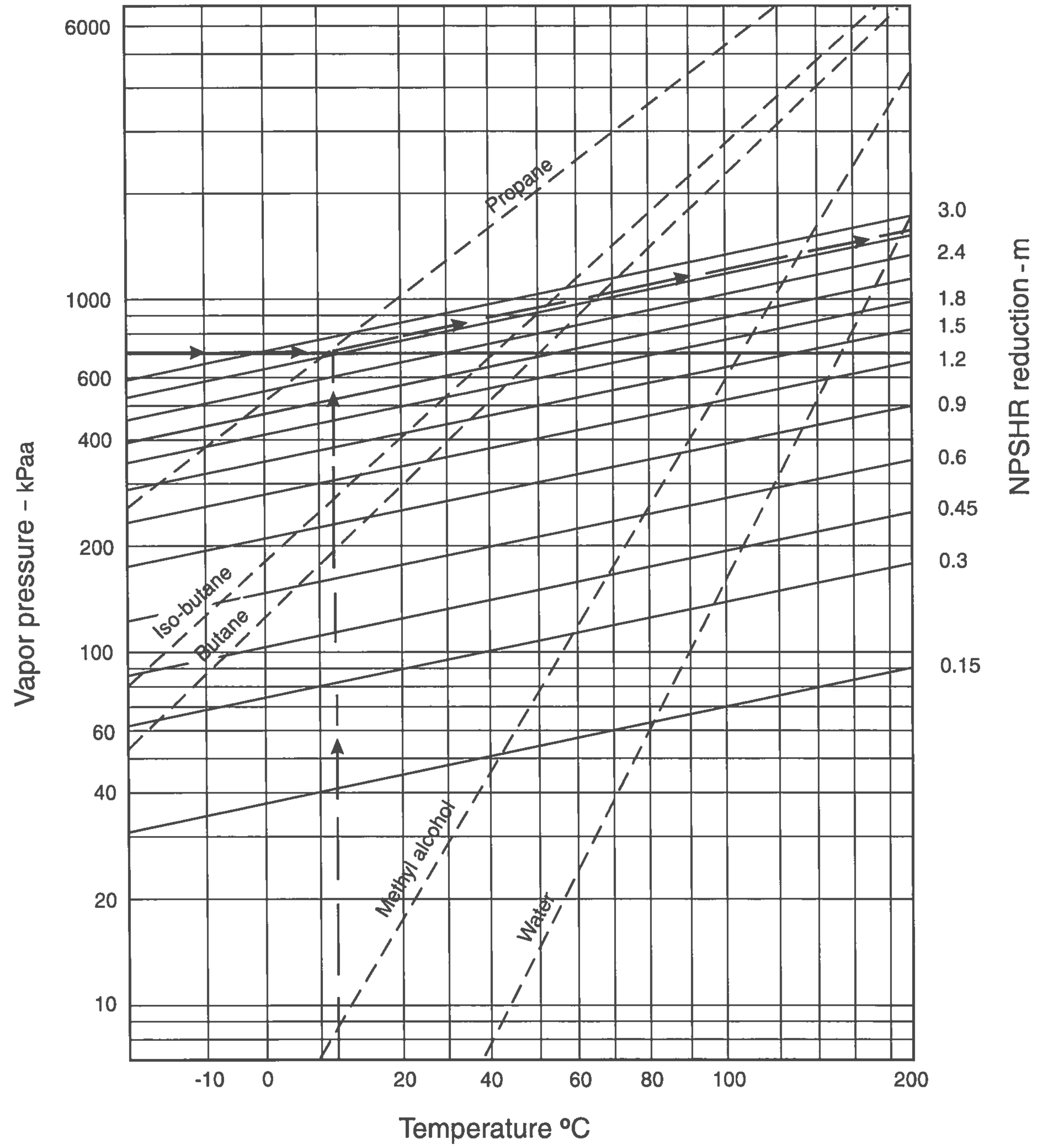

Figure 2.3.4.18a — NPSHR reduction for pumps handling hydrocarbon liquids and high-temperature water (metric units)

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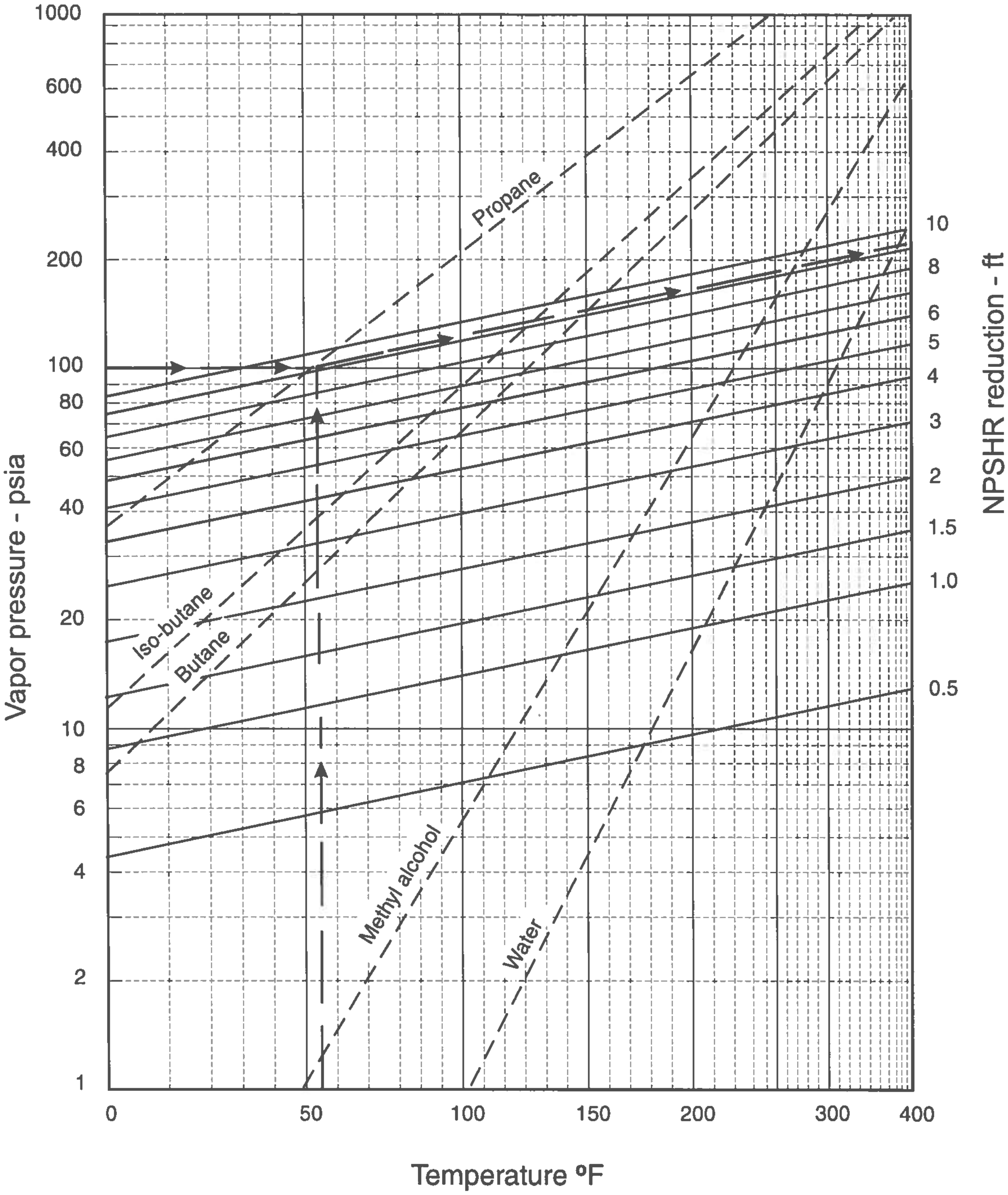

Figure 2.3.4.18b — NPSHR reduction for pumps handling hydrocarbon liquids and high-temperature water (US customary units)

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Example (US customary units): The pump in the example above has also been selected for another application: to handle propane at 14 °F, where it has a vapor pressure of 50 psia. In this case, the chart shows an NPSH reduction of 6 ft, which is less than one half the cold water NPSHR (required). The corrected value of NPSHR (required) is therefore 16 ft less 6 ft, or 10 ft.

2.3.4.19 Effects of handling viscous liquids

The performance of vertical pumps is affected when handling viscous liquids. A marked increase in input power, a reduction in head, and some reduction in rate of flow occur with moderate and high viscosities. For additional information, refer to ANSI/HI 9.6.7 *Effects of Liquid Viscosity on Rotodynamic (Centrifugal and Vertical) Pump Performance*.

2.3.4.20 Losses

In determining the overall pump efficiency, one must consider the hydraulic and power losses within the pump from the suction inlet to the outlet end of the discharge elbow. The hydraulic losses are from form (shape) resistance and surface friction losses within the pump suction and discharge conduits. The power losses are line-shaft bearing and thrust bearing friction losses. Overall pump efficiency will be less than the attainable bowl efficiency due to these losses. The internal head and power losses are added to bowl performance for purposes of delivering the required total head and determining the rated condition input power. Because of the variability in this style of pump, these corrections are specific for each application.

The suction bell is a flared tubular section for directing the flow into the impeller. Well pumps could have a threaded suction case, as in Figure 2.3.3.2.3a, part number 203, suitable for attachment of a suction pipe that allows an additional drawdown of the water level in the well while the pump is in operation. Local and surface friction losses for suction bell and threaded suction case are incorporated into the first-stage bowl performance and are not calculated separately. The hydraulic friction losses of the suction pipe used with threaded suction case have to be calculated. They will affect NPSH available to the pumps and overall head. Pumps equipped with optional suction strainers have some additional local loss, the value of which can be provided by the manufacturer.

The bowl efficiency is published or predicted by the pump manufacturer as a part of the bowl hydraulic characteristics for a one-stage pump. For any subsequent stage above the first stage, correction factors apply. They could affect head or efficiency or both. For pump bowls smaller than 500 mm (20 in), manufacturers often publish the head and efficiency performance curves on the basis of a minimum number of stages. Due to hydraulic losses at the inlet and discharge of the pump, single-stage attainable efficiency could be as much as six points below the bowl efficiency. This difference reduces as the number of stages increases. Typically this correction applies to four stages and less.

The losses in the column sections are due to friction on the column and the shaft. An estimate of these losses can be obtained from *HI Engineering Data Book*, 2nd ed. Refer to Figure IIIB-3(a) for column hydraulic loss, and Chart VE for line-shaft horsepower losses. The losses in the discharge head or elbow are a function of each design. If test data are not available, then refer to Figure IIIB-3(b) of the *HI Engineering Data Book* for an estimate.

Bearing spiders and enclosing-tube spiders are used to support the line-shaft bearing bushings and enclosing tube, respectively, and transmit the radial forces from the bearing bushings to the outer column. The friction losses in bearing bushings and spiders vary for different pump manufacturers. In addition, the method of lubrication when an enclosing tube is used affects the bearing losses. These losses have to be supplied by the pump manufacturer.

Losses due to column increasers and reducers, special pipe fittings, foot valves, etc. have to be calculated. A pump installed in a barrel will incur additional losses upstream of the pump suction, which will reduce the NPSHA.

Vertical electric motors, right gear drives, or pump bearing pots can be supplied with thrust bearing of various design or rating depending on the requirements of the pump installation. Because of this, the bearing friction loss will vary with the design and bearing type used. The manufacturer of the equipment with thrust bearing can provide

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the value of this loss. When no other information is available, for a single-row angular contact ball bearing, the loss can be estimated by:

$$P_{BL} = \frac{C \times n \times F_n}{100{,}000}$$

Where:

P_{BL} = thrust bearing friction loss, in kW (hp)

C = numerical coefficient:
= 0.0013 for metric units
= 0.0075 for US customary units

n = pump speed, in rpm

F_n = net pump assembly axial thrust, in N (lb)

Cylindrical, taper, and spherical roller and tilting pad bearings will have higher losses.

There is small loss in the shaft mechanical seal or stuffing box. It is usually disregarded unless the pump size driver is small where the relative value of the loss to the driver horsepower is proportionally noticeable.

2.3.5 Sound levels (dBA)

In general, for vertical pumps, the sound pressure level generated by the driver is greater than the sound pressure level generated by the pump. Additional consideration should be given to VFD systems.

Vertical turbine, mixed flow, double suction, and axial flow pumps (types VS0, VS1, VS2, and VS3) have their pumping elements submerged in the liquid below the station floor. Also vertical single- or multistage can pumps (types VS6 and VS7) have their pumping elements submersed inside the can, typically below ground level. Submersing the pumping element significantly dampens and reduces the pump sound pressure level as measured above at the floor.

The major sources of sound from a vertical pump, less driver, will be the elbow or turning losses in the pump discharge head and any residual sound from the pumping element. The driver sound pressure level will typically exceed the sound pressure level of the vertical pump. Sound emitted from the connected piping, valves, facility structures, etc. can be significant contributors to the overall sound pressure levels. If the sound pressure level of the pump were actually equal to the sound pressure level of the driver, it would add only 3 decibels to the driver sound pressure level. A conservative estimate of what the sound pressure level would be for the combined driver (data provided by driver manufacturer) and vertical pump is 3 decibels over the driver sound level.

For additional information on sound, see ANSI/HI 2.4 *Rotodynamic Vertical Pumps for Manuals Describing Installation, Operation, and Maintenance* and ANSI/HI 9.1-9.5 *Pumps - General Guidelines*.

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Appendix A

Pump classification and general application information

This appendix is included for informative purposes only and is not part of this standard. It is intended to help the user gain a better understanding of the factors referenced in the body of the standard.

A.1 Introduction to pump classifications

Rotodynamic pumps may be classified by such methods as impeller or casing configuration, end application, specific speed, or mechanical configuration. The method used within this standard (as indicated in Figures A.1, A.2, and A.3) is based primarily on commonly distinctive mechanical configurations with a demarcation between categories being determined by the arrangement of the rotor and the hydraulic configuration (impeller or casing).

Within these lines of demarcation there are pump types that can be clearly identified to fit into each of the defined categories. Commonly used pump types are classified as overhung (Type OH), between bearings (Type BB), and vertically suspended (Type VS).

ANSI/HI Standards (for design and application) have historically been subdivided into two categories:

rotodynamic centrifugal pumps (ANSI/HI 1.3)
and
rotodynamic vertical pumps (ANSI/HI 2.3).

For additional information on pump designations and types, refer to

- ANSI/HI 1.1-1.2 *Rotodynamic Centrifugal Pumps for Nomenclature and Definitions*
- ANSI/HI 2.1-2.2 *Rotodynamic Vertical Pumps for Nomenclature and Definitions*
- API 610 *Centrifugal Pumps for Petroleum, Petrochemical, and Natural Gas Industries*

A.2 Two-phase flow: Liquids with gas

Two-phase flow pumping applications include situations in which entrained gases or vaporized liquids are carried by the pumpage. One example is biological-fluid processing, such as the fermentation process used in yeast production. In this application, the process liquid is circulated from the bottom to the top of an aerator that injects large amounts of air into the process liquid. The fluid entering the pump may contain as much as 50% air by volume.

Oil production from wells often requires the pumping of crude oil that contains large amounts of natural gas mixed with the oil. In many cases, the inlet pressures for the pump range from 10,000 to 20,000 kPa (1500 to 3000 psi).

Other applications include those in which the inlet pressure is below atmospheric pressure. As a consequence, air can leak into the system, resulting in gas–liquid mixtures that must be handled by the pump. Even small amounts of air can cause problems because the air expands substantially under low pressure to increase its volume, particularly at the inlet of the pump impeller.

A.3 Effect of gas on performance

The most dramatic effect of gas or vapor on rotodynamic pump performance is the blocking of the impeller inlet. When this happens, the pump becomes airbound and the impeller acts as a centrifuge, tending to separate the heavier liquid from the gas that builds up at the impeller inlet. At low rates of flow, the liquid flow cannot even carry

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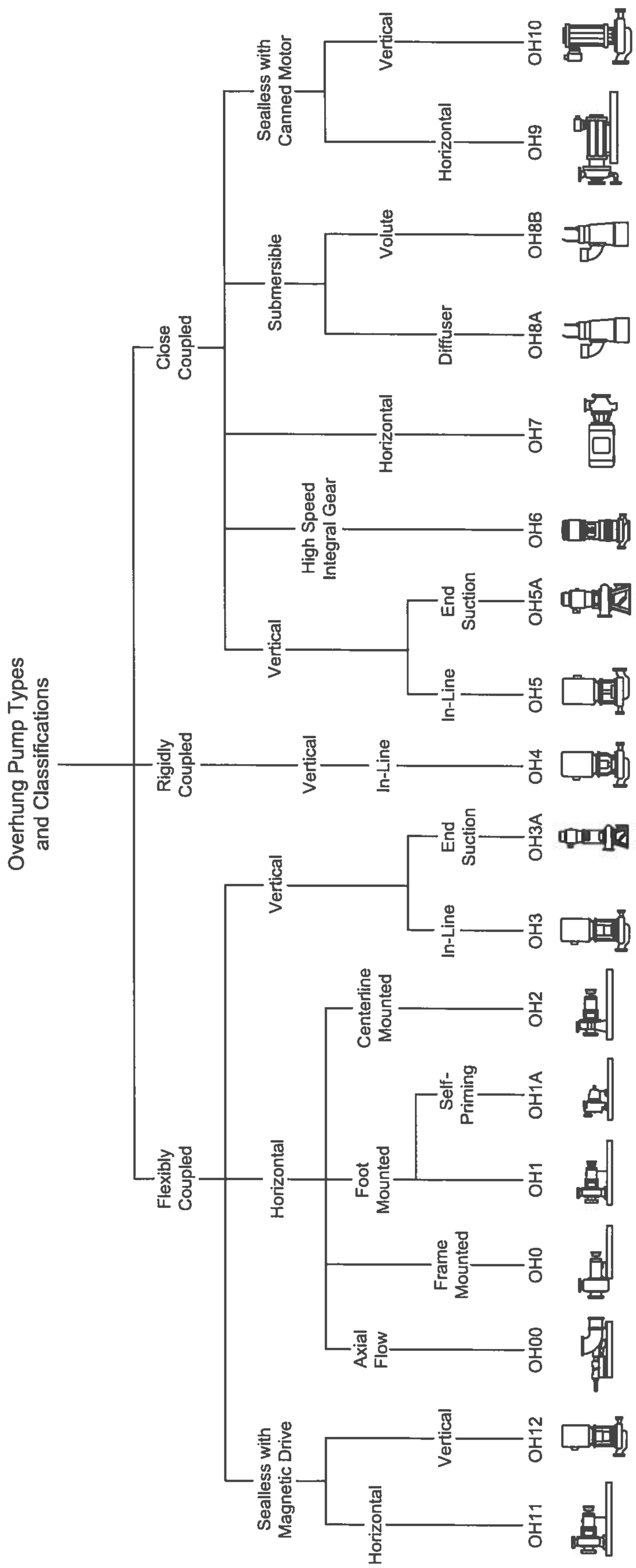

Figure A.1 — Overhung pump types and classifications

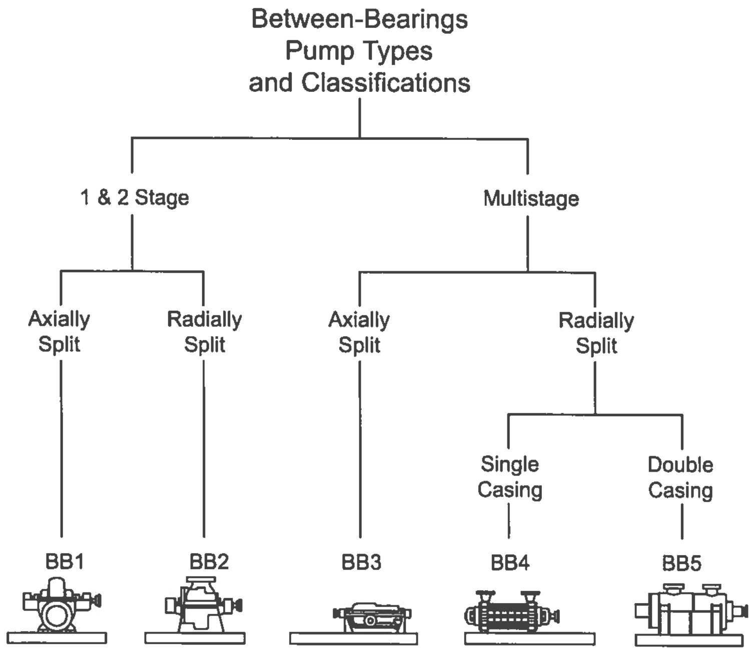

Figure A.2 — Between-bearings pump types and classifications

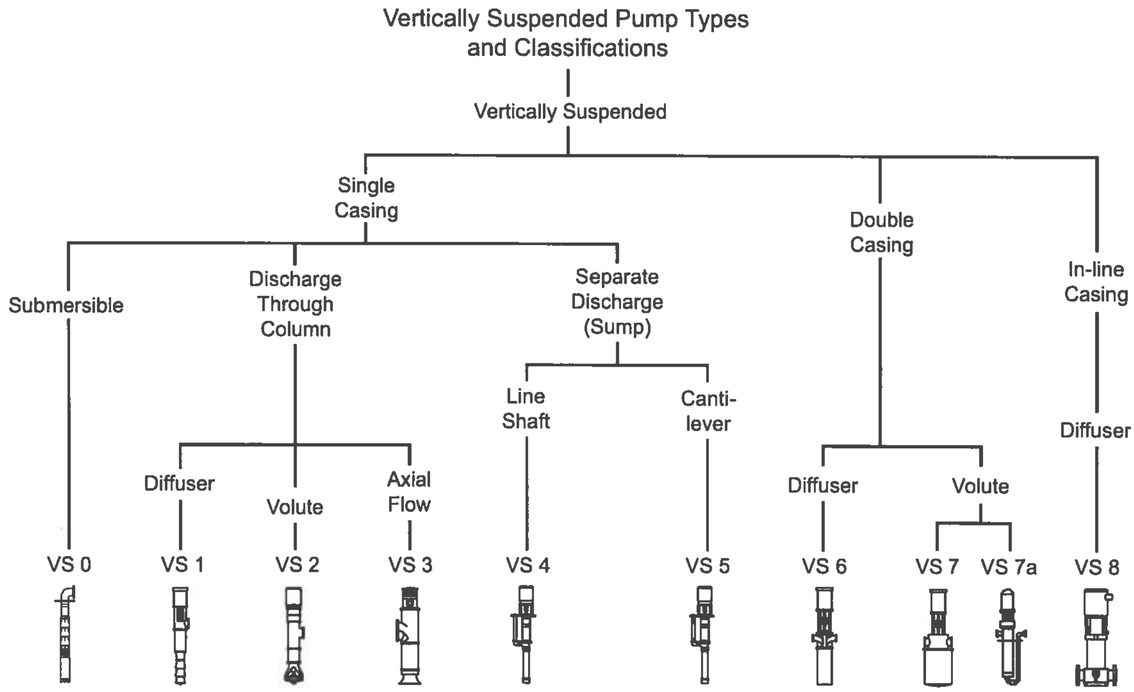

Figure A.3 — Vertically suspended pump types and classifications

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the gas through the impeller, and the gas bubble grows until it completely fills the impeller eye (suction side). The result is complete cessation of liquid flow.

Even when small amounts of gas are carried through the impeller, the liquid rate of flow and pump head are reduced (Figure A.4). This reduction is the result of the blockage of the flow by the gas, and a reduction in developed head due to the reduced specific gravity of the pumped mixture. Dissolved gases evolving out of solution can have similar effects. When the specific gravity of liquid alone is used to convert pressure to head, a lower head measurement is indicated.

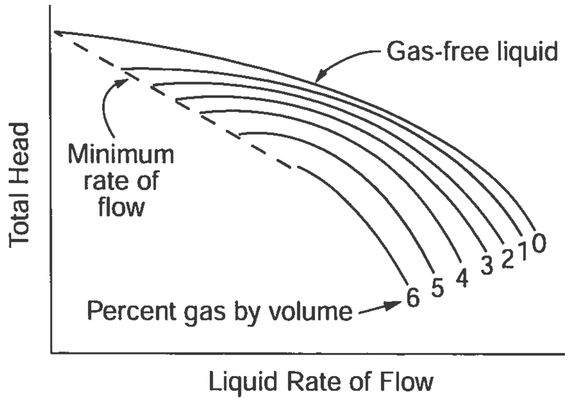

Figure A.4 — Effect of gas on pump performance

It can be seen from Figure A.4 that even with small percentages of gas, the unit stops pumping liquid due to accumulated gas in the impeller when operating near the shut-off condition of the pump. High velocities can carry higher percentages of gas. Therefore, when gas entrainment is a potential problem, pumps should be operated at or beyond the BEP rate of flow specified by the manufacturer.

Laboratory tests have shown that pumps with higher specific speed (above 60 [3000]) are affected less by the presence of gas than those with low specific speed (below 20 [1000]). The trend of gas affecting pump performance as a function of specific speed is only approximate and specific pump designs vary widely in their gas handling characteristics. In some cases, it may be helpful to use an inducer or higher specific speed impeller in the first (suction) stage of the pump.

Open impellers may handle gas better than closed impellers, particularly with large clearances between the impeller and the casing. The large clearance generates turbulence that helps prevent the accumulation of large gas pockets.

Another helpful action is to provide a gas vent at the pump inlet. The suction pipe should be sized about twice as large as the flange at the pump inlet in order to keep inlet velocities low. A vent connection should be located at the top of the pipe, close to the pump, so that gas can escape back to the source.

If the pump receives liquid from a closed system, then it may be possible to increase suction pressure thereby reducing the volume of entrained gas entering the pump.

By reducing the vapor temperature below its saturation temperature, some vapor will condense and thus reduce the volume of free vapor that must be handled by the pump.

A.4 Vertical pumps used as hydraulic turbines

Many rotodynamic vertical pump types may be adapted to operate in reverse rotation as hydraulic turbines.

While running in the turbine mode, the performance characteristics of a pump as turbine (PAT) differ significantly from pump operation. See Figure A.5. The applied head is usually constant, so the other parameters are shown as they vary with speed. The discharge nozzle of the pump becomes the inlet of the turbine, the suction nozzle of the pump becomes the outlet of the turbine, and the impeller of the pump, rotating in reverse direction, becomes the runner of the turbine. The impeller orientation to the casing is the same for both pump and turbine.

Fixed-geometry reverse running pumps are an alternative to custom-designed adjustable gate turbines. The efficiency of a pump operating as a turbine is comparable to the pump efficiency.

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For preliminary selection, a rough approximation procedure can be used to estimate the turbine performance from known pump performance.

$$Q_t = \frac{Q_p}{\eta}$$

$$H_t = \frac{H_p}{\eta}$$

Where:

Q_t = discharge rate of flow as turbine, in m^3/h (gpm)

Q_p = rate of flow as pump, in m^3/h (gpm)

H_p = total head as pump, in m (ft)

η = efficiency

H_t = net head as turbine, in m (ft)

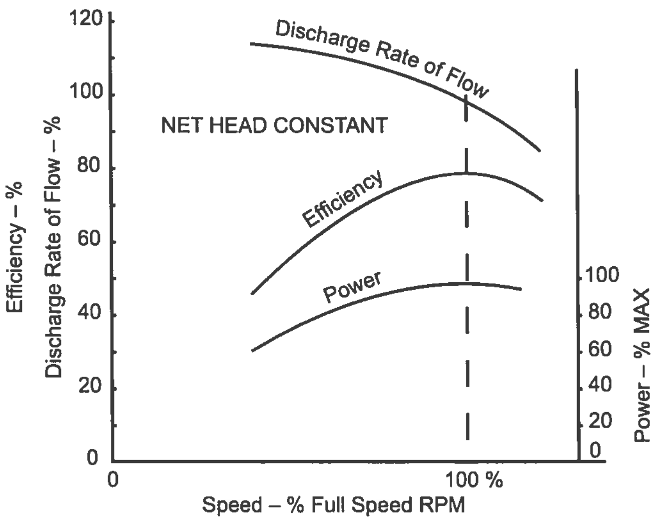

Figure A.5 — Turbine characteristics for pumps with n_s < 50 (2500)

Many vertical pumps, with some modification, are suitable and capable of operating as turbines. Because of the reverse rotation, one has to be sure that the bearing lubrication system will operate in reverse, and threaded shaft components, such as impeller locking devices, cannot loosen.

The power output is the rotational energy extracted by the reverse running pump. Its value is calculated in a similar manner as for a pump except for the placement of the efficiency term.

Metric units: $$P_t = \frac{Q \times H \times s \times \eta_t}{367}$$

US customary units: $$P_t = \frac{Q \times H \times s \times \eta_t}{3960}$$

Where:

P_t = power output from turbine, in kW (hp)

Q = discharge rate of flow, in m^3/h (gpm)

H = net head, in m (ft)

s = specific gravity

η_t = efficiency of the turbine

Special care should be taken in PAT applications to ensure that the mechanical design of the unit will allow safe operation. Frequently these applications subject the pump as turbine to increased mechanical stresses, torque, and speed levels beyond original pump design values. Additionally, the turbine characteristics are such that both hydraulic forces and torsional stresses increase with increasing rate of flow. All pumps applied as turbines should be subjected to a careful calculation of combined stresses in shafts.

Pumps operated in reverse as turbines tend to have relatively narrow operating bands, compared to variable nozzle (adjustable gate) turbines. At constant speed, the power developed and efficiency drop to zero at approximately

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40% of the hydraulic turbine best efficiency discharge rate of flow (see Figure A.6). Energy must be added to the hydraulic turbine for it to rotate at the constant speed below this rate of flow. Changing the runner (impeller) diameter has little effect on adjusting the performance of a hydraulic recovery turbine. These facts, coupled with the difficulty in predicting hydraulic turbine performance from pump performance, result in some uncertainty when applying a pump to a power recovery turbine application unless actual test data are available on the specific pump or model pump running in reverse as a turbine.

Precautions should also be taken to ensure that the pump as turbine operates without excessive cavitation. The pump industry uses the term *exhaust head* in place of *suction head* in reference to the backpressure energy level required at the outlet of the runner. There is no established industry definition for backpressure energy in vertical pump as turbine applications; however, for practical purposes the calculation is the same as for NPSH in pumps.

Some of the other factors that affect the use of pumps as turbines are:

- Runaway speed, overspeed trip, and control
- Sudden change of the discharge rate of flow due to turbine runaway
- Required solids passage
- Fluidborne abrasives
- Torque reversals during start-up or shut-down
- Lubrication of bearing bushings at the discharge stage

A.5 Rotative speed considerations

The maximum operating speed for a pump can be limited by the available NPSH in the system and the suction characteristics of the first stage. Excessive pump speed can result in unacceptable noise and vibration levels, abnormal wear, cavitation damage, and possible pump failure.

This section was introduced in the 1960s and provides rotative speed considerations based on a suction specific speed (*S*) of 165 (8500). The 165 (*S*) value is based on rate of flow in m^3/s. Refer to symbol *S* in Table 2.3.1.1 - Principal symbols.

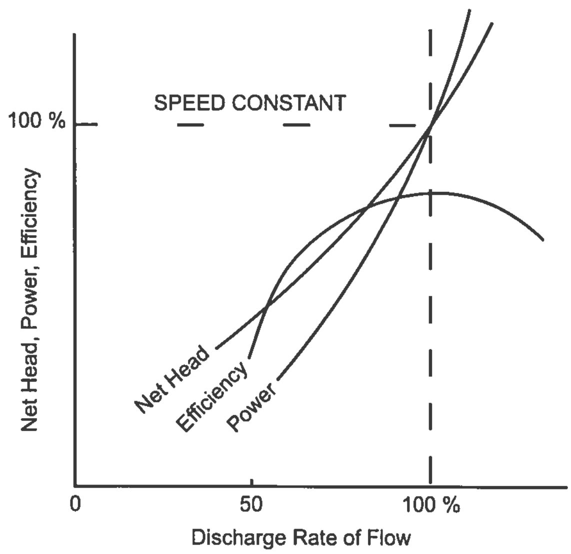

Figure A.6 — Turbine performance

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Since that time, newer pump design technology allows reliable operation of pumps with higher values of S through approximately 250 (13,000), depending on eye peripheral velocity, materials of construction, range of operation, pumped liquid properties, and other factors. This results in lower NPSHR, which allows higher rotative speeds with limited NPSHA. The following maximum operating speed information applies as well to the higher suction specific speed design pumps. In general, application of this rotative speed limitation information is for vertical turbine (diffuser) type single suction pumps with specific speeds (n_S) less than 95–115 (5000–6000).

The maximum speed for a pump (n) due to NPSHA can be calculated from the suction specific speed formula by expressing the rotative speed as a function of NPSHA, pump rate of flow (Q), and suction specific speed (S) as follows:

$$n = \frac{S \times NPSHA^{0.75}}{Q^{0.50}}$$

The curve presented in Figures A.7 and A.8 is based on a suction specific speed of 165 (8500) while operating at or near best efficiency. This represents a practical value for a typical pump handling cold water and liquids with similar properties. Obviously, operating speeds may be lower than those shown.

For pumps required to operate either continuously or for extended periods of time well above or below their point of optimum efficiency, a conservative suction specific speed should be used to ensure an adequate margin on NPSH to prevent cavitation damage. To ensure an adequate margin of NPSH to prevent cavitation damage, the available NPSH (NPSHA) must exceed the required NPSH (NPSHR) of the pump throughout the operating range. Some other factors that affect the degree of margin necessary are pump size, head per stage, product handled, and system transients or instabilities. Special materials may be used to minimize cavitation erosion damage as long as other detrimental effects are not present.

Example (metric units): Given a rate of flow of 10,000 m^3/h and NPSHA of 15 m, what is the rpm limit for 165 suction specific speed?

$$n = \frac{S \times NPSHA^{0.75}}{Q^{0.50}}$$

Therefore:

$$n = \frac{165 \times 15^{0.75}}{\left(\frac{10{,}000}{3600}\right)^{0.50}}$$

$$n = 755 \text{ rpm}$$

Referring to the curve in Figure A.7, the intersection of the vertical line for 10,000 m^3/h and the horizontal line for 15 m of NPSHA corresponds to 755 rpm.

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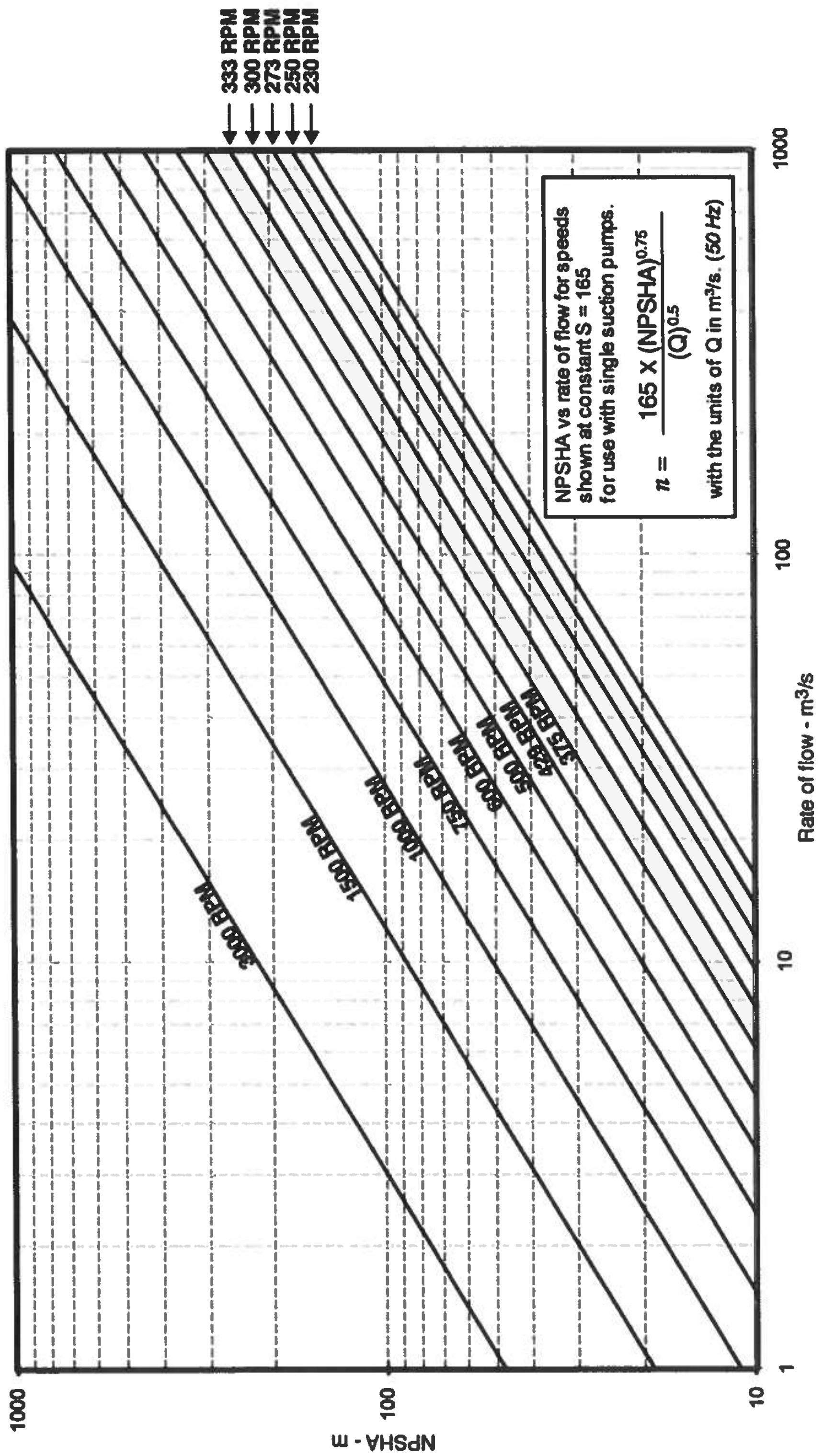

Figure A.7 — Recommended maximum operating speeds for single suction pumps (metric units)

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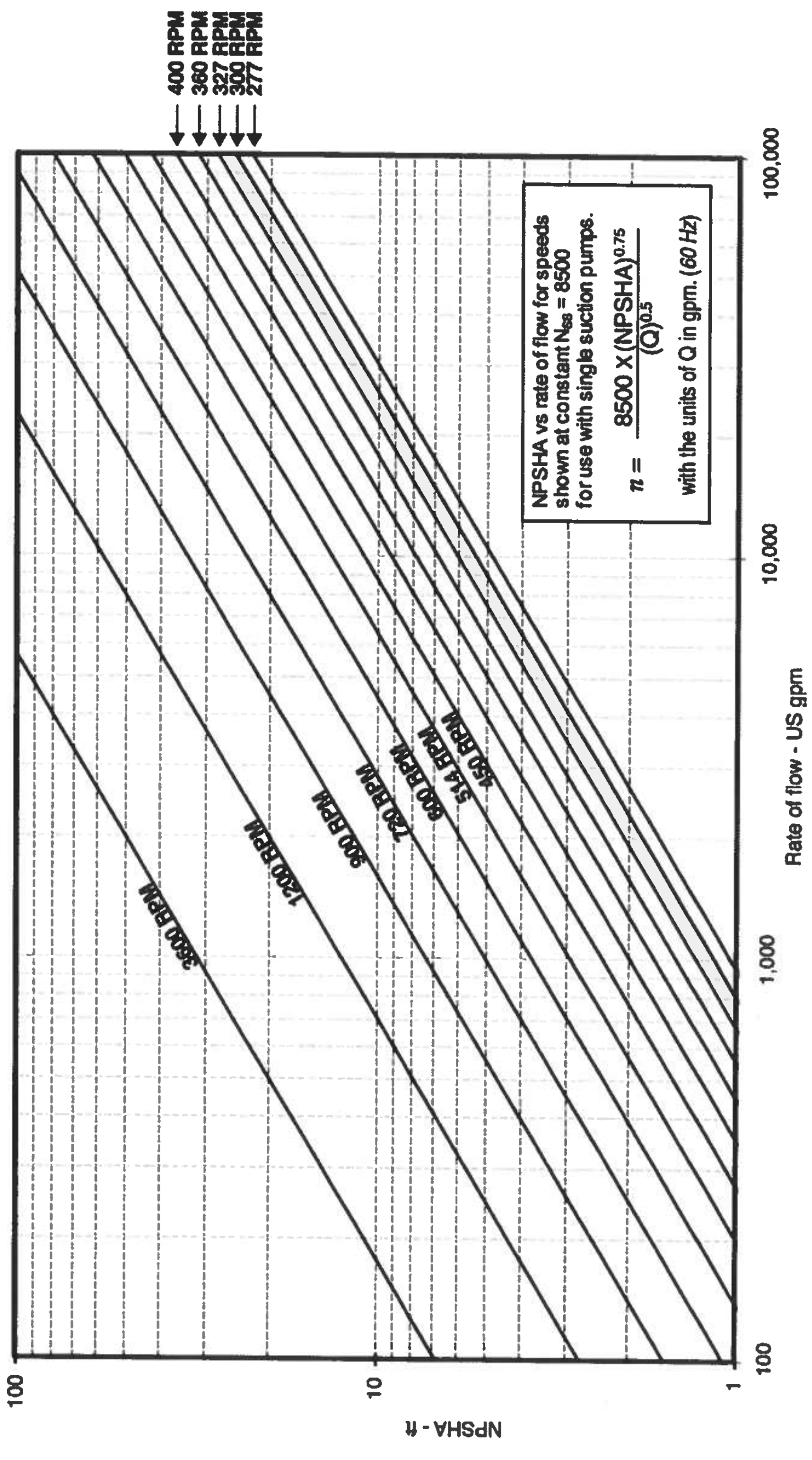

Figure A.8 — Recommended maximum operating speeds for single suction pumps (US customary units)

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Example (US customary units): Given a rate of flow of 90,000 gpm and NPSHA of 50 ft, what is the rpm limit for 8500 suction specific speed?

$$n = \frac{S \times NPSHA^{0.75}}{Q^{0.50}}$$

Therefore:

$$n = \frac{8500 \times 50^{0.75}}{90{,}000^{0.50}}$$

$$n = 533$$

Therefore, the recommended maximum operating speed is 533 rpm.

From Figure A.8, note that the intersection of the vertical line for 90,000 gpm and the horizontal line for 50 ft of NPSHA corresponds to 533 rpm.

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Appendix B

Other configurations

This appendix is included for informative purposes only and is not part of this standard. It is intended to help the user gain a better understanding of the factors referenced in the body of the standard.

B.1 Introduction to Appendix B

ANSI/HI Standards (for design and application) have historically been subdivided into rotodynamic centrifugal pumps (ANSI/HI 1.3) and vertical pumps (ANSI/HI 2.3) with a demarcation between the two categories being determined by the arrangement of the hydraulic configuration (impeller or casing). Within these lines of demarcation there are pump types that are clearly identified to fit into each of the two categories.

Appendix B provides a description of vertical cantilever sump pumps, previously only included in ANSI/HI 1.3, and discusses the design and application considerations that may be considered relevant to these specific configurations.

However, there are several pump types or arrangements that are not so clearly defined.

B.2 Sump pumps - introduction

Sump pumps can be considered something of a hybrid arrangement since they are configured to use the impeller and casing design elements described in the rotodynamic centrifugal pump standard (ANSI/HI 1.3). However, these hydraulic components are mechanically arranged to be vertically suspended and thereby immersed to a predetermined point in the sump, suitably below the liquid level.

Two distinct arrangements are commonly identified.

1) Single casing volute line shaft (type VS4)

   This arrangement uses many design elements covered in the rotodynamic vertical pump standard, including but not limited to 10 shaft (head), 12 shaft (line), 39 bushing (bearing), 70 coupling (shaft), 101 pipe (column), and 209 strainer.

   These types of pumps are used in many industrial process applications, including industrial sump wastes and tank unloading involving corrosive and noncorrosive liquid chemicals, hydrocarbon liquids, molten sulfur, and foul water.

2) Single casing volute cantilever (type VS5)

   This arrangement of stiff-shaft cantilever sump pumps is designed for operation without bottom bearing or flushing water, and is often used in corrosive, erosive environments and with thick, pulpy mixtures.

B.2.1 Description of sump pump type VS4 (line-shaft design)

Sump pumps are hybrid vertical pumps used for wet sump applications. In general, the pump has a casing attached to a column attached to the bottom of a support head (see Figure B.1). A motor on top of the support head assembly is mounted on top of the mounting plate over the pump shaft. A flexible coupling connects the pump and driver shafts. A discharge pipe is fixed to the pump casing and projects from the mounting plate. The whole pump-mounting, plate-driver assembly is then bolted to the top of a mounting hole in top of the sump.

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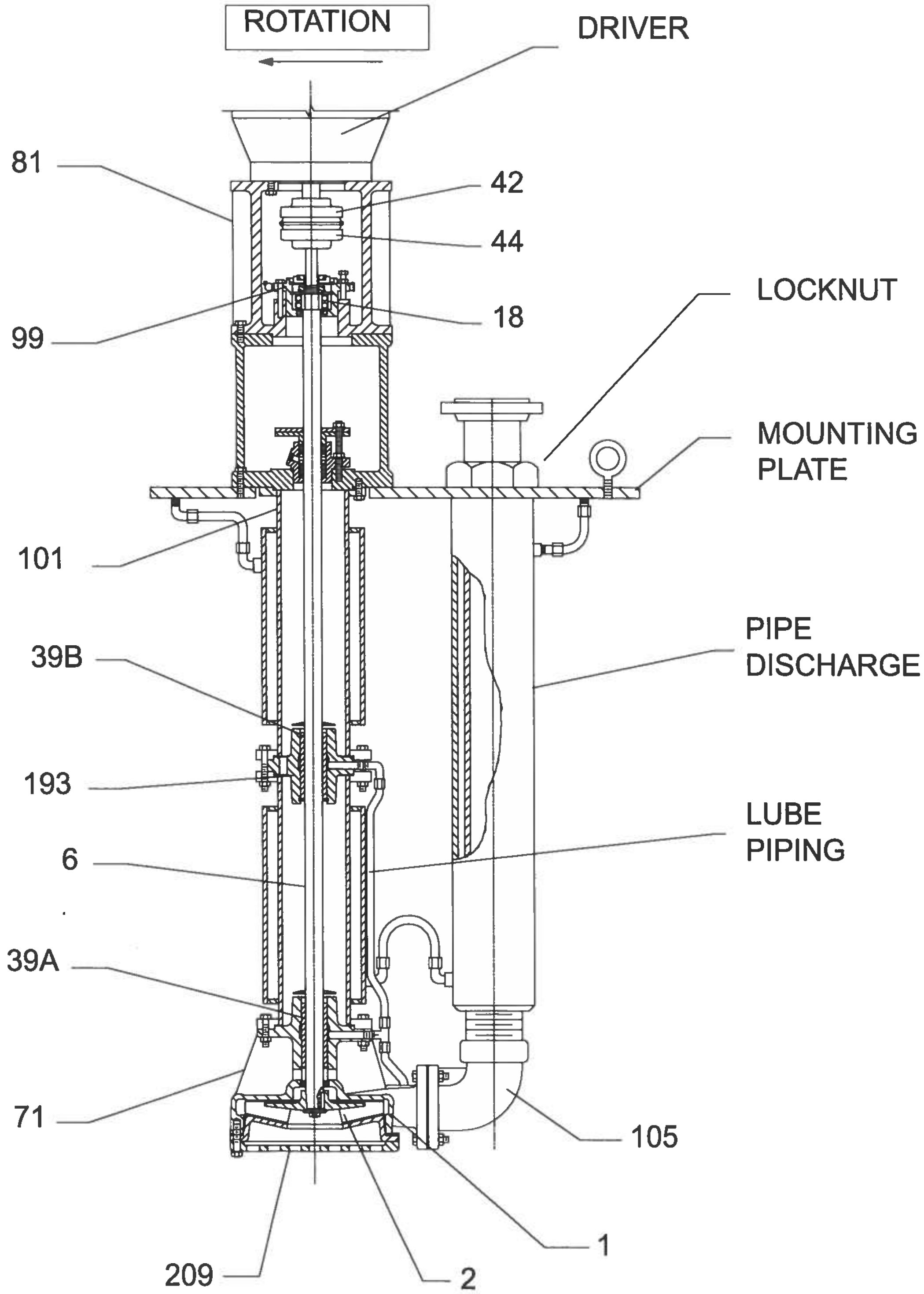

Figure B.1 — Type VS4 line-shaft design sump pump

These pumps use casings and impellers from the single-stage overhung designs. The casing is assembled to an adapter or can be part of an adapter bolted to a flange of a vertical column. The column is usually made from standard size pipe and can be constructed of one or more pieces. The top of the column is bolted to the bottom of a support head. The mounting plate fits on top of, and is bolted to, the sump cover opening. The support head is mounted on top of the mounting plate.

The impeller is mounted to a shaft that is longer than the column and is aligned radially by sleeve bearings in the adapter and intermediate bearings in the column. The shaft is supported axially by an antifriction bearing in a housing on top of the mounting plate. A flexible coupling is mounted between the top of the pump shaft and driver shaft.

The length of the one-piece shafts varies from 1 to 4 m (3 to over 12 ft). Sometimes multipiece shafts are offered. Some are as long as 6 m (20 ft).

An elbow is bolted to the discharge side of the casing directing the flow upward. A one-piece discharge pipe is attached to the elbow. The discharge pipe goes through the mounting plate where it is bolted or threaded to the piping system. The discharge pipe is held in position where it goes through the mounting plate by locknuts, weld, or with a separate plate bolted or welded to the mounting plate.

B.2.1.1 Definitions and terminology

B.2.1.1.1 Adapter (71)

This is a connecting piece between the casing and bottom flange of the column. It is the housing for the lower bearing. It may have column drain holes in the upper flange. There may be bypass holes between the impeller hub and the bottom of the bearing that are used to prevent dirty liquid from entering the bearing.

B.2.1.1.2 Bearing, outboard (18)

An antifriction bearing mounted in the bearing housing (99). Usually a single-row Conrad bearing for enclosed impeller designs and a double-row bearing for semi-open impellers. The bearings are grease or oil mist lubricated.

B.2.1.1.3 Bearing bushing in adapter (39A)

This is a sleeve bearing mounted in the adapter. It is capable of supporting the radial thrust from the impeller. These bearings may be water, grease, product, or self-lubricated.

B.2.1.1.4 Bearing bushing, intermediate (39B)

The bearings are mounted in the bearing retainer (193). They are to support the shaft at locations to prevent initiation of vibration from the first natural critical speed of the shaft. They are much shorter than the bearings in the adapter (71).

B.2.1.1.5 Bearing, retainer (193)

The retainer holds the intermediate bearings. They are drilled and taped for lubrication of the bearings. The axial location in the column assembly of the retainer depends on the first critical speed of the shaft. They are usually bolted between the flanges of the columns.

B.2.1.1.6 Bearing housing (99)

The housing is bolted to the top of the mounting plate. The housing design for semi-open impellers has adjustable axial movement.

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B.2.1.1.7 Casing (1)

Single-stage overhung water pump casings or ANSI B 73.1 casings.

B.2.1.1.8 Column (101)

There may be one or more columns that compose the entire column assembly. Usually made from standard pipe with pipe flanges. Top and bottom flanges are connected to the support head and pumping end adapter.

B.2.1.1.9 Coupling (42 and 44)

A flexible coupling is mounted between the pump shaft and driver shaft.

B.2.1.1.10 Discharge pipe

Consists of a one-piece pipe and an elbow. The elbow is bolted to the casing discharge flange. The pipe goes through a hole in the mounting plate. It is held in position in the mounting plate by two locknuts or is welded to the plate.

B.2.1.1.11 Driver

Usually standard *C*-face vertical motors. Some designs use *P*-face motors with rigid adjustable couplings. The *P*-face axial thrust bearings eliminate the need for separate bearing housings.

B.2.1.1.12 Impeller (2)

Single-stage enclosed or semi-open type impellers. Enclosed impellers are usually keyed to the shaft. Semi-open impellers are threaded to the shaft.

The rotation of the driver has to be verified before connecting the pump-coupling hub.

B.2.1.1.13 Lube piping

When used, there is usually a separate line to each bearing from a common manifold.

B.2.1.1.14 Mounting plate

This is the plate to which the motor support head, lubrication manifold, and discharge pipe are mounted or affixed. A gasket is put on the bottom of the plate, which is bolted to the top of the sump opening. The plate should be able to accommodate eyebolts for lifting the assembly with driver.

B.2.1.1.15 Shaft (6)

The shaft is made of precision ground steel. It is usually a one-piece construction. For extra-long designs, two pieces are rigidly coupled.

B.2.1.1.16 Strainer (209)

The strainer is bolted to the suction flange of the casing. The net open area is usually three times that of the suction nozzle.

B.2.1.1.17 Pedestal, Driver (81)

The bottom flange of the support head is rabbet fitted and bolted to the mounting plate. The top flange is rabbet fitted and bolted to the driver. The column is bolted to the bottom of the support head. The rabbet fits align the pump

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shaft, coupling, and driver shaft. The length of the support head is high enough to accommodate maintenance of the bearing housing and coupling.

B.2.1.1.18 Pit cover

Some applications have duplex pumps. In this case, the two pumps are mounted on a pit cover. Both pumps have their own mounting plates. The mounting plates are bolted to dual holes in the pit cover. The cover should accommodate eyebolts for lifting the entire assembly with drivers.

The cover can be round, square, or rectangular. They also have a manhole cover. The dual pumps can act as backups to each other or can operate in parallel.

B.2.1.2 Application

Because the casing is immersed below the liquid level at all times, these pumps are self-priming.

They are used on such applications as pump industrial waste, tank unloading, corrosive and noncorrosive liquids, and main turbine lube oil system. They have flows from 10 to 70 m^3/h (50 to 3000 gpm) with head from 15 to 120 m (50 to 400 ft), and temperatures from –40 °C to 230 °C (–40 °F to 450 °F). Most tanks are opened to the atmosphere, however, mounting plates for vapor-tight and pressurized sumps are offered. Sump depths go from 1 to 6 m (3 to over 20 ft.)

B.2.1.3 Performance

The performance that a customer requires is at the discharge connection on top of the mounting plate. The pump characteristic curve is usually measured at the discharge of the pump flange. Therefore, the static head of liquid in the tank and static head to the mounting flange connection, plus friction losses of the liquid in the casing discharge elbow and that in the discharge pipe, have to be taken into account. These losses are added to the customer-required head to derive the total head required from the pump when measured at the casing.

B.2.1.3.1 Static head

The distance from the "low" liquid level to the mounting-plate connection.

B.2.1.3.2 Friction loss

Friction loss through the discharge elbow plus loss of the discharge pipe from the elbow to mounting connection at the customer's flow condition.

B.2.1.3.3 Minimum submergence

The submergence to prevent vortex and vibration. Measured from the bottom of the strainer to liquid level (see Figure B.2).

B.2.1.3.4 Tailpipe

Tailpipes (see Figure B.3) are added to the suction nozzle of the casing for drawdown and stop service below the pump casing. Drawdown can be 2.5 m (8 ft) or more depending on the pump size and speed. They can be used for continuous service as long as proper minimum submergence is observed.

NPSHA should be greater than NPSHR. The friction loss through the pipe has to be accounted for in the NPSHA calculation.

Drain holes in the adapter have to be blocked to prevent air from entering the casing.

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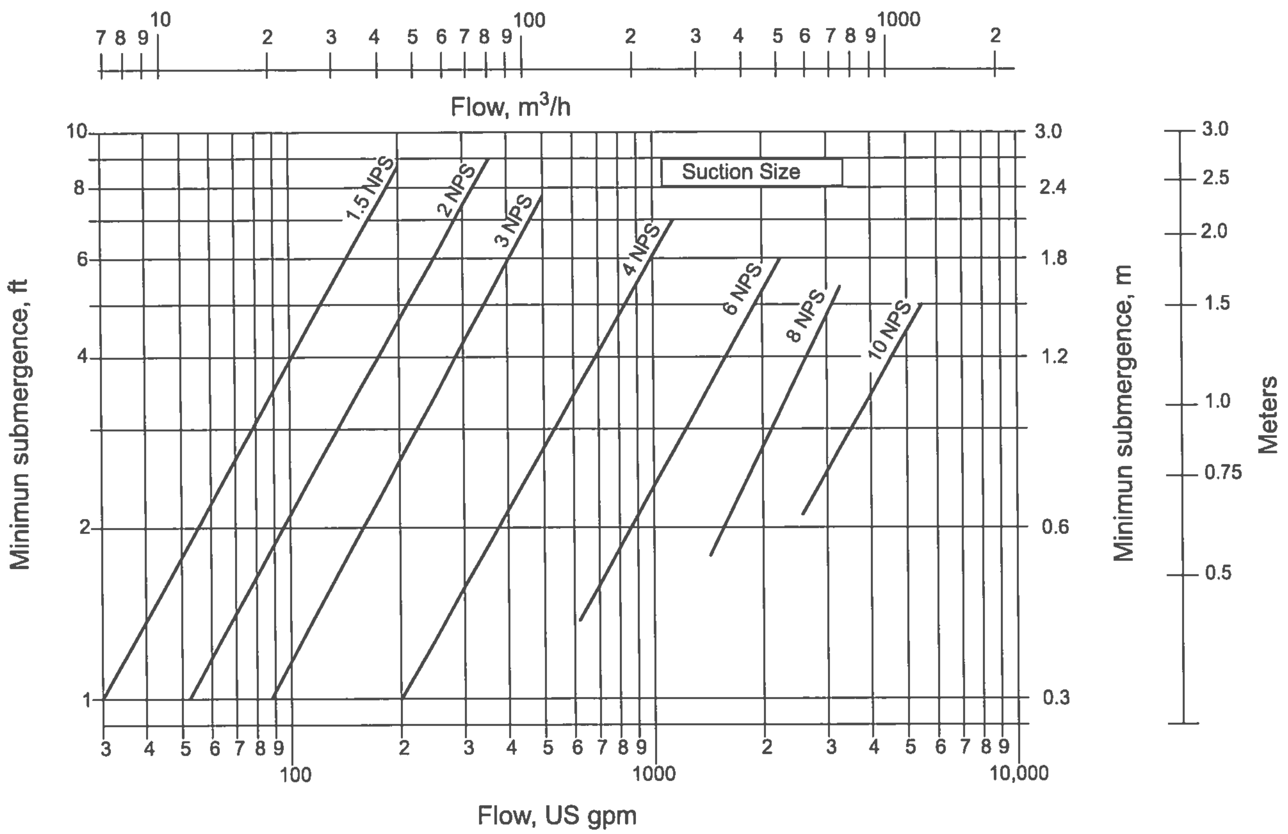

NOTE: Absolute minimum submergence for the above suction sizes at the lower flow rates are:

Suction size (DN)	40	50	80	100	150	200	250
Absolute minimum submergence (meters)	0.30	0.30	0.30	0.30	0.43	0.55	0.64

Suction size (NPS)	1.5	2	3	4	6	8	10
Absolute minimum submergence (feet)	1.0	1.0	1.0	1.0	1.4	1.8	2.0

Figure B.2 — Rate of flow versus minimum submergence

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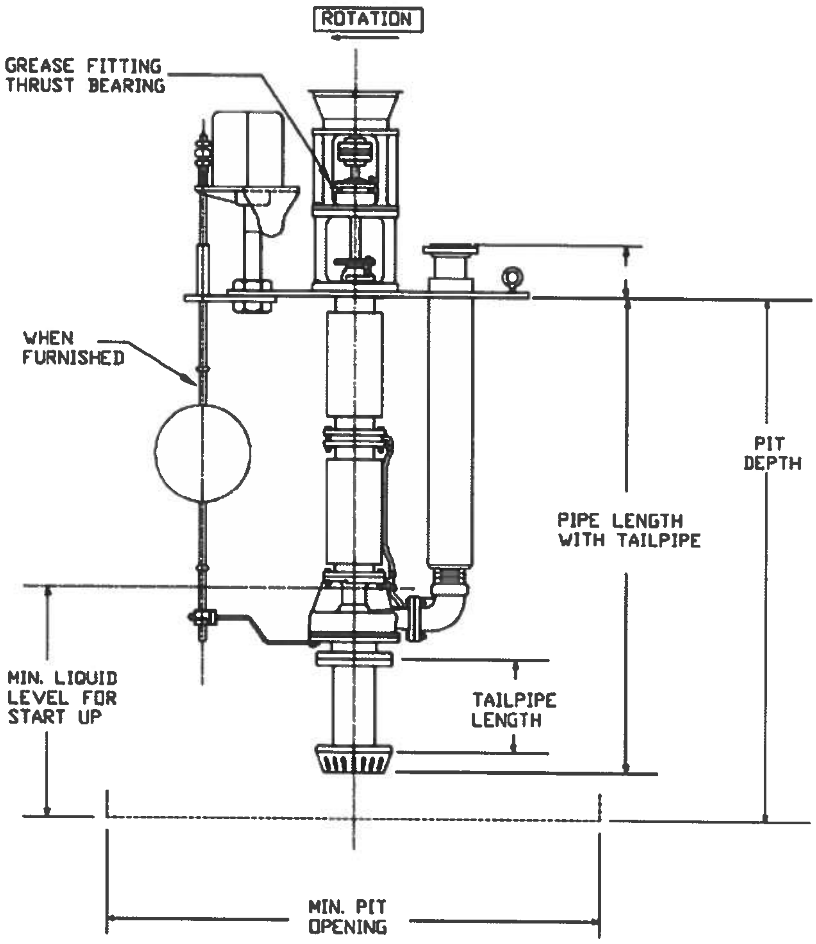

Figure B.3 — Tailpipe and float control

The flange of the tailpipe has to have a gasket to the suction flange of the casing and all pipe threads sealed.

The adapter bearing will require positive lubrication.

Tailpipes made of metal or polymer piping are used to reduce weight.

B.2.1.4 Special characteristics

B.2.1.4.1 Materials

Following is a list of major components representative for most applications.

- Casing and impeller: Cast iron, ductile iron, austenitic stainless steel, vinyl ester, epoxies.
- Column and discharge pipe: Carbon steel, austenitic steel, vinyl ester.
- Shaft: Carbon steel, plated carbon steel, reinforced vinyl ester.
- Bearings: The bearing material depends on the lubricant's temperature and cleanliness (particle size less than 10 micrometers [μm]). The following is a list of bearing materials and applications:
  - Carbon: Maximum 175 °C (350 °F), acids, general chemical, hydrocarbons

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- Bronze: Maximum 82 °C (180 °F), water and other compatible liquids
- Cast iron: Maximum 82 °C (180 °F), water, alkaline caustics
- Rubber: Maximum 71 °C (160 °F), general abrasive liquids
- Teflon: Maximum 175 °C (350 °F), liquid lube; 82 °C (180 °F), grease lube, clean acids not compatible with carbon
- Viton: Maximum 148 °C (300 °F), dirty acids not compatible with carbon or rubber

B.2.1.4.2 Temperature range of pump assembly

- Cast iron or bronze fitted: –28 °C to 121 °C (–20 °F to 250 °F)
- Carbon steel: –28 °C to 232 °C (–20 °F to 450 °F)
- Austenitic stainless steel: –73 °C to 175 °C (–100 °F to 350 °F)
- Polymers: –28 °C to 121 °C (–20 °F to 250 °F)

B.2.1.4.3 Temperature concern

For services above 121 °C (250 °F with metallic and 150 °F for polymers), the design of components and axial setting of impellers has to account for the difference in thermal expansion between (1) column and discharge pipe, and (2) the shaft.

B.2.1.4.4 Pressurized and vapor-tight construction

For closed systems, pressurized or vacuum service, the mounting plate, column flange, pit cover, stuffing box, mechanical seal, bearing housing, discharge pipe, and lubrication lines have to be designed to contain pressure or maintain vacuum. Refer to Figure B.4.

The purchaser and vendor should mutually agree on the mounting-plate design and thickness.

B.2.1.4.5 Lubrication cleanliness

Lubrication with less than 10-µm size particles can be handled by most bearing materials. If particles larger than 10 µm are present, filters or cyclone separators should be used. Filters may need frequent replacement or cleaning. Separators are installed on top of the mounting plate. A lube line is taken from the discharge and goes to the side of the separator. The clean liquid goes from the top of the separator to the bearing lube line manifold. The liquid with the concentrated particles is ejected from the bottom of the separator back to the sump.

The number of separators depends on the discharge pressure and number of bearings. To determine the total flow of the pump, add 8 L (2 gal) per separator to the customer's required flow.

B.2.1.4.6 Alarms and controls

Optional features include the addition of high and low level on control floats and a high liquid level alarm. See Figure B.5.

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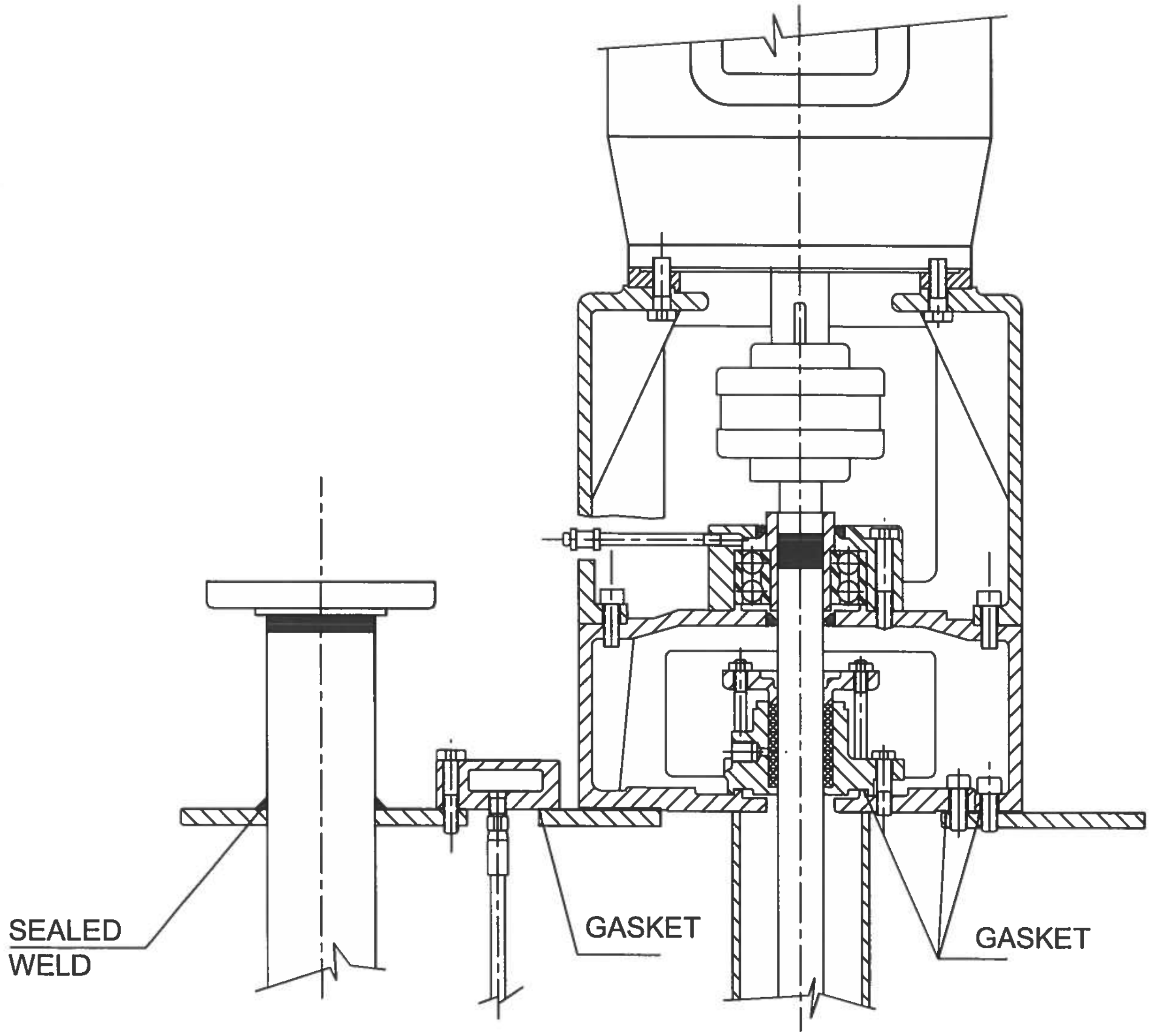

Figure B.4 — Schematic showing vaporproof/pressurized design

B.2.1.4.7 Nozzle loads

For pump type VS4 (and VS5), the allowable nozzle loads depend on the mounting-plate thickness, plate material, attachment of the discharge pipe to the mounting plate, type of impeller, and bearing design. The customer should discuss with the manufacturer the amount of allowable forces and moments (see Figure B.6).

B.2.2 Description of sump pump type VS5 (cantilever shaft design)

A sump pump is a hybrid vertical pump used for wet sump applications. The cantilever sump pump (See Figure B.7) is used where the line-shaft bearing cannot be used because the liquid is not compatible with journal bearings. Primarily cantilevers are used because of the contaminants in the sump (solids/slurry) that would damage line-shaft journal bearings. They are also used where the temperature is too high for submersibles. The cantilever over-hang is typically 1.2 to 1.8 m (4 to 6 ft). The cantilever design limits the shaft length to about 2.4 to 3.4 m (8 to 11 ft).

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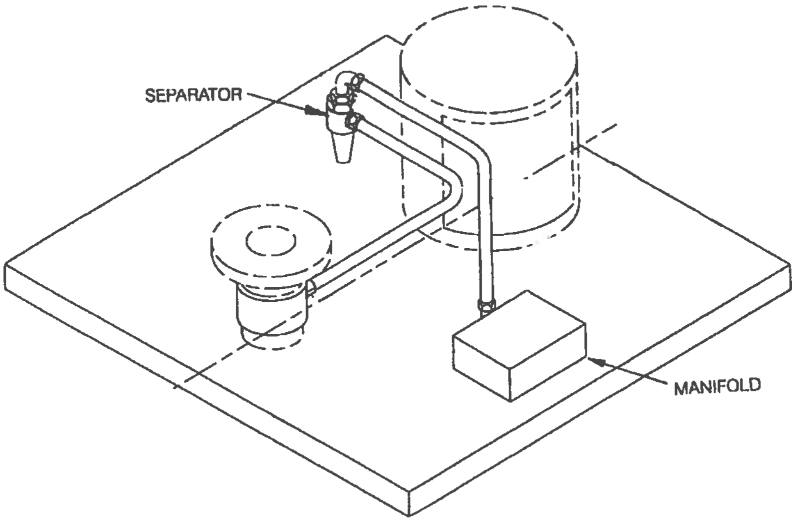

Figure B.5 — Alarms and controls

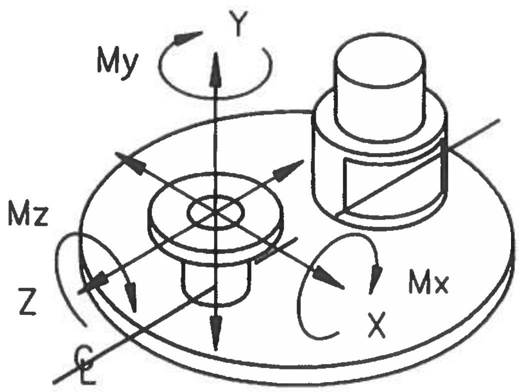

Figure B.6 — Applied forces and moments

The reasons to use a cantilever include the following:

- Eliminate packing or mechanical seal
- Requires no submerged bearings
- Requires no in-sump lubrication
- Requires no check or foot valves
- Is easily primed
- Can run dry
- Bearings and motor are located above the liquid level

In general, the pump has a casing attached to a column attached to the bottom of a support head. A motor on top of the support head assembly is mounted on top of the mounting plate over the pump shaft. A flexible coupling connects the pump and driver shafts. Also, some designs have a side-mounted motor with a belt drive to the pump

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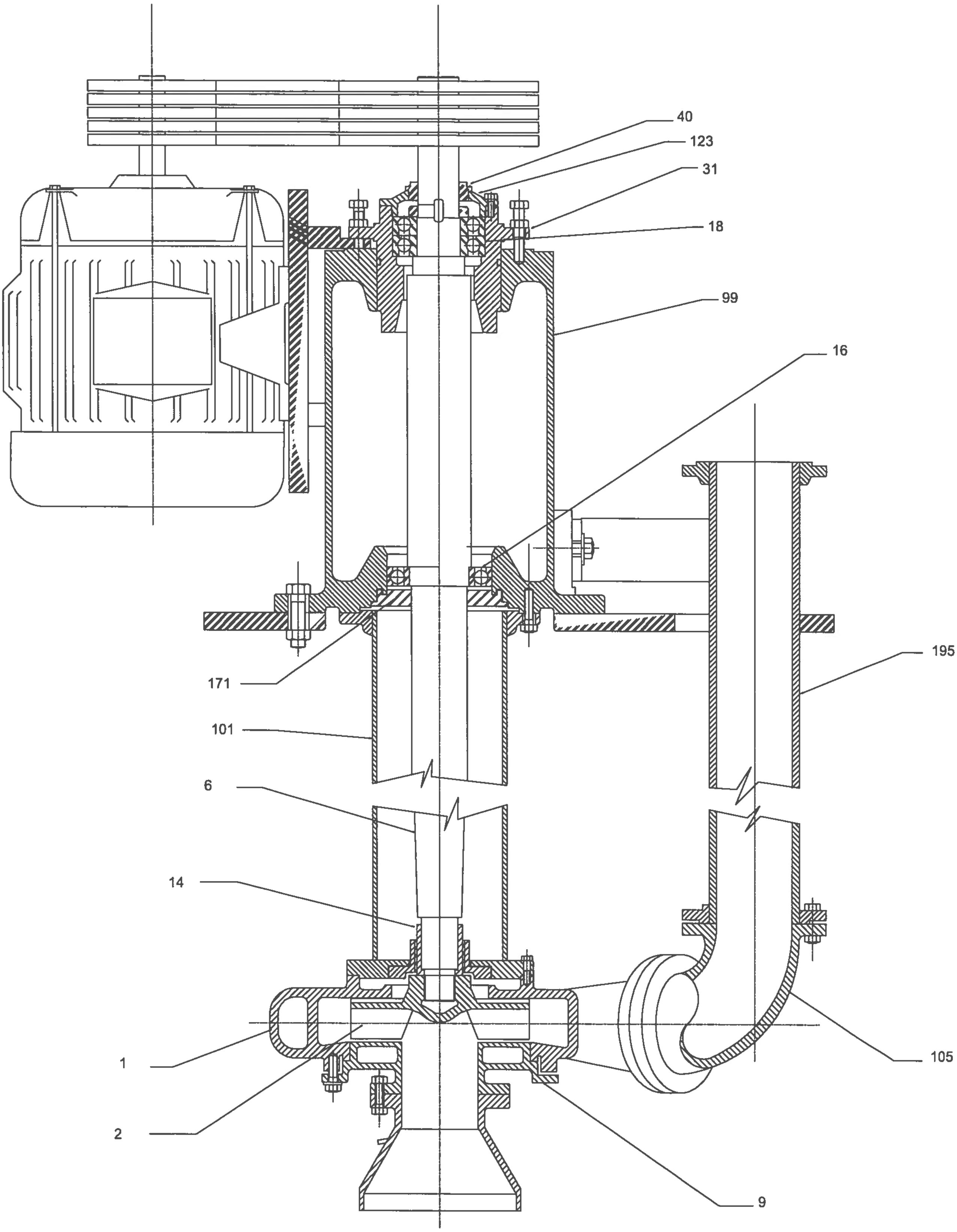

Figure B.7 — Type VS5 cantilever shaft design sump pump

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shaft. A discharge pipe is fixed to the pump casing and projects from the mounting plate. The whole pump mounting-plate–driver assembly is then bolted to the top of a mounting hole in top of the sump.

These pumps use casings and impellers from the single-stage overhung designs. The casing is assembled to an adapter or can be part of an adapter that is bolted to a flange of a vertical column. The column is usually made from standard size pipe. The top of the column is bolted to the bottom of a support head. The mounting plate fits on top of, and is bolted to, the sump cover opening. The support head is mounted on top of the mounting plate.

The semi-open, enclosed, or vortex-type impeller is mounted to a shaft that is longer than the column. An antifriction bearing in a housing on top of the support head supports the shaft axially.

A vertical elbow is bolted to the discharge of the casing. A one-piece discharge pipe is attached to the elbow. The discharge pipe goes through the mounting plate where it is bolted or threaded to the piping system. The discharge pipe is held in position where it goes through the mounting plate by locknuts, weld, or with a separate plate that is bolted or welded to the mounting plate.

B.2.2.1 Definitions and terminology

B.2.2.1.1 Bearing, outboard (antifriction) (18)

An antifriction bearing mounted in the bearing housing (99). The bearings are grease or oil mist lubricated.

B.2.2.1.2 Bearing housing (99)

The housing is bolted to the top of the support head. Sometimes the housing itself functions as the support plate. The housing, which is designed for semi-open, enclosed, or recessed impellers, has adjustable axial movement.

B.2.2.1.3 Casing (1)

Single-stage overhung water pump casings or ASME B 73.1 casings.

B.2.2.1.4 Column (101)

Usually made from standard pipe with pipe flanges. Top and bottom flanges are connected to the support head and pumping end adapter.

B.2.2.1.5 Discharge pipe

Consists of a one-piece pipe and an elbow. The elbow is bolted to the casing discharge flange. The pipe goes through a hole or slot in the mounting plate. It is held in position in the mounting plate by two locknuts or welded to the plate, secured to the bearing frame, or uses a "U" secured to the support head.

B.2.2.1.6 Driver

Standard *C*-face or *P*-base vertical motors or side-mounted vertical motors with a pulley (*V*-belt) drive to the pump shaft. The rotation of the driver has to be verified before connecting the pump-coupling hub.

B.2.2.1.7 Impeller (2)

Semi-open, enclosed, or vortex impeller. Impellers are threaded or keyed to the shaft.

B.2.2.1.8 Mounting plate

This is the plate to which the support head and discharge pipe are mounted or affixed. A gasket is put on the bottom of the plate, which is bolted to the top of the sump opening. The plate should be able to accommodate eyebolts

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for lifting the assembly with driver. On direct-drive pumps, lifting with the driver is not desirable due to the height of the frame and motor. In this case, the motor should be removed.

B.2.2.1.9 Shaft (6)

The shaft is one piece and is cantilevered from the bearings in the support head.

B.2.2.1.10 Pedestal, Driver (81)

The bottom flange of the support head is bolted to the mounting plate. The driver is bolted to the support head. The top flange may be rabbet fitted. The column is bolted to the bottom of the support head. The rabbet fits align the pump shaft, coupling, and driver shaft.

B.2.2.1.11 Pit cover

Some applications have duplex pumps. In this case, the two pumps are mounted on a pit cover. Both pumps have their own mounting plates. The mounting plates are bolted to dual holes in the pit cover. The cover should accommodate eyebolts for lifting the entire assembly with drivers.

The cover can be round, square, or rectangular. They also may have a manhole cover. The dual pumps can act as backups to each other or can operate in parallel. Because of the irregular size and shape, pit covers are used to cover the sump. A standard support plate is not sized for all sump configurations.

B.2.2.2 Application

Sump pumps are self-priming because the casing is immersed below the liquid level at all times. They are used on such applications as pump industrial waste, tank unloading, and corrosive and noncorrosive liquids. They have flows from 10 to 160 m^3/h (50 to 7000 gpm) with head from 15 to 120 m (50 to 400 ft). Temperature range is from –40 °C to 120 °C (–40 °F to 250 °F). Most tanks are opened to the atmosphere. Sump depths go from 1 to over 3 m (3 to over 10 ft). Tailpipes are used to reduce the level in the sump.

B.2.2.3 Performance

The performance that the customer requires is at the discharge connection on top of the mounting plate. The pump's characteristic curve is at the discharge of the pump flange. Therefore, the static head of liquid in the tank, static head to the mounting flange connection, and friction losses of the liquid in the elbow on the casing discharge and that in the discharge pipe have to be accounted for. These losses are added to what the customer requires to obtain head required for the pump.

B.2.2.3.1 Static head

The distance from the "low" liquid level to the mounting-plate connection.

B.2.2.3.2 Friction loss

Friction loss through the discharge elbow plus loss of the discharge pipe from the elbow to mounting connection at the customer's flow condition.

B.2.2.3.3 Minimum submergence

The submergence to prevent vortex and vibration. There is a priming level; usually the casing that must be completely submerged before priming can be accomplished. See Figure B.8.

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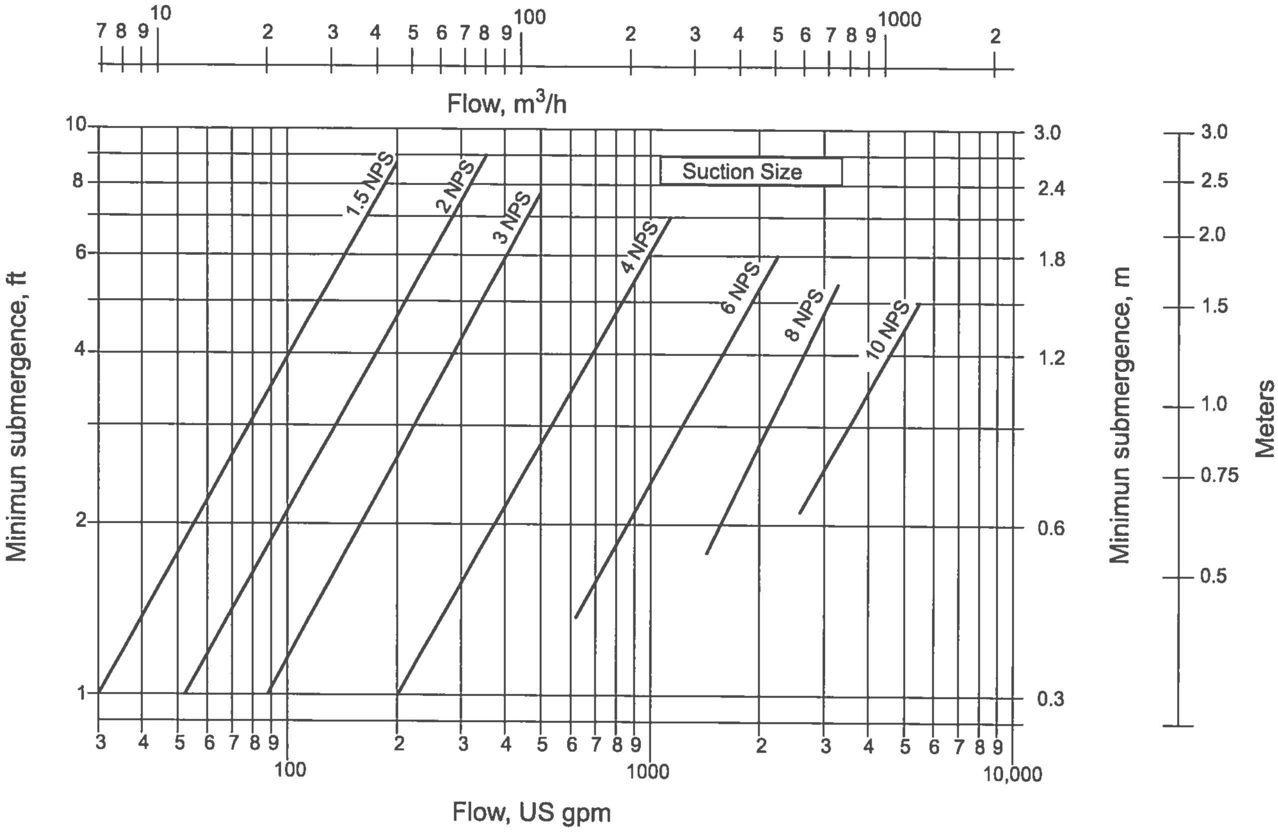

NOTE: Absolute minimum submergence for the above suction sizes at the lower flow rates are:

Suction size (NPS)	1.5	2	3	4	6	8	10
Absolute minimum submergence (feet)	1.0	1.0	1.0	1.0	1.4	1.8	2.1

Suction size (DN)	40	50	80	100	150	200	250
Absolute minimum submergence (meters)	0.30	0.30	0.30	0.30	0.43	0.55	0.64

Figure B.8 — Rate of flow versus minimum submergence

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B.2.2.3.4 Tailpipe

Tailpipes are added to the suction nozzle of the casing for drawdown and stop service below the pump casing. Drawdown can be 2.5 m (8 ft) or more depending on the pump size and speed. They can be used for continuous service as long as proper minimum submergence is observed.

NPSHA should be greater than NPSHR. The friction loss through the pipe has to be accounted for in the NPSHA calculation.

The flange of the tailpipe has to have a gasket to the suction flange of the casing and all pipe threads sealed. Thin metal piping or polymer piping is used to reduce weight.

B.2.2.4 Special characteristics

B.2.2.4.1 Materials

Major components, representative for most applications are as follows:

- Casing and impeller: Cast iron, ductile iron, austenitic stainless steel
- Column and discharge pipe: Carbon steel, austenitic steel
- Shaft: Carbon steel, austenitic stainless steel
- Mounting plate: Carbon steel, plated carbon steel

B.2.2.4.2 Thermal range of assembly

Temperature range of pump assembly:

- Most cantilevers are used in services of less than 120 °C (250 °F). Because the basic cantilever design does not lend itself to sealing the sump, sump liquid vaporization will affect the life of the motor bearings.
- Cast iron or bronze fitted: –28 °C to 121 °C (–20 °F to 250 °F)
- Carbon steel: –28 °C to 121 °C (–20 °F to 250 °F)
- Austenitic stainless steel: –73 °C to 121 °C (–100 °F to 250 °F)

B.2.2.4.3 Temperature concerns

For services above 120 °C (250 °F), with metallic construction, the design of components and axial setting of impellers has to account for the difference in thermal expansion between column and discharge pipe, and column and shaft. Expansion rate between the shaft and bearing housing can also be an issue when using alloy shafting and cast-iron bearing frame.

B.2.2.4.4 Alarms and controls

The addition of high and low level on control floats is an optional feature. Also a high-liquid-level alarm can be employed.

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B.2.2.4.5 Nozzle loads

For pump type VS5 (and VS4), the allowable nozzle loads depend on the mounting-plate thickness, plate material, attachment of the discharge pipe to the mounting plate, type of impeller, and bearing design. The customer should discuss with the manufacturer the amount of allowable forces and moments. Refer to Figure B.9.

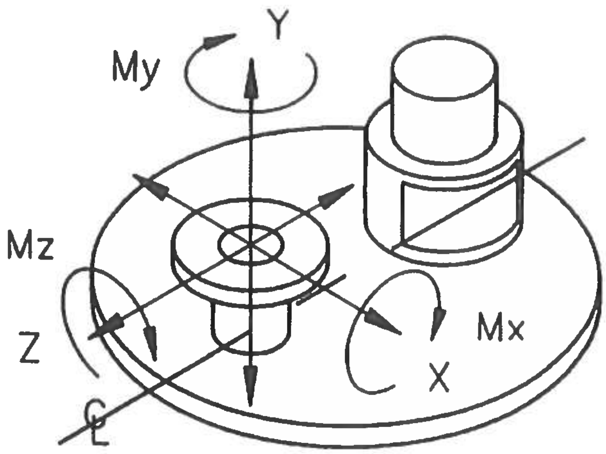

Figure B.9 — Applied forces and moments

Appendix C

Materials

This appendix is included for informative purposes only and is not part of this standard. It is intended to help the user gain a better understanding of the factors referenced in the body of the standard.

C.1 Corrosion and erosion in vertical turbine pumps

The waste of metal due to corrosion and erosion is a very important problem. Corrosion and erosion can be so closely related as to make them hard to distinguish.

a) *Corrosion* may be broadly defined as the destruction or deterioration of metal by direct chemical or electrochemical reaction with its environment.

b) *Erosion* is material wear due to abrasion or cavitation, and is often combined with corrosion.

The mechanism of corrosion is the same for all metals and alloys, differing in degree but not in kind. It is well established that in most cases of corrosion in the presence of water, the driving force of the corrosive reaction between a metal and its environment, is electrochemical. The degree of protection of the oxide layer in neutral (pH = 7) pH liquids accounts for the great variation in the corrosiveness found. The resulting coating formed in alkaline (pH >7) solutions is more protective than one formed in neutral solutions, and therefore, results in a lower rate of corrosion. The initial rate of corrosion in alkaline solutions is about as great as that of neutral pH liquids, but the corrosion rate decreases as the protective film is formed. In most cases, in the acid range (pH < 7), the protection afforded by protective films is slight.

The main forms of corrosion and erosion that may be encountered shall be guarded against by proper selection of materials and careful matching of the pump to the application.

Galvanic corrosion:
When two dissimilar metals contact or are otherwise electrically connected to each other in an electrically conductive liquid, a galvanic cell is formed and current flows from one to the other. Proper choice of metals for the given environment can minimize the effects of galvanic corrosion.

Uniform corrosion:
This, as the term implies, is a uniform attack on the metal and may be solved by selecting a material more resistant to the corroding solution, the use of inhibitors, protective coating, or combinations of each.

Erosion – corrosion:
Most metals depend on the formation of a protective surface film to prevent further chemical attack. It is understandable, therefore, that an increase in velocity or abrasives in the pumped fluid could erode the protective film and result in accelerated corrosion.

Cavitation erosion:
Most pumping applications experience some degree of cavitation, especially when the pump is operated at off-design conditions. Cavitation erosion can be significant with higher pump internal velocities, combined with a low margin ratio (NPSHA/NPSH3), and other factors.

Intergranular corrosion:
This form of corrosion is usually confined to the austenitic, nonmagnetic, chrome–nickel stainless group and is evidenced by attack to the grain boundaries. The cause is precipitation of chromium-carbides to the grain

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boundaries and it results in practically complete destruction of the mechanical properties of the metal for the depth of attack. Improper heat treatment or heating in the range of 650 to 1500 °F produces the same result.

Pitting corrosion:
Pitting corrosion takes the form of localized attack in which rapid penetration may take place at several small areas of random location. Pitting is most likely to occur when chlorides or other halogens are present in an oxidizing solution.

Other factors that must be considered are temperature, aeration, quality and quantity of dissolved gases, and numerous others. For example, consider aeration. In most solutions aeration is desirable against any of the stainless steels, but can be detrimental with copper or copper base alloys.

It is always safest to secure a water analysis. This provides a starting point in an investigation. In the interpretation of a water analysis, it is necessary to recognize that an appreciable difference exists between what might be classified as satisfactory water as regards to its end use and whether or not it is corrosive to the pump handling the water. Values indicated in the analysis indicate the approximate magnitude at which noticeable corrosion and erosion can be anticipated. The interpretation of the water analysis requires a knowledge of chemistry; however, there are a few parameters to consider, including the following:

Color:
Color has little significance except to indicate its source. It may be due to mineral or organic matter in solution or as a colloid or in suspension. A yellow tinge may indicate an iron-bearing water when the iron precipitates out of solution either by oxygen or through the release of carbon dioxide. Iron salts frequently indicate that buildups can be expected on pipe and shaft surfaces.

Odors:
Odors and taste in water may result from any one or a combination of such conditions as the presence of microorganisms, either alive or dead; dissolved gases, such as hydrogen sulphide, marsh gas, carbon dioxide, or oxygen combined with organic matter; mineral substance, such as sodium chloride, iron, carbonates and sulphates; phenols and other tarry or oil wastes.

Nitrogen:
Nitrogen in any of its specified forms indicates the extent to which organic matter is present.

Chlorides:
Any amount of chlorides warns us that electrochemical corrosion can be expected. Chlorides are among the most common contaminants in natural waters. They occur in seawater, natural lakes, and some underground water sources. Chlorides may also indicate contamination from animal excreta or industrial waste.

Alkalinity:
Alkalinity exists as normal carbonate, as bicarbonate, and as hydroxyl or caustic alkalinity. The degree of alkalinity is indicated on the pH scale when the values are over 7.0, with the concentration increasing as the pH rating increases.

Acidity:
The degree of acidity increases as the value of pH decreases below 7.0. Acidity generally requires corrosion-resistant construction.

Solids:
The quantity of solids per unit volume, size of the solids, and their hardness will help guide the material selection process, especially for the bearing bushings and wear rings.

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C.2 Protective coatings

Some of the principal types of coatings include the following:

- Asphalt and coal-tar base
- Plastics, polyesters, vinyls, and epoxy (and many compounds with fillers)
- Rubber (natural and synthetic)
- Glass, porcelain ceramic

The effectiveness of any of these coatings, however, is contingent on proper surface preparation of the base metals, the smoothness of corners and welds, and on design details of critical areas of the equipment.

C.3 Materials for rotodynamic vertical pumps

Corrosion can be minimized and, in many cases, completely prevented by proper choice of materials. In designing a structure or machine, the materials engineer or chemist must be conscious not only of the overall cost of the equipment, but also to obtain the optimum in strength, corrosion, erosion, and wear resistance.

The typical material types used in vertical pump applications include:

- Cast iron and ductile iron
- Bronzes and brasses
- Steel
- Martensitic stainless steels
- Austenitic stainless steels
- Duplex and superduplex stainless steels
- Nonmetallics and composites

The specifying engineer and user must carefully select materials of construction that are compatible with the corrosive and erosive attributes of the liquid pumped.

Additional information on pump materials is available in ANSI/HI 9.1–9.5 *Pumps – General Guidelines for Types, Applications, Definitions, Sound Measurement and Decontamination.*

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Appendix D

Motors/Drivers

This appendix is included for informative purposes only and is not part of this standard. It is intended to help the user gain a better understanding of the factors referenced in the body of the standard.

D.1 Introduction

Each driver must be selected based on its ability to perform numerous functions. The correct driver selection requires a review of the application parameters. The primary function of any driver is to translate the available energy source into mechanical energy in the form of rotary motion with the required characteristics of torque and speed. The driver must be sized to transmit all anticipated torque loads including start-up and continuous operation. Additionally the driver must be sized to meet any anticipated overload conditions by design or by acceptable excursion into built-in service factors.

In addition to being able to start the pump, the driver shall be sized to meet the load requirements of the driven equipment throughout the normal operating range of the pump. These load requirements must consider torque, thrust, and inertia and must also reflect any additional requirements of accessory equipment, such as gears and fluid couplings.

D.2 Driver types

- Electric motors: Induction, synchronous, AC, DC, canned motor, submersible motor
- Engines: Diesel oil, gasoline, natural gas, crude oil
- Turbines: Gas, steam, hydraulic
- Speed increasing: Gears, fluid coupling, frequency converters
- Speed reducing: Gears, fluid coupling, frequency converters

Table D.1 — Drivers – functions and parameters for selection

Functions of a Driver	Parameters for Selection
Translate input energy into rotary motion.	Energy source available: Electrical (voltage, amperes, frequency) Steam Combustive fuel (diesel oil, gasoline, natural gas) Hydraulic
Provide required torque to drive the pump shaft at given speed.	Torque loads (transient and continuous) Speeds Service factors Shaft orientation
Accelerate from rest to operating speed within an acceptable time frame.	Pump design Pump performance specification Mass moment of inertia

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Table D.1 — Drivers – functions and parameters for selection (*continued*)

Functions of a Driver	Parameters for Selection
Provide continuous operation at constant speed or at a number of variable speeds.	Pump design Pump performance specification Mass moment of inertia
Provide the correct sense of rotation for the pump.	Pump design Pump performance specification
Allow fitting and location of coupling hub.	Coupling design
Allow safe operation in hazardous areas.	Customer specification Area classification Protection requirements Local safety regulations
Allow reliable operation in local environment.	Customer specification Operating environment Temperature, humidity, altitude Local environmental regulations Maintenance requirements
Allow efficient and cost-effective operation in the given environment.	Customer specification First-time cost, energy cost, efficiency
Provide end float limitation.	Driver type and design
Maintain required degree of unbalance at given speed (speeds).	Customer specification Operating speed(s) Limits of vibration Precision of manufacture Potential methods of balance correction
Provide smooth running operation. Stable rotordynamic operation at all defined operating speeds. Confirmed by lateral and torsional analyses and validations to prove adequate separation between operating (excitation) frequencies and rotor natural frequencies and/or sufficient rotor damping.	Rotor design characteristics Operating speed(s) Excitation frequencies
Provide smooth running operation. Stable structural operation at all defined operating speeds. Confirmed by natural frequency analyses and validations to prove adequate separation between operating (excitation) frequencies and structural natural frequencies and/or sufficient damping.	Structural design characteristics, including mounting arrangement and support Operating speed(s) Excitation frequencies

The type of driver will normally be specified by the purchaser.

In the majority of cases, after due consideration of the selection parameters identified above, the most effective and convenient method of providing the drive of a vertical pump turns out to be either vertical electric motor or horizontal engine combined with a right-angle gear. However, both methods may have limitations, which dictate the need for other solutions.

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D.3 Electric motors

Vertical electric motors, when directly mounted on the pump, become an integral part of the pump. The quality and integrity of the pump is affected by the quality of the motor being used. Therefore, it is recommended that the pump manufacturer be involved in the pump and motor procurement process.

An electric motor is a device used to convert electric power into mechanical power. This can be considered an ideal driver for a vertical pump because of its low initial cost, high efficiency, and ease of installation and maintenance.

An electric motor is normally rated in kilowatts or horsepower at a given speed. For sizing, electric motors are specified by rated power and service factor. Usual practice is to select the motor such that its rated power is at least equal to the expected maximum continuous power required by the pump. The service factor is intended to cover incidental overload; thus, its magnitude depends on the expected extent of overload. Refer to standards of the National Electrical Manufacturers Association (NEMA), latest edition, for further information.

The motor must also be sized to develop sufficient torque to accelerate the pump to full speed. For smaller pumps with over 700 kPa (gauge) (100 psig) suction pressure, the motor size may need to be chosen based on the starting torque required for the stuffing box(es) or sealing device(s).

On high specific speed pumps, especially axial flow pumps, the input power increases significantly as flow is reduced. These pumps may have to be started with an open discharge valve to prevent motor overload.

In addition to these basic sizing requirements, a number of other considerations must be taken into account to ensure proper motor selection. These include the basic motor type, motor enclosure, service factor, power supply characteristics, starting methods, ambient temperature, and altitude.

For direct motor-driven equipment, the thrust bearing in the driver should be sized so that it adequately handles axial thrust from minimum flow or shutoff to maximum flow. In addition, provisions should be made in the design of the driver to limit momentary upward movement of the rotor as a result of hydraulic shocks or upthrust within the pumping system.

D.3.1 Motor types

Motors types are defined by a combination of their input electrical characteristic and construction features. Input power is usually defined as either AC (alternating current) or DC (direct current). However, DC motors are seldom used on vertical pumps.

AC motors may be designed for use on either a single-phase or polyphase power system. Single-phase motors receive their power from a single-phase power source with two leads. Polyphase motors are typically configured with three phases and receive input from a three-wire power supply. Single-phase motors are usually induction-type machines but may be series-wound universal motors that have a commutator.

D.3.1.1 AC single-phase motors

AC single-phase motors are typically used on smaller pumps less than 7.5 kW or 10 hp and are not considered in this motor section.

D.3.1.2 AC polyphase motors

AC polyphase motors may be squirrel-cage induction, wound rotor induction, or synchronous.

Types of polyphase motors include the following:

Squirrel-cage induction: A squirrel-cage induction motor has a primary winding, known as the *stator*, which is connected to an AC power source. The secondary winding, known as the *rotor*, is constructed of aluminum or copper bars that are shorted at each end.

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Wound rotor induction: A wound rotor induction motor has an AC primary winding that is connected to a power source. The secondary winding consists of a polyphase winding connected to slip rings. Stationary brushes, riding on the slip rings, are used to either short-circuit the secondary winding or to add external impedance to the secondary circuit. Changing the secondary impedance will alter the speed and torque characteristic of a wound rotor motor.

Synchronous motor: A synchronous motor has a primary winding connected to an AC power source. The secondary circuit consists of separate pole pieces that are excited from a separate source of DC power. A synchronous motor is more expensive than a squirrel-cage induction motor. Its advantages are that its average speed is exactly proportional to the frequency of its power system and that the rotor excitation can be changed to improve system power factor. The efficiency of a synchronous motor can be the highest of all AC electric motor types. Synchronous motors may also have a permanent magnet secondary. Synchronous motors with a permanent magnet secondary will have an average speed exactly proportional to the applied frequency, but the excitation cannot be changed to provide system power factor correction. The pull-in torque required for a synchronous motor to lock into full synchronous speed must be evaluated carefully for rotodynamic vertical pump applications. Line voltage reduction at start-up must be considered. Motor torque margin must be included for the pump performance test (impeller trim) margin and start-up at off-design conditions.

D.3.2 Electric motor construction

D.3.2.1 Enclosures

An electric motor's enclosure serves many purposes. It contains the unit's internal components, protects those components from harsh environments, protects personnel from live or moving parts, and plays a major role in the unit's cooling and performance.

Electric motor enclosures can be divided into two main categories, open and totally enclosed.

Open motors are cooled by drawing in outside air, circulating it over and through the internal components, and expelling it back into the atmosphere. Although totally enclosed motors are not airtight, their internals have much less interaction with the outside atmosphere.

Totally enclosed motors typically rely on surface cooling by either free or forced convection. On some larger totally enclosed designs, cooling is achieved by circulating internal air through an externally fed heat exchanger.

Great care must be taken when selecting a motor enclosure as each has advantages in certain applications. Totally enclosed motors are most often employed in heavily contaminated environments, while open enclosures fare better in cleaner applications. Although totally enclosed units are often more efficient than open motors, more horsepower can be achieved in an open motor of a given frame size than in a totally enclosed motor of the same frame size.

Several enclosure types have been given specific designations recognized by international standards committees. Table D.2 shows a brief description of several of the most common enclosure types.

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Table D.2 — Common electric motor enclosure types (Source: NEMA MG 1)

	Designation	Definition
OPEN	Open Drip-Proof	Constructed with ventilation opening such that successful operation is not interfered with by drops of liquid or solid particles striking or entering the enclosure at any angle from 0 - 15° downward from vertical.
	Weather Protected Type I	A guarded machine constructed such that its ventilating passages minimize the entrance of precipitation and airborne particles to the electric parts.
	Weather Protected Type II	A machine that, in addition to the requirements of WPI, is constructed so that airborne particles blown in by high-velocity winds can be discharged without entering the ventilation passages leading to the electrical components. The path leading air into the internal components must include at least three abrupt changes of directions of at least 90°.
TOTALLY ENCLOSED	Totally Enclosed Nonventilated	A frame surface-cooled machine equipped only for cooling by free convection.
	Totally Enclosed Air Over	A frame surface-cooled machine intended to be provided with cooling air by external means.
	Totally Enclosed Fan-Cooled	A frame surface-cooled machine cooled by an integral fan mounted external to all enclosed parts.
	Totally Enclosed Air-Air-Cooled	Cooled by circulating internal air through a heat exchanger where it is cooled by external air.
	Totally Enclosed Water-Air-Cooled	Cooled by circulating internal air through a heat exchanger where it is cooled by water.
	Totally Enclosed Liquid-Cooled	Motor is fully submerged and requires a flow of liquid past it to dissipate heat.

D.3.2.2 Degree of protection

Regardless of what name is assigned, motor enclosures can be classified by their degree of protection. This designation consists of the letters IP followed by two numerals. The first numeral represents the enclosure's level of protection against incidental penetration of solid objects into the enclosure, or accidental human contact with live or rotating components. The second numeral defines the level of protection against the ingress or penetration of liquids (water) into the motor enclosure. Tables D.3 and D.4 define the IP designation system. For example, a motor with a degree of protection of IP13 would not allow accidental contact with live parts exceeding 50 mm (2 in) and would not be adversely affected by a spray of water up to 60° from vertical. IP designations with first numerals 4 or higher are typically used when describing totally enclosed machines.

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Table D.3 — Definition of first numeral in IP classification system

First Numeral	Definition
0	No special protection against accidental or inadvertent contact with live or moving parts or against the ingress of solid material.
1	Protection against accidental inadvertent contact with live or moving parts by a large surface of a body, such as a hand or similar object exceeding 50 mm (2 in).
2	Protection against accidental inadvertent contact with live or moving parts, such as by a finger or similar object exceeding 12 mm (0.5 in).
3	Prevents the entry of tools, wires, or other solid objects of a diameter or thickness greater than 2.5 mm (0.10 in).
4	Prevents the entry of tools, wires, or other solid objects of a diameter or thickness greater than 1.0 mm (0.04 in).
5	Complete protection against contact with live or moving parts inside enclosure. The entry of dust is not completely prevented but it does not enter in sufficient quantity to interfere with operation of the motor.
6	Complete protection against contact with live or moving parts inside enclosure. Complete protection against the ingress of dust.

Table D.4 — Definition of second numeral in IP classification system

Second Numeral	Definition
0	No special protection
1	Vertically falling drops
2	Drops falling at any angle up to 15° from vertical
3	Spray at any angle up to 60° from vertical
4	Water splashing against the enclosure from any direction
5	Water projected from a nozzle from any direction
6	Water from heavy seas or projected from powerful water jets
7	Entry of water in a harmful quantity is prevented when the unit is immersed in water under stated conditions of pressure and time
8	Immersed under water specified pressure for an indefinite time

D.3.2.3 Cooling methods

Electric motors must dissipate the heat generated within their windings in order to operate. If a unit fails to be adequately cooled, it can overheat and cause damage to itself and the driven equipment. To guard against this, thermal protection devices are available that will trigger the safe shut-down of a motor if the temperature exceeds a predetermined maximum.

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There are a variety of cooling methods used in motor design. When the cooling air is drawn from the surrounding environment, circulated around the internal components, and expelled back into the surroundings, the cooling method is called an *open circuit*. This type of cooling is only possible in open enclosure motors.

Closed-circuit cooling involves internal coolant in a closed loop that passes heat to another coolant either through the surface of the machine or a heat exchanger. This type of cooling is by definition associated with totally enclosed machines because the primary coolant remains contained within the motor.

Most motors use shaft-mounted fans to circulate air as the primary coolant. One drawback of this approach is that the velocity at which the cooling air is circulated decreases if the speed of the motor decreases. In some applications, a constant velocity of air is necessary. In these cases, separately powered fans are often employed to deliver a regular velocity of air regardless of the motor's rotational speed.

Although air is the most common fluid used as primary and/or secondary coolant in electric motor design, units can be built using others, such as refrigerant, hydrogen, nitrogen, carbon dioxide, water, and oil.

VS0-style pumps, with the motor submerged, must have a minimum flow of cooling liquid past the motor during operation to properly dissipate heat. In applications such as open channels with relatively low velocity of flow around the motor or installations where the flow will not naturally flow past the motor, a flow sleeve needs to be installed to draw flow around the motor casing and protect the motor internals from overheating. For hot pumped-liquid applications, consult the pump manufacturer.

D.3.2.4 Motor bearings

Electric motors are constructed using either antifriction or journal bearings (refer to Section 2.3.3.11.2 Thrust bearings, in this standard) with the former being used in the vast majority of motors from fractional to 375 kW (500 hp). Submersible motors use exclusively radial journal bearings on VS0-style pumps.

Antifriction bearings are widely available in a variety of sizes and configurations and are capable of accepting both radial and thrust loading. They can be operated with either grease or oil lubrication and have an initial lower cost than journal bearings. However, there are some disadvantages to the use of antifriction bearings. The limiting speed and geometry can affect motor or bearing housing design. Replacement requires the complete disassembly of the motor enclosure. The most obvious disadvantage for an antifriction bearing is its limited load-carrying capacity.

Journal bearings may be used in higher horsepower, more specialized motors. These bearings are not as widely available. Journal bearings equipped with pivot shoes can be used to help stabilize rotordynamic effects. Journal bearings have a theoretically infinite life because there is no fatigue involved in their operation.

D.3.3 Performance characteristics

D.3.3.1 Relationship of voltage and current

When AC voltage is applied to the stator, current flows through the windings. The magnetic field developed in a phase winding depends on the direction of current flow through that winding. The following chart is used here for explanation only. It assumes that a positive current flow in A1, B1, and C1 windings result in a north pole.

	Current Flow Direction	
Winding	Positive	Negative
A1	North	South
A2	South	North
B1	North	South
B2	South	North
C1	North	South
C2	South	North

D.3.3.2 Torque versus speed

Motors are designed with certain speed–torque characteristics to match speed–torque requirements of various loads. A motor must be able to develop enough torque to start, accelerate, and operate a load at rated speed.

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Torque can be calculated by transposing the formula for power.

Metric units:

$$P = \frac{\tau \times n}{9654} \qquad \tau = \frac{P \times 9654}{n}$$

US customary units:

$$P = \frac{\tau \times n}{5252} \qquad \tau = \frac{P \times 5252}{n}$$

Where:

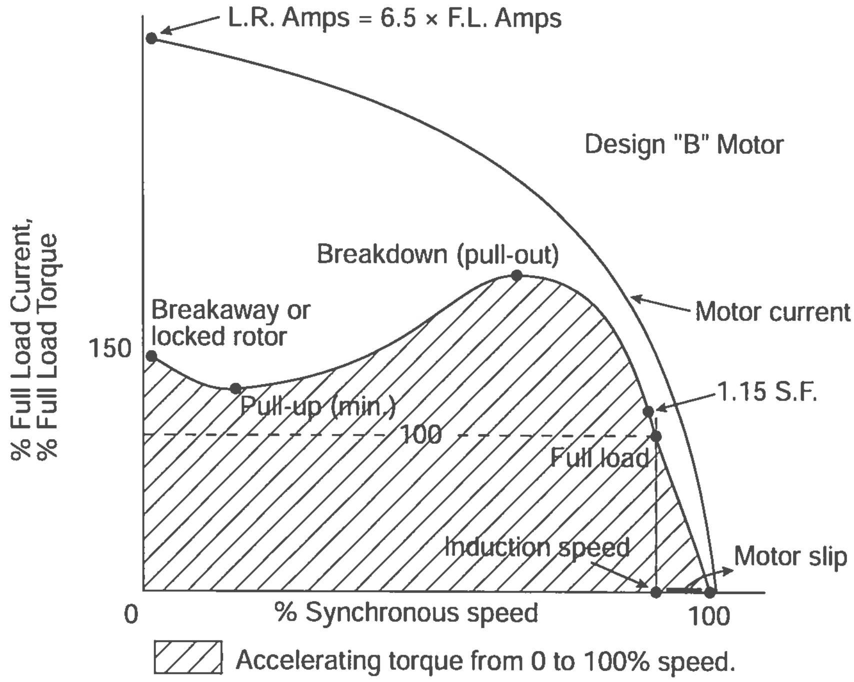
τ = torque, in N•m (lb-ft)

P = power, in hp (kW)

n = speed, in rpm

A graph, like the one shown below in Figure D.1, shows the relationship between the speed and torque the motor produces from the moment of start until the motor reaches full-load torque at rated speed.

Starting torque is also referred to as *locked rotor* (L.R.) or *breakaway torque*. This torque is developed when the rotor is held at rest with rated voltage and frequency applied. This condition occurs each time a motor is started. When rated voltage and frequency are applied to the stator there is a brief amount of time before the rotor turns. At this instant a NEMA Design B motor develops approximately 150% of its full-load torque.



S.F. = Service factor (motor), L.R. = Locked rotor, F.L. = Full load (100% torque)

Figure D.1 — Typical torque–speed curves for NEMA design AC motors

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The magnetic attraction of the rotating magnetic field will cause the rotor to accelerate. As the motor picks up speed, torque decreases slightly until it reaches the point identified on the graph as pull-up torque. As speed continues to increase from this point to the next point, breakdown torque (pull-out) torque increases until it reaches its maximum at approximately 200%. Breakdown torque is the maximum torque a motor can produce. If the motor was overloaded beyond the motor's torque capability, then it would stall or abruptly slow down at this point.

Torque decreases rapidly as speed increases beyond breakdown until it reaches full-load torque at a speed slightly less than 100% synchronous speed. Full-load torque is the torque developed when the motor is operating with rated voltage, frequency, and load. The speed at which full-load torque is produced is the slip speed or rated speed of the motor.

D.3.3.2.1 NEMA design motors

The National Electrical Manufacturers Association has assigned a simple letter designation to four of the most common three-phase AC electric motors. These vary in starting torque and speed regulation. They are all of squirrel-cage construction, and are available in many sizes. Figure D.2 shows the performance curve for each type. Note that this figure has torque on the vertical axis and speed on the horizontal.

D.3.3.2.2 Defined operating definitions (NEMA A, B, C, and D)

NEMA Design A
Design A has normal starting torque (typically 150-170% of rated) and relatively high starting current. Breakdown torque is the highest of all NEMA types. It can handle heavy overloads for a short duration. Slip ≤ 5%. A typical application is powering of injection-molding machines.

NEMA Design B
Design B is the most common type of AC induction motor sold. It has normal starting torque, similar to Design A, but offers low starting current. Locked rotor torque is good enough to start many loads encountered in industrial applications. Slip ≤ 5%. Motor efficiency and full-load power factor are comparatively high, contributing to the popularity of the design. Typical applications include pumps, fans, and machine tools.

NEMA Design C
Design C has high starting torque (greater than the previous two designs, e.g., 200%), useful for driving heavy breakaway loads. These motors are intended for operation near full speed without great overloads. Starting current is low. Slip ≤ 5%.

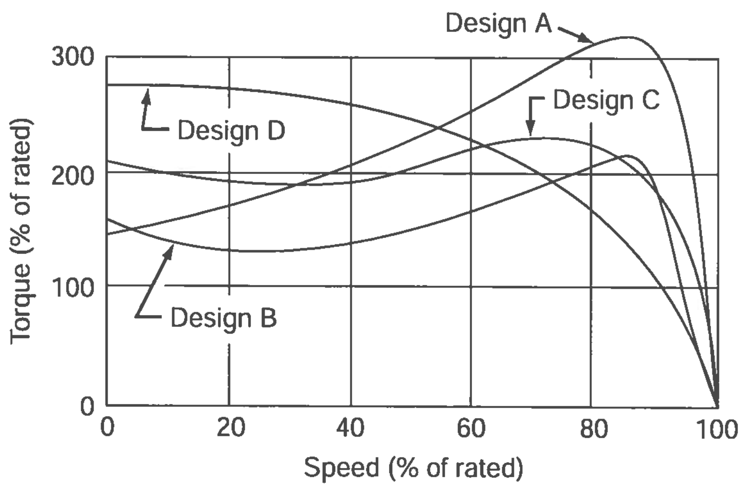

Figure D.2 — Torque–speed curves for design A, B, C, and D for AC motor

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NEMA Design D

Design D has high starting torque (highest of all the NEMA motor types). Starting current and full-load speed are low. High slip values (5-13%) make this motor suitable for applications with changing loads and attendant sharp changes in motor speed, such as in machinery with flywheel energy storage. Speed regulation is poor, making the D design suitable for punch presses, cranes, elevators, and oil well pumps. Several design subclasses cover the rather wide slip range. This motor type is usually considered a special-order item.

D.3.3.3 Efficiency

AC motor efficiency is expressed as a percentage. It is an indication of how much input electrical energy is converted to output mechanical energy. The higher the percentage, the more efficiently the motor converts the incoming electrical power to mechanical horsepower. A 22-kW (30-hp) motor with 93.6% efficiency would consume less energy than a 22-kW (30-hp) motor with an efficiency rating of 83%. This can mean a significant savings in energy cost. Lower operating temperature, longer life, and lower noise levels are typical benefits of high-efficiency motors. For additional information regarding motor efficiencies, see www.nema.org/premiummotors.

D.3.3.4 Power factor

Power factor is an electrical term used to rate the degree of the synchronization of power supply current with the power supply voltage.

It is important that the meaning of *power factor* and its effect on the electrical supply system is understood for the following reasons:

a) A low power factor can increase the cost of power to the user.

b) A low power factor can increase the cost of power transmission equipment to the user.

c) Overcorrection of the power factor (above 95% power factor) by the addition of excessive capacitance is sometimes dangerous to a motor and the driven equipment.

D.3.3.5 Service factor

A motor designed to operate at its nameplate horsepower rating has a service factor of 1.0. This means the motor can operate at 100% of its rated horsepower. Some applications may require a motor to exceed the rated horsepower. In these cases, a motor with a service factor of 1.15 can be specified. The service factor is a multiplier that may be applied to the rated power. A 1.15 service factor motor can be operated 15% higher than the motor's nameplate horsepower. For example, a 20-kW motor with a 1.15 service factor can be operated at 23 kW. (A 30-hp motor with a 1.15 service factor can be operated at 34.5 hp.)

It should be noted that any motor operating continuously at a service factor greater than 1.0 will have a reduced life expectancy compared to operating it at its rated horsepower. In addition, performance characteristics such as full-load rpm and full-load current will be affected.

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D.3.3.6 Altitude

Altitude		Derating Factor
(meters)	(feet)	
> 1000 ≤ 1500	> 3300 ≤ 5000	0.97
> 1500 ≤ 2000	> 5000 ≤ 6600	0.94
> 2000 ≤ 2500	> 6600 ≤ 8300	0.90
> 2500 ≤ 3000	> 8300 ≤ 9900	0.86
> 3000 ≤ 3500	> 9901 ≤ 11,500	0.82

Standard motors are designed to operate below 1000 m (3300 ft). Air is thinner and heat is not dissipated as quickly above 1000 m (3300 ft). Most motors must be derated for altitude. The following chart gives typical horsepower derating factors, but the derating factor should be checked for each motor.

D.3.3.7 Frequency

Frequency is defined as the rate at which alternating current makes a complete cycle of reversals. It is expressed in cycles per second. In the United States, 60 Hz (cycles per second) is the standard while in other countries 50 Hz is common. The frequency of the AC current will affect the speed of a motor, i.e., synchronous speed = (120 × *f* [frequency])/number of poles.

A variation in the frequency at which the motor operates causes changes primarily in speed and torque characteristics. A 5% increase in frequency, for example, causes a 5% increase in full-load speed and a 10% decrease in torque.

D.3.3.8 Starting

Motors shall start and accelerate to running speed a load, which has a torque characteristic and an inertia value not exceeding that required by NEMA, with the voltage and frequency specified as follows:

a) Plus or minus 10% of rated voltage, with rated frequency for induction motors.

b) Plus or minus 5% of rated frequency, with rated voltage.

c) A combined variation in voltage and frequency of 10% (sum of absolute values) of the rated values, provided the frequency variation does not exceed plus or minus 5% of rated frequency.

Performance within these voltage and frequency variations will not necessarily be in accordance with the standards established for operation at rated voltage and frequency (NEMA MG 1).

The limiting values of voltage and frequency under which a motor will successfully start and accelerate to running speed depend on the margin between the speed–torque curve of the motor at rated voltage and frequency and the speed–torque curve of the load under starting conditions. Because the torque developed by the motor at any speed is approximately proportional to the square of the voltage and inversely proportional to the square of the frequency, it is usually desirable to determine what voltage and frequency variations will actually occur at each installation, taking into account any voltage drop resulting from the starting current drawn by the motor. This information and the torque requirements of the driven machine define the motor-speed–torque curve at rated voltage and frequency, which is adequate for the application.

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D.3.3.8.1 Number of starts

Induction motors having horsepower ratings and performance characteristics in accordance with MG1 Part 12 shall be capable of accelerating without injurious heating load inertia referred to the motor shaft equal to or less than the values required by NEMA under the following conditions:

a) Applied voltage and frequency in accordance with NEMA.

b) During the accelerating period, the connected load torque is equal to or less than a torque that varies as the square of the speed and is equal to 100% of rated load torque and rated speed.

c) Two starts in succession (coasting to rest between starts) with the motor initially at the ambient temperature or one start with the motor initially at a temperature not exceeding its rated load operating temperature.

D.3.4 Classified (or regulated) areas (hazardous atmospheres)

D.3.4.1 Explosion-proof and dust ignition-proof motors

The term *explosion-proof* is sometimes used to define enclosure requirements for motors used in hazardous locations. *Hazardous locations* are defined as places where flammable volatile liquids, flammable gases, combustible dusts, and easily ignitable fibers or materials producing combustible flyings are handled, manufactured, stored, or used. Technically, the term *explosion-proof* defines enclosure requirements for motors used in locations where a hazardous gas or vapor is present. *Dust ignition-proof* defines enclosure requirements for motors used in a location where ignitable dust is present. Following are the definitions from the National Electrical Manufacturers Association (NEMA) MG 1:

Explosion-Proof Machine

An explosion-proof machine is a totally enclosed machine whose enclosure is designed and constructed to withstand an explosion of a specified gas or vapor that may occur within it and to prevent the ignition of the specified gas or vapor surrounding the machine by sparks, flashes, or explosions of the specified gas or vapor that may occur within the machine casing.

Dust Ignition-Proof Machine

A dust ignition-proof machine is a totally enclosed machine whose enclosure is designed and constructed in a manner that will exclude ignitable amounts of dust or amounts that might affect performance or rating, and that will not permit arcs, sparks, or heat otherwise generated or liberated inside of the enclosure to cause ignition of exterior accumulations or atmospheric suspensions of a specific dust on or in the vicinity of the enclosure. Successful operation of this type of machine requires avoidance of overheating from such causes as excessive overloads, stalling, or accumulation of excessive quantities of dust on the machine.

D.3.4.2 National Electrical Code

D.3.4.2.1 Hazardous locations and materials: class, division, group

Articles 500 through 504 of ANSI/NFPA 70, *National Electrical Code*, serve as the basis for the NEMA definitions. The National Electrical Code provides guidance for the use of electrical equipment in hazardous locations and lists three distinct types of hazardous areas:

Class I – Locations where flammable gases or vapors are or may be present in the air in quantities sufficient to produce explosive or ignitable mixtures.

Class II – Locations that are hazardous because of the presence of combustible dust.

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Class III – Locations that are hazardous because of the presence of easily ignitable fibers or flyings, but not in sufficient quantities to produce ignitable mixtures.

Locations are further categorized as Division 1 or Division 2, depending on whether the hazard is always present under normal operation (Div 1) or only under abnormal conditions (Div 2), such as an accidental container rupture or a malfunction of handling or processing equipment. Typically, a local authority is responsible for determining the degree of hazard present in various plant areas. The final authority on safe equipment installation rests with a "local authority having jurisdiction" – an organization, office, or individual responsible for approving equipment, an installation, or a procedure. A Division 1 location will usually require the use of an approved (listed) motor. Purged, pipe-vent, and submerged motors are other Division 1 options. Motors without sparking devices typically can be used in Division 2 locations.

Hazardous materials themselves are grouped based on similar characteristics. Flammable gases and vapors are covered under Class I, Groups A, B, C, or D. The gases or vapors are grouped by the severity of explosion pressure expected, the extent of flame propagation between parts, and related characteristics of the material. Ignitable dusts are covered under Class II, Groups E, F, or G. Dusts are grouped by their combustibility, penetrability, conductivity, ability to contribute to creation of an ignition source, blanketing effect, and ignition temperature.

D.3.4.2.1.1 Examples of hazardous materials classes and groups

Class I, Group A – Acetylene. No listed Division 1 motors are available. Group A gases are too explosive to be contained. Nonlisted Division 2 motors are available.

Class I, Group B – Hydrogen. No listed Division 1 motors are available. Group B gases are too explosive to be contained. Nonlisted Division 2 motors are available.

Class I, Group C – Ethylene. Division 1 and 2 motors are available.

Class I, Group D – Gasoline. Division 1 and 2 motors are available.

Class II, Group E – Magnesium dust. Division 1 and 2 motors are available.

Class II, Group F – Coal dust. Division 1 and 2 motors are available.

Class II, Group G – Grain dust. Division 1 and 2 motors are available.

D.3.4.2.1.2 Explosion-proof motor requirements

- The enclosure must be able to contain an explosion; enclosure parts have minimum tensile strength and thickness requirements. Fasteners must be sized to contain the explosion and have minimum thread engagement requirements.

- The motors must not ignite the atmosphere surrounding the motor. Carefully controlled clearances (flame paths) between mating parts vent hot gasses while preventing propagation of flames from inside the motor to external atmosphere. The flame path includes the shaft clearance so shaft side-loading capability can be affected. Enclosure surface temperature must be limited to preclude heat from causing ignition of a gas or vapor in the vicinity of the motor. Totally enclosed fan-cooled motors must have a nonsparking fan.

- Motors must be marked with the class, group, and an operating temperature code. The temperature marking must not exceed the ignition temperature of the specific gas or vapor to be encountered. The motor manufacturer must therefore be provided with the class, group, and operating temperature limits any time an explosion-proof motor is specified.

D.3.4.2.1.3 Dust ignition-proof motor requirements

- The enclosure must exclude amounts of dust that can either ignite or affect performance.

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- The motors must not ignite the atmosphere surrounding the motor. Carefully controlled clearances (flame paths) between mating parts prevent the entry of dust into the motor while preventing propagation of flames from inside the motor to external atmosphere. Enclosure surface temperature must be limited to preclude heat from causing ignition of a dust accumulated on the enclosure or suspended in the atmosphere in the vicinity of the enclosure. The motors must not generate arcs or sparks that could ignite internal or surrounding dust. Totally enclosed fan-cooled motors must therefore have a nonsparking fan.

Motors must be marked with the class, group, and an operating temperature code. The temperature marking must not exceed the ignition temperature of the specific dust to be encountered. The motor manufacturer must therefore be provided with the class, group, and operating temperature limits any time an explosion-proof motor is specified.

D.4 Variable-speed drives

As with any centrifugal pump, there is often sizable energy savings in changing the total head and rate of flow of a vertical pump through varying the speed of the pump rather than by throttling the discharge with a valve. Many types of variable-speed drivers are on the market and have been used to drive vertical pumps, such as variable-frequency (current or voltage), wound rotor motors, hydraulic and magnetic couplings, steam and gas turbines, and gasoline or diesel engines.

All of these and other types of variable-speed drivers have their particular advantages and disadvantages, which should be evaluated for each specific application. A review should be made with the pump manufacturer prior to the final selection. It is recommended that the pump manufacturer coordinate all components and controls.

Variable-speed drivers can add to the complexity of the system and may dictate that a thorough analysis of the pump/driver assembly be performed. Some special items of possible concern follow: the potential for a reverse torque from variable-frequency drives, which could unscrew the line-shaft couplings; the adequacy of fan cooling on electric motors with VFDs; the mass that may be added to the top of the discharge head of the pump and its effect on structural vibrations; line-shaft critical speed considerations; and torsional pulsations that could excite a rotor critical speed. Drives equipped with tilting pad thrust bearings, such as found on large motor, high-thrust designs and submersible motors for VS0-style pumps, require a minimum rotational speed (rpm) for the fluid film-type thrust bearing to support the rotor loads.

For additional information, see ANSI/HI 9.6.4 *Rotodynamic Pumps for Vibration Measurement and Allowable Values*.

D.5 Gears

Gears must be sized to adequately handle pump torque, thrust, and inertia requirements. Gears should be rated for continuous duty and should have an adequate service factor. Right-angle gears are also quite common with vertical pumps, whether in conjunction with a variable- or constant-speed driver. Right-angle gears are most commonly used with engine drivers where electric power is not available. Gear units may also be used to change speed or reverse the direction of rotation from that of the driver.

D.6 Deceleration devices

In some applications, it is desirable to provide additional rotating inertia to a pump to slow its rate of deceleration when power from the driver is cut off. This slower deceleration may be necessary to maintain some limited flow and head for a longer period of time. This allows more time for check valves and other flow-control devices to work or to mitigate water hammer. The result is less chance of damaging backflow through the pump or water hammer effects on the system.

The additional rotating inertia is usually provided by adding a flywheel to the drive train. The flywheel may be mounted on its own bearings, may be part of the pump, or may be mounted on the end of the motor shaft.

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The moment of inertia of the flywheel might be equal to or greater than the moment of inertia of the pump/motor combination. While larger flywheels would increase the coast-down time of the pump, they are also more costly, and the driver size may have to be increased to accelerate the increased inertia of the system.

Flywheel applications should be carefully analyzed to match need and performance before they are installed. On spin down, a vertical pump will normally pass through one or more structural natural frequency modes. With the added rotational inertia, the additional transient time at resonant conditions needs to be considered.

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Appendix E

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This appendix is included for informative purposes only and is not part of this standard. It is intended to help the user gain a better understanding of the factors referenced in the body of the standard.

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Appendix F

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.

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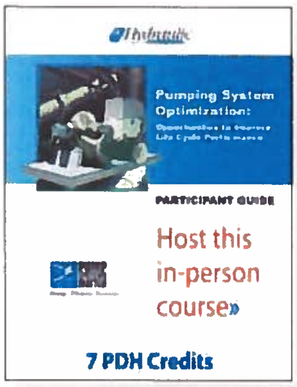

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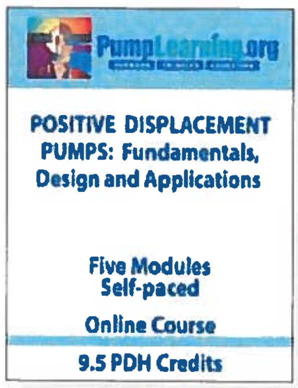

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