ANSI/HI 1.3-2013

Approved American National Standard

ANSI

American National Standard for

Rotodynamic Centrifugal Pumps

for Design and Application

ANSI/HI 1.3-2013

Hydraulic Institute

6 Campus Drive
First Floor North
Parsippany, New Jersey
07054-4406
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Sponsor

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Approved December 19, 2013

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 manufacturers and to further the interests of the public in such matters as are involved in manufacturing, engineering, distribution, safety, transportation and other problems of the industry, and to this end, among other things:

a) To develop and publish standards for pumps;

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

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

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

e) To support educational and research activities;

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

Purpose of Standards

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

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 sent 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 the content of an Institute Standard or an answer provided by the Institute to a question such as indicated above, the point in question shall be sent in writing to the Technical Director of the Hydraulic Institute, 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 using the ANSI canvass procedure.

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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 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

Consensus for this standard was achieved by use of the canvass method. The following organizations recognized as having interest in rotodynamic centrifugal pumps for design and application 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 and Consulting, LLC
Bechtel Power Corporation
Black & Veatch Corp.
ekwestrel Corp.
J.A.S. Solutions Ltd.
KCWTD
Kemet Inc.
Patterson Pump Company
Pentair Water
Pump Design, Development & Diagnostics
Sulzer Pumps (US) Inc.
TACO, Inc.
The Gorman-Rupp Company
Weir Floway, Inc.
Xylem Inc.

Committee list

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

Chair – Al Iseppon, Pentair Water
Vice-chair – Joseph Salah, Sulzer Process Pumps (US) Inc.

Committee members	Company
Greg Case	Pump Design, Development & Diagnostics
Michael Cropper	Sulzer Pumps (US) Inc.
Michael Cugal	Weir Hazleton, Inc.
Lucian Dobrot	TACO, Inc.
Mike Noble	Lewis Pumps
Aleksander Roudnev	Weir Minerals North America
Gary Saylor (Alternate)	Weir Hazleton, Inc.
LeRoy Sell	PumpWorks 610

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1.3 Design and application

The purpose of this standard is to provide a guide for the design and application of rotodynamic centrifugal pumps for various services. This is not an attempt to cover all phases of rotodynamic pump design and application but an endeavor has been made to recognize and identify the application requirements of the most common industry segments. Principal features of pumps and the necessary precautions for proper use are pointed out.

Rotodynamic pumps are kinetic machines in which energy is continuously imparted to the pumped fluid by means of a rotating impeller, propeller, or rotor. The most common types of rotodynamic pumps are centrifugal (radial), mixed flow, and axial flow pumps.

Centrifugal pumps use bladed impellers with essentially radial outlet to transfer rotational mechanical energy to the fluid primarily by increasing the fluid kinetic energy (angular momentum) and also increasing potential energy (static pressure). Kinetic energy is then converted into usable pressure energy in the discharge collector.

It can be considered that a rotodynamic pump (centrifugal pump) thereby converts liquid that is at a low pressure to a higher pressure by the use of a rotating shaft that spins a specially designed disk(s) within the boundaries of an enclosed vessel. This transaction converts rotating mechanical energy into hydraulic energy by increasing the fluid kinetic energy (angular momentum).

In the pump industry, the vessel described above is referred to as the *casing*. It has an inlet or suction port for the low-pressure liquid to enter the vessel and a discharge port for the high-pressure liquid to exit. There is a seal (or seals) to control the leakage of liquid where the shaft penetrates the casing. A motor, turbine, or engine is used to drive the rotating shaft. The special disk attached to the shaft is called the *impeller*. There are semiradial integral passageways enclosed within or exposed on the impeller. The liquid enters at the lower diameter of these passageways. The mechanical energy of the rotating shaft is transferred to the impeller. The impeller then increases the liquid kinetic energy as it moves through the impeller passageway to the outer diameter. As the liquid exits the outer diameter of the impeller it enters a discharge collector within the stationary casing. The collector converts kinetic energy into pressure energy through a diffusion process.

Both the impeller vane passage area and the discharge collector inlet area (cutwater or throat area) are controlled in size to regulate the rate of fluid pumped (capacity) and to define a certain rate of flow typically known as the *pump best efficiency point* (BEP). The pumped fluid exits the casing via the pump discharge port or outlet.

1.3.1 Scope

This standard is for rotodynamic centrifugal, regenerative turbine, and Pitot tube pumps of all industrial/commercial types except vertical single and multistage diffuser pump types. It includes design and application.

Included rotodynamic centrifugal pump types are as shown in Figure 1.3.1.

1.3.2 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.2, and 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, casing, bowl, or diffuser). Within these lines of demarcation there are pump types that can be clearly identified to fit into each of the defined categories.

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

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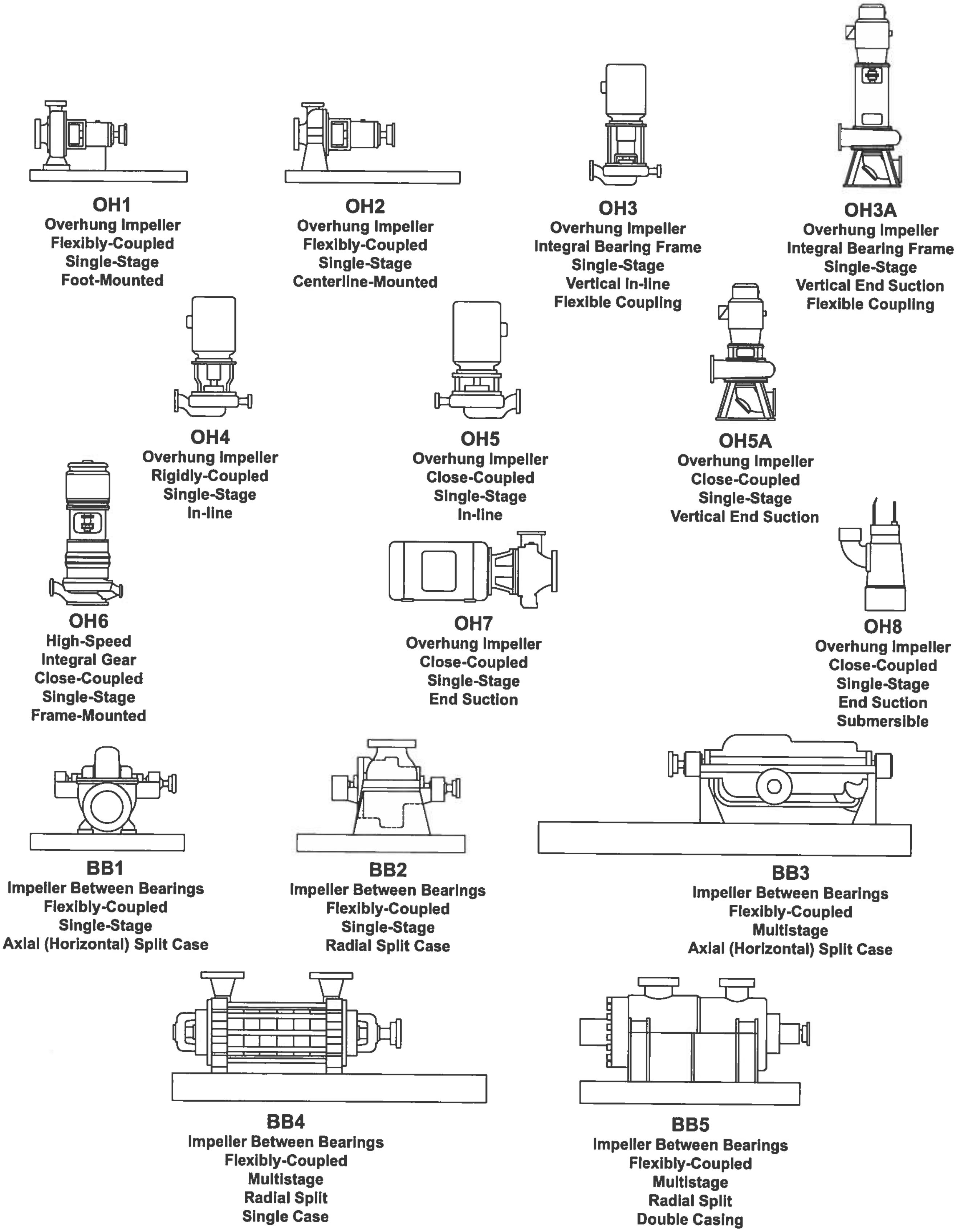

Figure 1.3.1 — Rotodynamic centrifugal pump types

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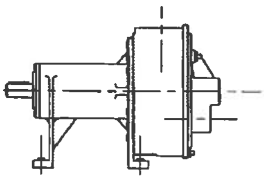
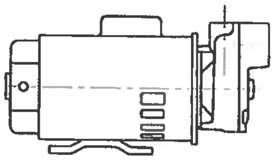
Appendix B provides an identification and introduction into such arrangements and discusses any design and application considerations that may be considered relevant to these specific configurations.

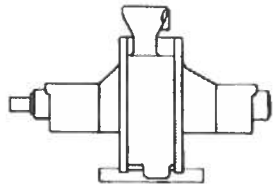
Pump types included in Appendix B are as follows:



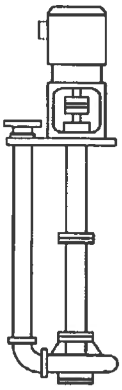
Pitot Tube Pump (unclassified)



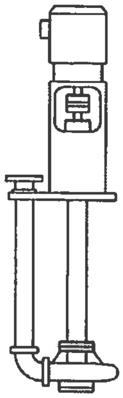
RT1- RT2
Regenerative Turbine
Overhung, Close-Coupled
Side Channel & Peripheral



RT3-RT4
Regenerative Turbine
Between-Bearings
Side Channel & Peripheral



VS4
Sump Pump
Line-shaft Design



VS5
Sump Pump
Cantilever Shaft Design

1.3.2.1 Introduction to pump industry segments and general applications

To identify the most appropriate pump configuration for a specific application, specific knowledge relevant to the industry segment served is needed.

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The following is a list of industry-specific application categories established as a guideline:

Pump Industry Segments	General Services
• Chemical industry • Petroleum Oil production Oil and gas transportation (pipeline) Hydrocarbon processing • Pulp and paper • Slurry applications • Water and wastewater • Irrigation applications • Residential applications • Electric power industry • Cooling tower • Fire pumps	However, within these application categories many services are described that are not unique to one segment of industry. For this purpose **Appendix A** is included to cover numerous aspects of **General Service Applications** and operational aspects that are commonly of interest across a variety of industry segments. Principal features of pumps and the necessary precautions for proper use are pointed out. Some elements of pump service are also covered by this category. These services cover a diverse range of commonly encountered pump application issues.

1.3.2.2 Preferred units for pump applications

Preferred terms, units, and symbols to be used in the technology of pump applications are shown in Table 1.3.2.2.

Comments on preferred units for 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 in related textbooks and literature.

Metric units

When calculating the value for specific speed and suction specific speed, the unit of measurement used within this standard for rate of flow is cubic meter per second (m^3/s).

(An alternative method of calculating this value is to use cubic meter per hour (m^3/h) as the unit of measurement for rate of flow, which then results in a value which is $[3600]^{0.5}$, i.e., 60 times greater).

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).

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Table 1.3.2.2 — 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	Frequency	cycles/second (hertz)	Hz	cycles/second (hertz)	Hz	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	dimensionless	—	dimensionless	—	1
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 $n_s = \frac{n(Q)^{0.5}}{(H)^{0.75}}$	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 (force)/ 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 = $\frac{n(Q)^{0.5}}{(NPSH3)^{0.75}}$	Index number	—	Index number	—	0.0194
s	Specific gravity	dimensionless	—	dimensionless	—	1
t	Temperature	degree Celsius	^{o}C	degree Fahrenheit	^{o}F	$(^{o}F - 32) \times {}^{5}/_{9}$
τ (tau)	Torque	newton-meter	N·m	pound (force)-foot	lbf-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.

[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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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 rpm

Q = total pump flow rate, in m^3/s (gpm)

H = head per stage, in m (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.

Suction specific speed: An index of pump suction operating characteristics determined at the BEP rate of flow with the maximum diameter impeller. 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.

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1.3.3 Casings

The function of a discharge casing is to collect output from the rotating impeller, decrease the velocity momentum of liquid leaving the impeller before it reaches the next stage impeller or pump discharge, and to transform increased kinetic energy of liquid at the impeller outlet into pressure.

The three basic discharge casing types are the volute, diffuser/collector, and concentric casing. Each has specific purposes and advantages as shown in the following sections. The casing provides the method for connecting the pump to the system by any number of nozzle configurations.

It is possible to arrange each type of casing for single-stage, two-stage, and multistage configurations. The number of stages used in any given casing is dependent on required total head, flow, and limiting mechanical parameters, such as speed and maximum working pressure.

1.3.3.1 Single volute

This is the most common casing style due to relative ease of manufacture and accessibility for inspection. An impeller discharges into a single spiral-shaped passage with one cutwater (tongue) that directs the liquid into the system or into the next stage of a multistage pump. The volute has a constantly increasing area cross section from the tongue, around the casing, to the discharge nozzle. The typical design criterion (see Figure 1.3.3.1) is for the liquid exiting the impeller to maintain a constant mean velocity or constant velocity momentum or slow slightly through the spiral to the discharge at the design point.

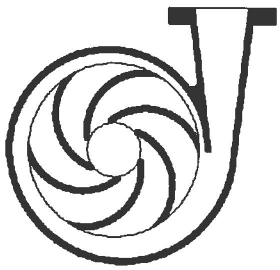

Figure 1.3.3.1 — Single volute casing

Radial thrust on the impeller varies with pump rate of flow, being lowest near BEP, and higher at reduced or increased flow rates. The thrust at off BEP can be very high for large-diameter impellers producing high head. Radial thrust also varies with impeller diameter, impeller width, and total head. See Section 1.3.5.1 for method of calculation. Shaft deflection, combined maximum stress and bearing loads must be kept within acceptable limits by various means for best operation.

1.3.3.2 Double volute

An impeller discharges into two spiral passages with two cutwaters (tongues). The pumped fluid then discharges via these two passages into a system or into the next stage of a multistage pump. The cutwaters are usually diametrically opposed in the casing. Care must be taken in the design to minimize the loss of pump efficiency. With properly designed passages, radial thrust is minimized, especially at off BEP flows. The double volute design (see Figure 1.3.3.2) is typically used to reduce shaft deflections and bearing loads to permit use of a smaller shaft and bearing sizes or to prolong the life of the pump. The casing complexity is greater than that for the single volute type due to the inaccessibility of the outside chamber.

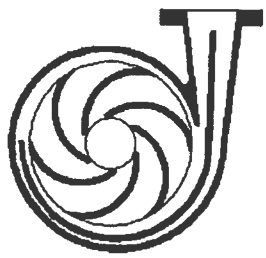

Figure 1.3.3.2 — Double (dual) volute casing

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1.3.3.3 Diffuser

An impeller discharges into multiple divergent passages (normally two or more) with the outer casing functioning as a collector, directing fluid into the pump discharge or the next pump stage. While net radial thrust exists, the multiple, equally spaced diffuser passages minimize the magnitude to be considered negligible. The collector is typically circular in shape. Designs also exist where the diffuser discharges into a more traditional volute casing (see Figure 1.3.3.3).

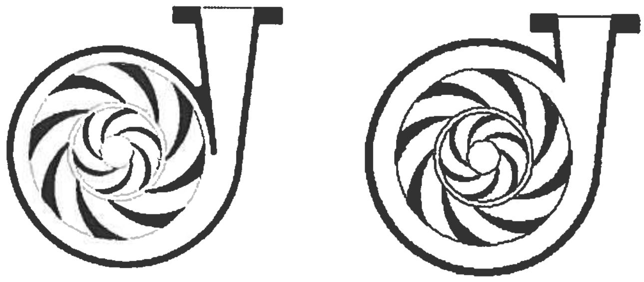

Figure 1.3.3.3 — Diffuser casing

1.3.3.4 Circular (concentric) casing

An impeller discharges into a circular collector with a single discharge port. The casing is generally concentric with the impeller (see Figure 1.3.3.4). It is mainly used for its simplicity. It is very common in fabricated pump casings. A circular (concentric) casing is often used where efficiency is not a concern, however, it may improve the efficiency of very low specific-speed pumps. It is also one of the casing types used to improve the wear life in slurry pumps. Refer to ANSI/HI 12.1–12.6 *Rotodynamic (Centrifugal) Slurry Pumps for Nomenclature, Definitions, Applications, and Operations.*

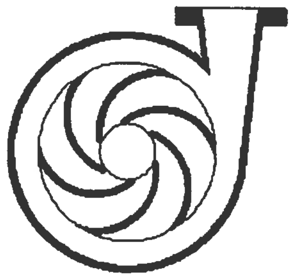

Figure 1.3.3.4 — Circular (concentric) casing

1.3.3.5 Multistage arrangements

There are many service applications that require high pressures that are outside of practical limits of a single-stage pump performance. (One single-stage being defined as one impeller and its related discharge collector.) Multistage arrangements are made with all of the basic casing hydraulic profiles. Single volutes, double volutes, and diffuser casing designs can all be readily configured into multistage settings to allow increasing head to be generated. The final total developed head is a multiple of the single-stage head and the number of stages.

Fluid enters the first stage or suction impeller from the suction inlet channel and subsequently passes through a given number of series impellers via each volute or diffuser casing and suitably arranged crossover channels until it is discharged from the final-stage impeller into the discharge channel.

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1.3.3.5.1 Type BB3 between-bearings axial split volute

Type BB3 between-bearings axial split volute casing (see Figure 1.3.3.5.1) designs are arranged in back-to-back configuration and can be successfully made in stage settings of between 3 and 14 (or more) depending on pump size and impeller diameter. The back-to-back terminology describes the way the stages are arranged with an equivalent number (of stages) oriented in opposing banks either side of a center stage (high-pressure breakdown) bushing. The opposing impellers tend to balance the hydraulic axial thrust loads and thereby minimize the thrust loads on the bearings.

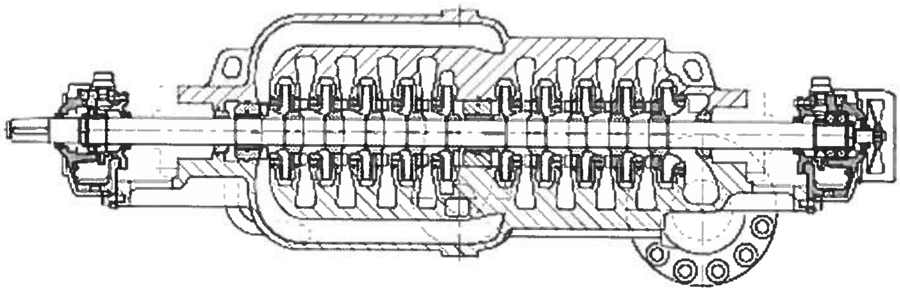

Figure 1.3.3.5.1 — Type BB3 between-bearings axial split multistage pump

1.3.3.5.2 Type BB4 between-bearings radial split single case

Type BB4 between-bearings radial split casing (see Figure 1.3.3.5.2) designs are normally arranged with diffuser casings and impellers in a stacked in-line configuration and can be also successfully made in stage settings between 3 and 14 (or more) depending on pump size and impeller diameter.

The stacked in-line terminology describes the way the casing and impellers are arranged with all stages and impellers oriented (stacked) in the same direction. To provide compensation for the net resulting thrust loads, a balance device such as piston or disk is located at the high-pressure end.

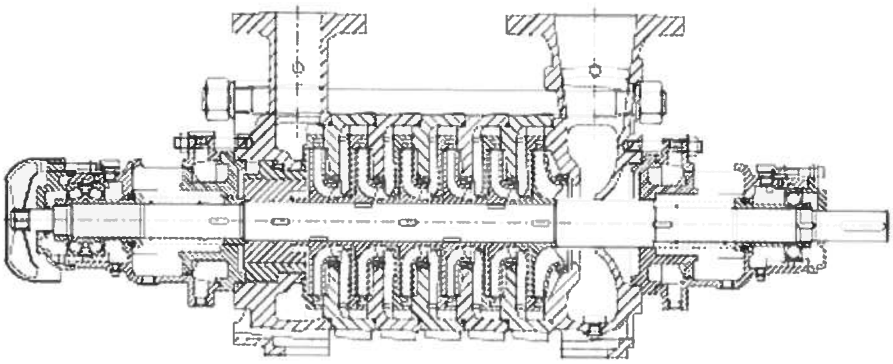

Figure 1.3.3.5.2 — Type BB4 between-bearings radial split single case multistage pump

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1.3.3.5.3 Type BB5 between-bearings radial split double case volute

Type BB5 between-bearings radial split double casing volute pump (see Figure 1.3.3.5.3) designs all are essentially double case versions of type BB3. The volute style casing can be arranged as either back-to-back or an in-line design. The double casing refers to the outer casing or barrel, which forms the pressure boundary and allows for safe containment of hazardous, toxic, corrosive fluids at extremes of pressure and temperature.

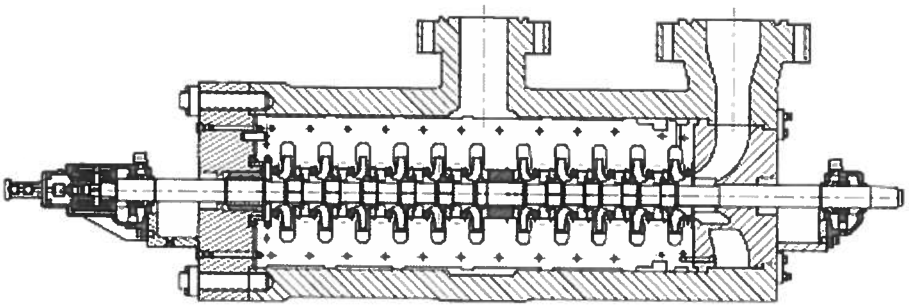

Figure 1.3.3.5.3 — Type BB5 between-bearings radial split double case multistage volute pump

1.3.3.5.4 Type BB5 between-bearings radial split double case diffuser

Type BB5 between-bearings radial split double casing diffuser pump designs all are essentially double case versions of type BB4. However, configurations can be made either back-to-back or stacked.

1.3.3.5.4.1 Stacked in-line arrangement

For high-speed pump designs, the stacked in-line arrangement (see Figure 1.3.5.5.4.1) is most prevalent because it provides for the heavy shaft design and short bearing span required for stable rotor-dynamic performance.

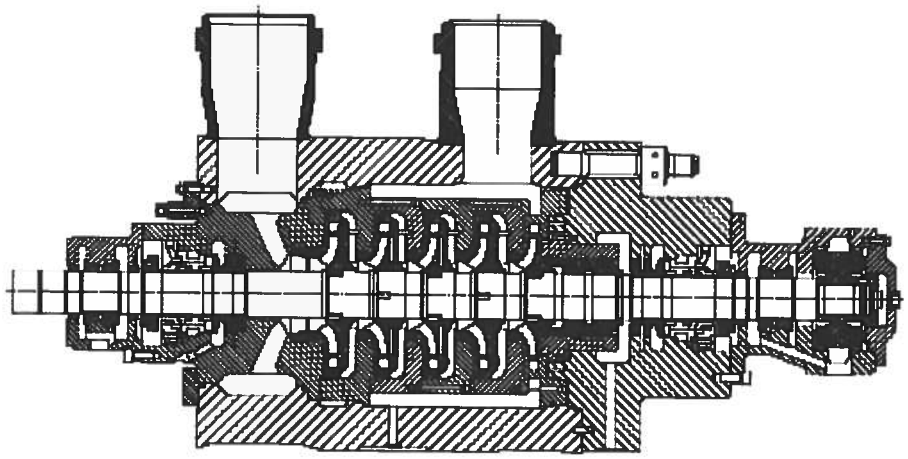

Figure 1.3.5.5.4.1 — Type BB5 between-bearings radial split double case multistage pump with stacked in-line diffuser construction

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1.3.3.5.4.2 Back-to-back arrangement

The back-to-back type BB5 diffuser pump arrangement (see Figure 1.3.3.5.4.2) may be preferred for certain high-speed, high-pressure applications.

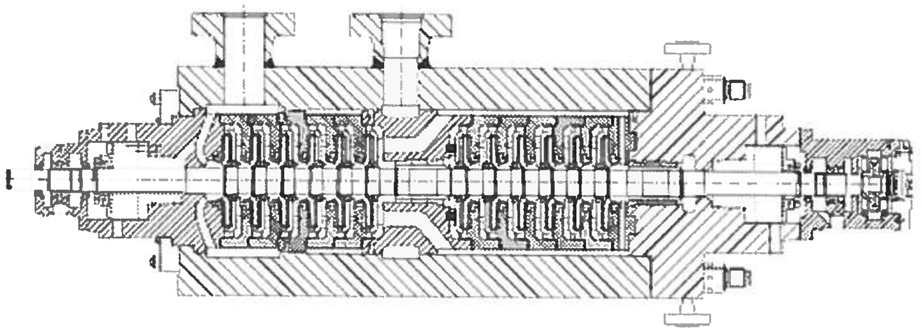

Figure 1.3.3.5.4.2 — Type BB5 between-bearings radial split double case multistage pump with back-to-back diffuser construction

1.3.3.6 Corrosion allowance for metallic rotodynamic pumps

The internal walls of pressure-containing pump components subjected to corrosive attack shall be provided with additional metal thickness over and above that required to meet the design conditions of deflection, pressure, stress, nozzle loading, casting tolerance, and temperature. The end user should be cognizant of the manufacturer's corrosion allowance in the selection of pumping equipment.

Material added for this purpose need not be the same thickness throughout if different rates or types of corrosion attack are expected. Typical corrosion allowances often specified for cast components in contact with the pumped liquid are:

• Water pumps	1.5 mm	(0.06 in)
• Hydrocarbon processing pumps	3 mm	(0.12 in)
• Chemical pumps	3 mm	(0.12 in)
• Boiler feed pumps	3 mm	(0.12 in)

The above criteria relate to pumps located in noncorrosive operating environments. A pump submerged in a sump whose external surfaces are also subjected to corrosion may require additional corrosion allowance.

1.3.4 Impellers

The bladed impeller rotated by an appropriate driver is a principal element of a rotodynamic pump.

Transfer of energy from impeller to liquid is conducted by way of dynamic interaction between impeller vanes and pumped liquid. Both liquid pressure and velocity increase during this process.

1.3.4.1 General impeller types

Impellers described in the following paragraphs are used in a wide variety of services. For any given service, one type may be preferred over the others. In most cases, however, the users' and manufacturers' experience is the best guide for their selection.

Other special impeller types (such as vortex) may be offered for special applications.

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Rotodynamic pump designs are generally described as any of three types: radial flow, mixed flow, or axial flow. See the chart below for examples (Figure 1.3.4.1). Radial flow impellers are designed such that the liquid exits purely radially or perpendicular to the shaft centerline. They have lower specific speeds, in the range n_S 10 to 50 (N_S 500 to 2500), and most often are used for lower flow, high head applications. As design flow increases, specific speed increases and the impeller will become more axial in its configuration, with fluid flow in line with the shaft centerline. Fully axial impellers produce very high flow rates with little head. Between these two extremes the liquid exit angle transitions from radial to axial as shown in the figure below. These transitional designs are referred to as *mixed flow impellers*.

Special impeller types such as Pitot tube and regenerative turbine are discussed in Appendix B.

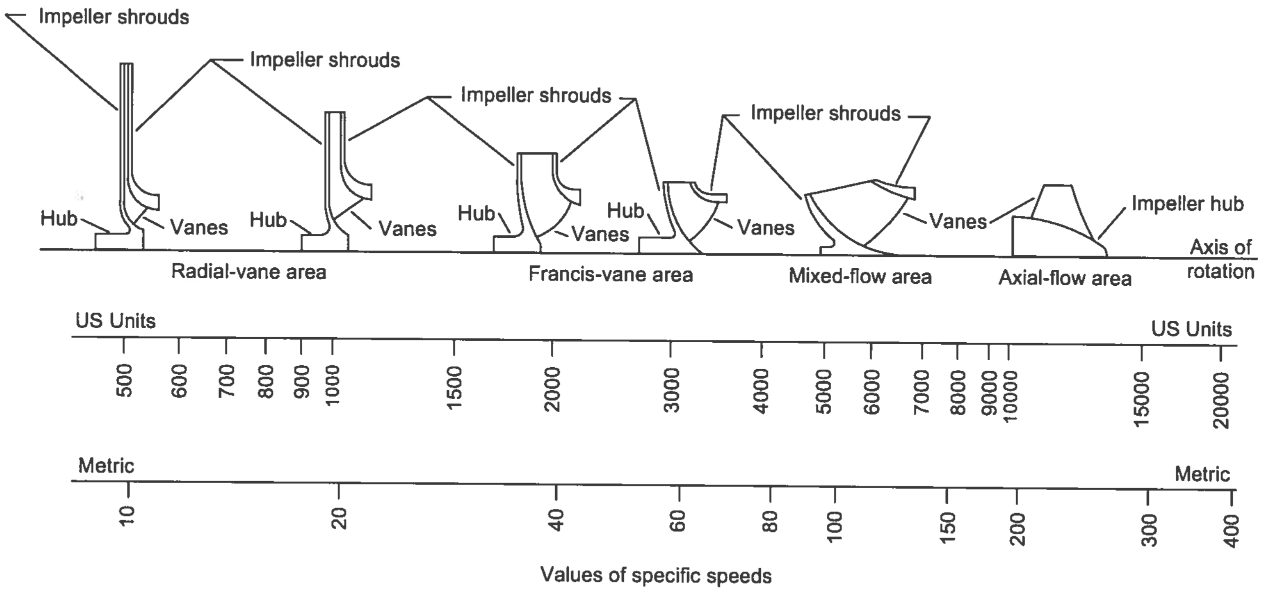

Figure 1.3.4.1 — General impeller types

1.3.4.2 Single suction

Impellers of this type (see Figure 1.3.4.2) have a single inlet for liquid entry, from where it is discharged either radially or semi-axially. The casing can be any of the three basic types described earlier. Axial hydraulic loads on the shrouds of this style impeller are not balanced due to the impeller's asymmetrical design. Various means to control axial imbalance are used, such as balance chambers, recirculation lines, and pump-out vanes. A pressure-balancing area (balance chamber) is formed by including a wear ring on the hub shroud of the impeller. The pressure in this area is then reduced by either holes through the shroud to the eye or through a recirculation line to the pump inlet piping. Another method to reduce pressure is the use of pump-out vanes, which are vanes cast integral with, or otherwise attached to, the hub shroud as a means to pump the liquid from behind the impeller. The net axial thrust imbalance must be considered in the design and specification of support bearings.

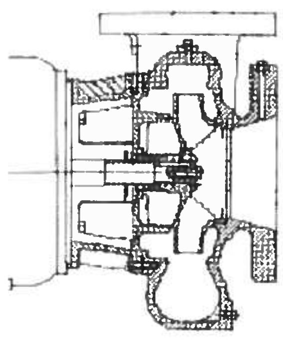

Figure 1.3.4.2 — Single suction impeller

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1.3.4.3 Double suction

Double suction impellers (see Figure 1.3.4.3) are used as pump flow rates increase (and values of NPSHA decrease). Impellers of this type have dual, opposed inlets for liquid entry, from where it is discharged radially into a common stream. For a given diameter, this impeller type provides a greater net inlet area than an equivalent single suction impeller and thereby allows for operation where NPSHA is restricted for the desired pump operating speed. This type of design normally uses a shaft with bearings at each end. This results in two mechanical seal chambers or stuffing boxes as well as two bearing housings, one on either end of the shaft with the impeller (or impellers) mounted between. In theory, the symmetrical nature of the impeller design (in conjunction with proper installation piping) balances out the hydraulic axial forces in each direction. The resultant axial thrust generated should therefore be negligible when operating the pump under stable conditions.

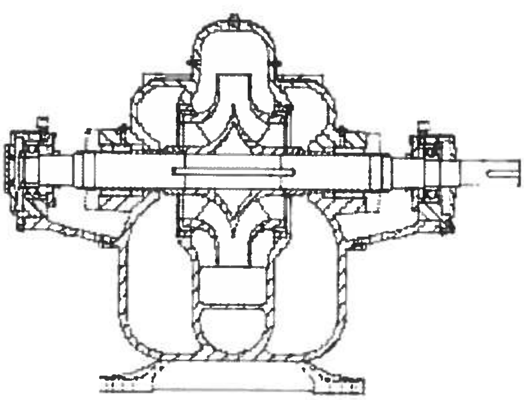

Figure 1.3.4.3 — Double suction impeller

In practice the suction flow may favor one side of the impeller relative to the other due to poor piping practices. Recognizing this possibility, the outboard bearing selection may include axial thrust capability. The hydraulic radial thrust is produced by the impeller and the shaft, and impeller weight is divided evenly between the two bearing housings.

1.3.4.4 Multistage pumps

Many pumps are built in multistage arrangements to produce a total head that a single impeller cannot effectively produce at the same operating speed. The head at each stage is additive, while the rate of flow remains the same. Thus, for example, when a single-stage pump at a given speed delivers 225 m^3/h (1000 gpm) at 152.5 m (500 ft) of head, a four-stage pump would deliver 225 m^3/h (1000 gpm) at 610 m (2000 ft) of head. This is basically a series of impellers operating within one casing. Separate multistage pumps can be put in series where it is required for ultimate high pressure or where individual pumps can take care of part of a process. When pumps are operating in series the rate of flow remains the same and the resultant pressure is additive.

1.3.4.5 Enclosed

This impeller type (see Figure 1.3.4.5) is used for the majority of pump types and for most applications, both single and double suction. The shrouds are attached to the vanes and fully enclose the flow passage. The higher pressure discharge flow is separated from the lower pressure inlet flow by close running clearance, referred to as *wear rings*, that reduce internal leakage and help to maintain a high level of pump volumetric efficiency. After extensive operation and wear, pump efficiency can normally be restored to original levels by renewal of the original clearances at the impeller inlet wear rings to the adjacent casing or casing wear ring. This type of impeller works best with clear liquids. For slurry type pumps, see ANSI/HI 12.1–12.6 *Rotodynamic (Centrifugal) Slurry Pumps for Nomenclature, Definitions, Applications, and Operations.*

Figure 1.3.4.5 — Enclosed impeller

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1.3.4.6 Semi-open

This impeller type (see Figure 1.3.4.6) is applicable only to single suction pump stages and has the vanes enclosed with a full or partial shroud on one side only. The performance and efficiency of the enclosed and semi-open impellers are similar. However, the semi-open impeller requires a tight clearance be maintained between the open face and its mating stationary surface. This clearance should be between 0.25 and 0.38 mm (0.010 and 0.015 in). The rotating element should be axially adjustable in the pump to control this clearance. Semi-open impellers are used for fibrous or potentially clogging material in the pumped liquid. These types of materials block (or clog) the passages, resulting in reduced flow and/or excessive vibration of the pump. The design can be either open front or back. Both accomplish the function of easy cleaning. Semi-open impellers are also more effective in handling small quantities of air.

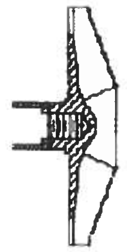

Figure 1.3.4.6 — Semi-open impeller

A variation of this design is used in the vortex pump. This uses a semi-open impeller typically with a clearance width up to one volute width between the impeller and the mating casing face. It creates a forced vortex motion in the casing. The increased clearances limit the head that can be generated and reduce the attainable efficiency. This style impeller is common when pumping large solids.

1.3.4.7 Open

This type of impeller (see Figure 1.3.4.7) has no front or back shroud with vanes running in close proximity to mating casing walls or liners. This creates very low axial forces. The impeller must be designed structurally to support the forces generated in the action of pumping the fluid as well as withstand the stresses generated by centrifugal forces at operating speed. Open impellers are used for high-speed pumps of over 10,000 rpm. They are usually small in diameter and have the material and blade shape to support the developed pressure. As with the semi-open impeller, axial adjustment should be possible.

Figure 1.3.4.7 — Open impeller

1.3.4.8 Special suction impeller (inducer)

Pumps operating at higher speeds have led to the development of special suction devices referred to as *inducers*. Inducers are single-stage axial flow helixes installed in the suction eye of centrifugal pump impellers to lower the NPSHR of the pump (see Figure 1.3.4.8). This allows use of increased rotating speed for a given NPSHA or a lower NPSHR for a given speed. Shallow blade inlet angles are used to draw liquid into the inducer channels, which are shaped to impart enough energy to provide sufficient NPSHA for the main impellers to avoid detrimental cavitation.

Whereas centrifugal pump impellers often have a suction specific speed, S, of about 155 to 230 (N_{SS} = 8000 to 12,000), use of inducers can increase S to a range of 290 to 680 (N_{SS} = 15,000 to 35,000). This typically allows lower-flow pumps to operate at 11,000 rpm with the same NPSHA, which would be required at 3600 rpm without inducer, or reduce NPSHR of all pumps to less than half at the same speed. See units of measurement (Section 1.3.2.2).

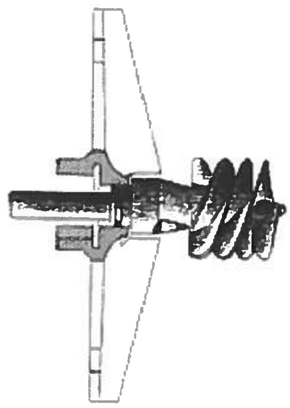

Figure 1.3.4.8 — Inducer

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Cavitation damage of inducer blades may occur above certain (experience-established) inducer tip speeds. At lower tip speeds, cavitation will not produce damage regardless of *S* value. Allowable tip speed is also a function of inducer materials; e.g., for titanium alloy it is 20% higher than for stainless steel. Tip speeds may also be increased for lower specific gravity liquids. Optimum design is frequently a compromise between the cavitation and performance characteristics of the inducer.

Inducers must be designed and selected to provide sufficient *S* (N_{ss}) over the intended operating range to prevent the occurrence of cavitation within the main impeller. Suction performance typically is reduced at low and high ends of inducer performance range.

1.3.5 Mechanical features

1.3.5.1 Calculation of radial thrust for volute pumps

Single and dual volute pumps are typically designed for uniform mean velocity around the volute of the casing at the design rate of flow (BEP). For pumps in operation near or at BEP, the radial thrust may approach zero. For operation at flow rates higher or lower than BEP, the pressure distribution is not uniform, resulting in a radial thrust on the impeller. The magnitude and direction of the radial thrust changes with change in flow rate. The following expression is used to calculate radial thrust:

Metric units

$$F_R = K_R \times H \times \rho \times g \times D_2 \times b_2$$

US customary units

$$F_R = K_R \times \left(\frac{H \times s}{2.31}\right) \times D_2 \times b_2$$

Where:

F_R = radial thrust, in N (lbf)

K_R = thrust factor, which varies with rate of flow and specific speed. See Figure 1.3.5.1.d for single volute and Figure 1.3.5.1.f for double volute.

H = developed head per stage, in m (ft)

ρ = density of pumped liquid, in kg/m^3

s = specific gravity of pumped liquid

D_2 = impeller diameter, in m (in)

b_2 = impeller width at discharge including shroud(s), in m (in)

g = 9.81 m/s^2 – gravitational constant

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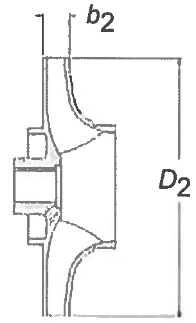

Figure 1.3.5.1a — Additional sketch to illustrate D_2 and b_2

For single volute pumps, the highest value of radial thrust is typically at zero flow in a direction from the shaft centerline towards the narrower part of the volute casing. As the rate of flow increases, the load loses magnitude until it reaches its minimum at or near the BEP. For pumps of low specific speed design, n_s = 10 (N_s = 500), the minimum radial thrust value may be observed at about 80% of the BEP rate of flow. The magnitude of the force then increases at rates of flow over BEP and its direction shifts towards the wider volute area. Exact direction of radial force at various rates of flow is highly dependent on the pump specific speed and design parameters of the volute. Zero value of radial thrust is not often realized. Figure 1.3.5.1b shows approximate boundaries of radial thrust direction for single volute pumps.

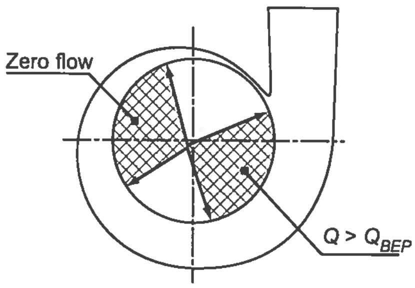

Figure 1.3.5.1b — Single volute radial thrust distribution

Dual (double) volute is when two volutes enclose the impeller. The use of a dual volute casing reduces the radial load to 0.1 to 0.27 times the load of a single volute (see Figures 1.3.5.1d and 1.3.5.1f).

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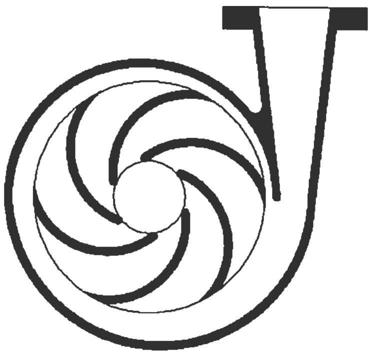

Figure 1.3.5.1c — Single volute

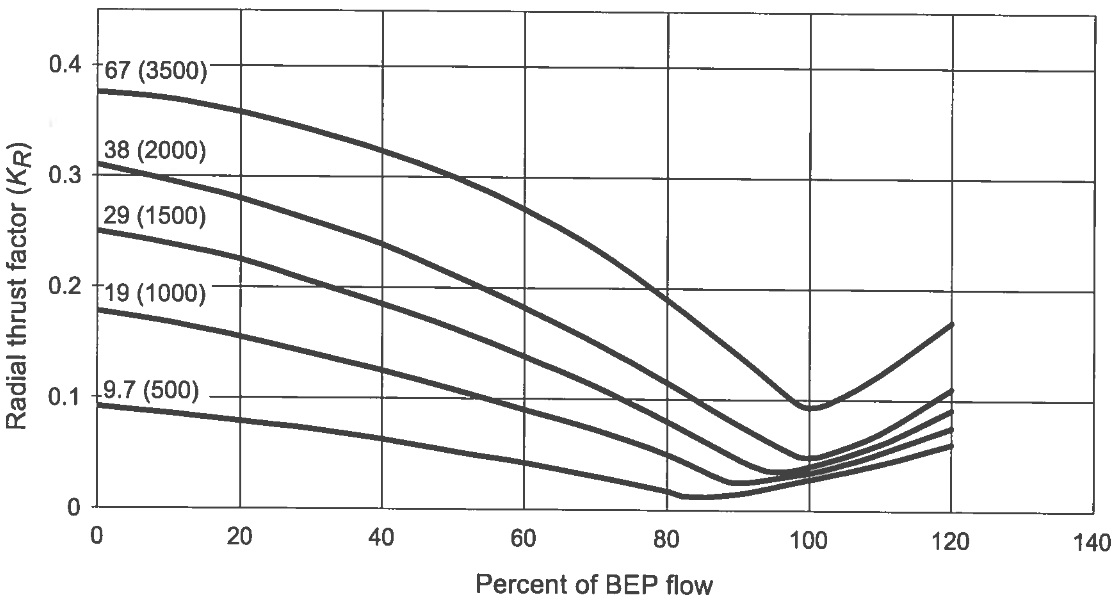

Figure 1.3.5.1d — Radial thrust factor for single volute with various specific speeds

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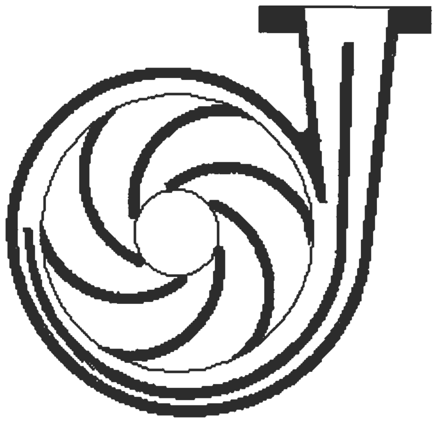

Figure 1.3.5.1e — Double volute

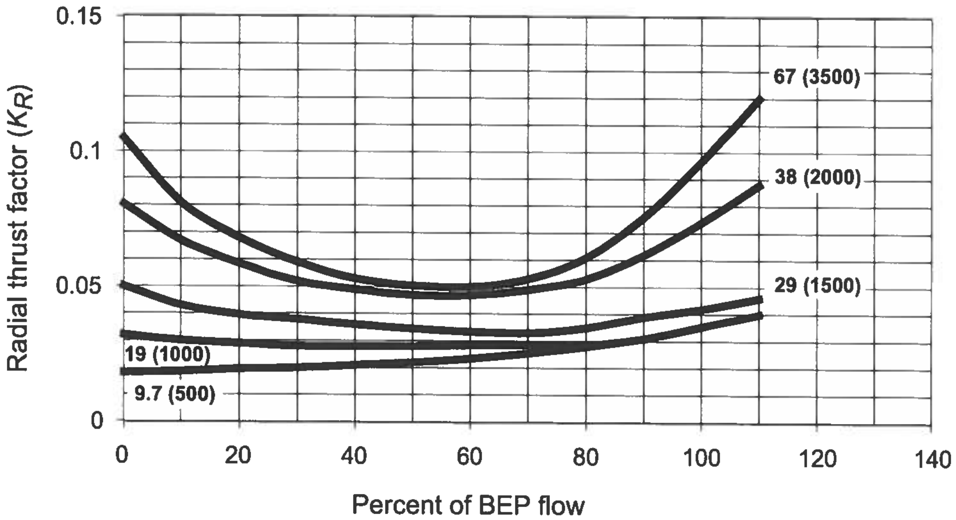

Figure 1.3.5.1f — Radial thrust factor for double volute with various specific speeds

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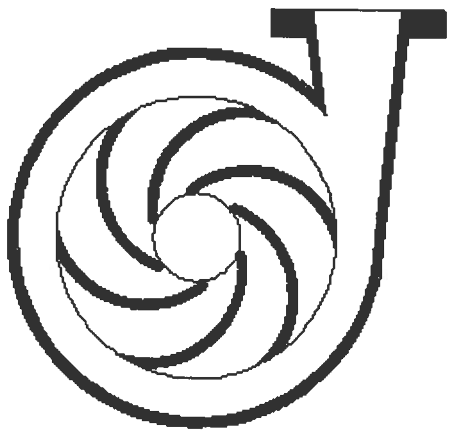

Figure 1.3.5.1g — Circular (concentric) casing

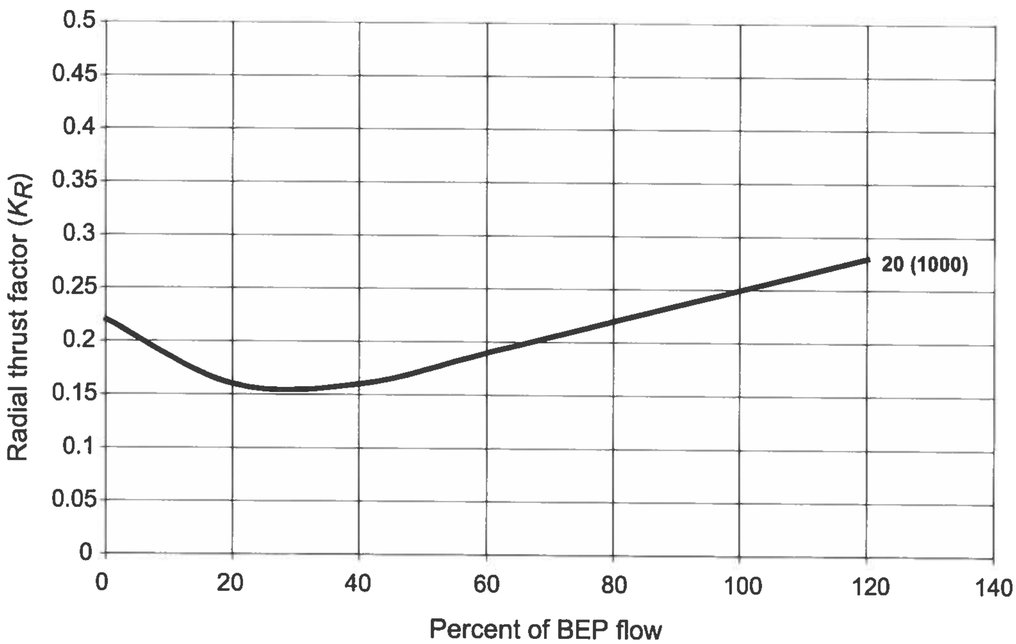

Figure 1.3.5.1h — Radial thrust factor for circular (concentric) casing typical for specific speed 20 (1000)

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Radial thrust changes with impeller diameter ratio to the third power because, at a constant speed of rotation, thrust is proportional to *H*×*D* and the head changes as the diameter squared. This is approximate because b_2 and n_s also change as the diameter is changed:

$$F_{R2} \approx F_{R1}\left(\frac{D_2}{D_1}\right)^3$$

NOTE: When only the impeller vane diameter is changed while the shroud remains the same (see Figure 1.3.6.1.4c), the new vane diameter but not the shroud diameter should be considered for radial thrust calculation.

Radial thrust changes with change of speed of the pump to the second power of the speed ratio because head changes as the square of the speed (n):

$$F_{R2} = F_{R1}\left(\frac{n_2}{n_1}\right)^2$$

Radial thrust values for circular (concentric) casings also can be predicted, but with an increased level of uncertainty due to their more arbitrary design. As a general guide, circular casings are used on pump designs with specific speeds of 20 (1000) or less (see Figure 1.3.5.1h). The radial force is dependent on the geometry of the casing cross section. K_R is similar to values for volute casings based on Agostinelli et al, *An Experimental Investigation of Radial Thrust in Centrifugal Pumps*, ASME Journal of Engineering, April 1960. Basically the K_R value increases in magnitude with increasing flow rate within the allowable operating region (K_R is lower at minimum flow and highest at BEP flow or greater).

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1.3.5.2 Calculation of axial thrust for impellers

In a horizontal pump the axial thrust is a net force acting through the rotor on the thrust bearing, including dynamic loading from pressure and momentum acting on impeller and other rotor components.

The net axial thrust is the sum of all effects. It is important that the appropriate equations for the effects of pressure on axial thrust, as shown in Sections 1.3.5.2.1 or 1.3.5.2.2, are used in conjunction with the equations for the effects of fluid momentum change on axial thrust as shown in Section 1.3.5.2.3.

1.3.5.2.1 Axial thrust on enclosed impellers

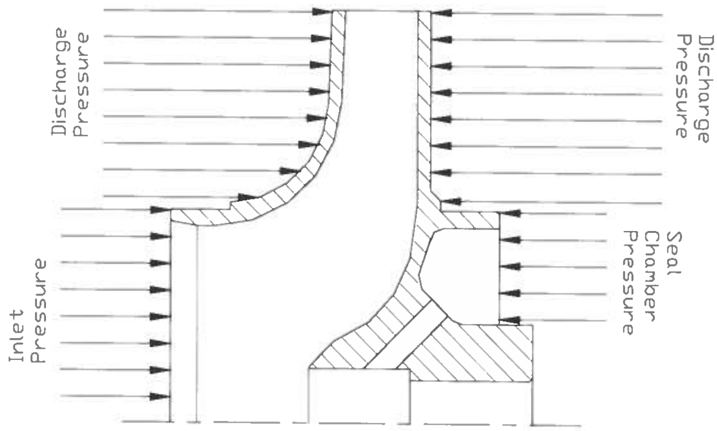

Figure 1.3.5.2.1a — Pressure distribution on enclosed impeller

The forces creating thrust on an enclosed impeller (Figure 1.3.5.2.1a) are due to the difference in pressure distribution on the front and back shrouds along with a force from the momentum due to change of direction of the flow through the impeller. Following is a standard calculation method for a single-stage casing to determine the axial thrust force that is developed and acts on the impeller in a direction parallel to the shaft. This specification is for single-stage single suction pumps only (specific speed range of 10 to 67 [500 to 3500]) with enclosed impellers having no back vanes with a plain horizontal ring(s) and with a diametral clearance of 0.25 to 0.5 mm (0.010 to 0.020 in). The complexity of the interaction of interstage pressures, casting tolerances, and machining tolerances does not make it practical to have an overall method of axial thrust calculation for multistage pumps.

For pumps in a vertical position, the weight of impeller and shaft should be added to the axial thrust. Also, depending on the construction, the coupling weight, the driver motor, or balancing device weight may also be added to the total axial thrust.

The values of axial thrust are for 25% to 125% of BEP rate of flow. Within this range, maximum axial thrust will be developed and can be determined by the following equation:

Metric units

$$F_A = (Hg\,\rho)[(\overline{K_B}A_B) - (\overline{K_F}A_F)] - 1000\,p_sA_h + 70{,}000\,A_{BAL}$$ [1]

US customary units

$$F_A = \left(\frac{Hs}{2.31}\right)[(\overline{K_B}A_B) - (\overline{K_F}A_F)] - p_sA_h + 10\,A_{BAL}$$ [1]

[1] The average pressure behind the back ring is 70 kPa (10 psi) greater than suction pressure when the area of the balance holes equal three times the area of the clearance of the back ring. For impellers with no back ring, this term is 0.

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Where:

F_A = net axial thrust, in N (lbf)

K_A = factor of developed head at some location on the impeller shroud (Figure 1.3.5.2.1b)

$\overline{K_B}$ = average factor of developed head on the back shroud

$\overline{K_F}$ = average factor of developed head on the front shroud

A_B = area exposed to pressure on the back shroud, in m^2 (in^2)

A_F = area exposed to pressure on the front shroud, in m^2 (in^2)

A_{BAL} = area between the back ring and impeller hub, in m^2 (in^2)

p_s = suction pressure, in kPa (psi)

A_h = area of shaft, shaft sleeve, or mid-point of mechanical seal rotating face exposed to atmosphere, in m^2 (in^2)

H = head per stage, in m (ft)

g = 9.81 m/s^2, gravitational constant

s = specific gravity

ρ = density, in kg/m^3

The axial thrust factor given in the basic chart is derived from an assumption of a free vortex between the casing wall and impeller shroud. The liquid velocity is to be one half the peripheral velocity of the impeller. The values have been adjusted using empirical data.

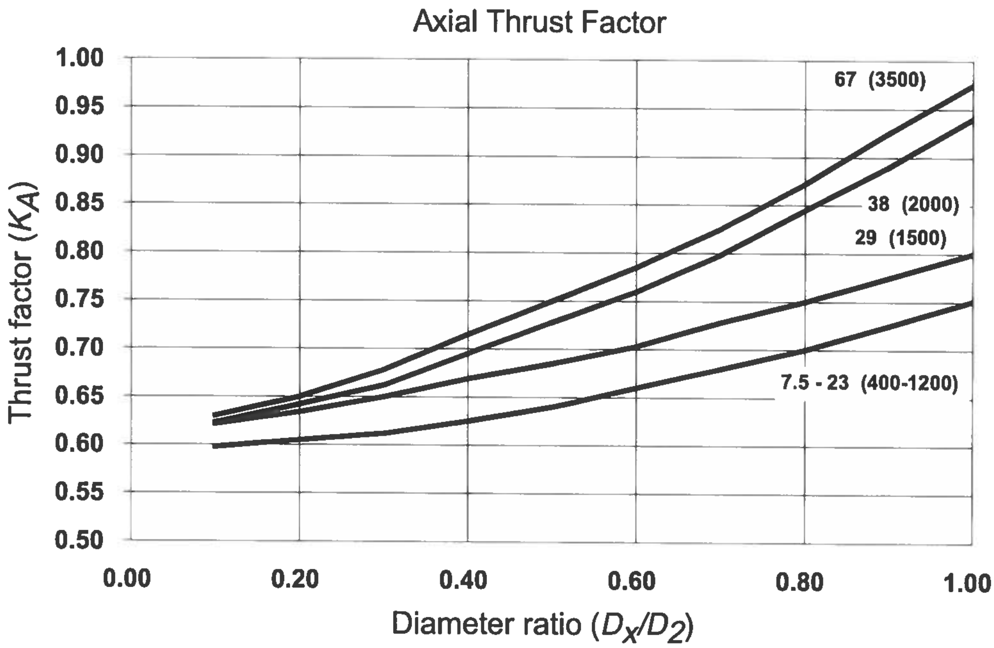

Figure 1.3.5.2.1b — Pressure distribution on enclosed impeller shrouds for various specific speeds

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The typical ratio of the distance from the casing wall to the impeller shroud to maximum impeller diameter is as follows for various specific speeds:

n_s (N_s)	$\frac{\text{Casing wall to shroud}}{\text{Maximum impeller diameter}}$
7.5 – 23 (400 – 1200)	0.02 to 0.04
29 (1500)	0.03 to 0.05
38 – 67 (2000 – 3500)	0.04 to 0.06

The values of K_A will increase or decrease as these ratios change.

K_A is based on 0.25- to 0.50-mm (0.010- to 0.020-in) diametrical ring clearance and is highly influenced by ring clearance. For further discussion of the subject, and review of other methods of controlling axial thrust balance (such as balance chambers and pump out vanes), refer to *Centrifugal Pumps Design and Application* by Lobanoff and Ross; *Centrifugal and Axial Pumps* by A.J. Stepanoff; and *Pump Handbook* by Karassik, Messina, Cooper, and Heald (see Appendix D).

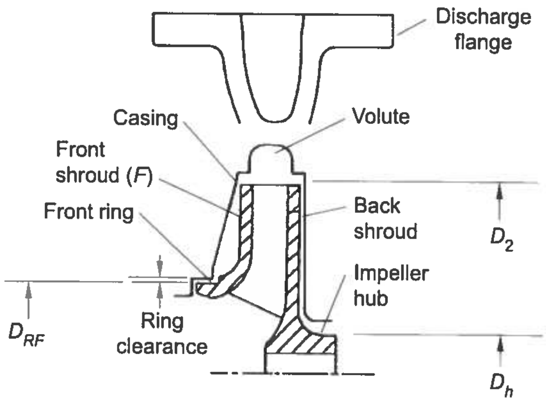

Figure 1.3.5.2.1c — Thrust calculation example for enclosed impeller with plain back shroud

Thrust calculation example for enclosed impeller with plain back shroud (Figure 1.3.5.2.1c) in metric units

Where:

Q = 145 m^3/h

H_{BEP} = 100 m

H_{MAX} = 114 m

p_s = 0 kPa

N = 3550 rpm

D_2 = 0.240 m

D_{RF} = 0.121 m

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D_h = 0.038 m

ρ = 1000 kg/m^3

g = 9.81 m/s^2

$$n_s = \frac{3550\sqrt{\frac{145}{3600}}}{100^{0.75}} = 22.5$$ From Figure 1.3.5.2.1b, $K_{A_2} = 0.75$

$$\frac{D_{RF}}{D_2} = \frac{0.121}{0.240} = 0.504$$ From Figure 1.3.5.2.1b, $K_{A_{RF}} = 0.64$

$$\overline{K_F} = \frac{K_{A_2} + K_{A_{RF}}}{2} = \frac{0.75 + 0.64}{2} = 0.70$$

$$\frac{D_h}{D_2} = \frac{0.038}{0.240} = 0.158$$ From Figure 1.3.5.2.1b, $K_{A_h} = 0.60$

$$\overline{K_B} = \frac{K_{A_2} + K_{A_h}}{2} = \frac{0.75 + 0.60}{2} = 0.68$$

$$A_B = \frac{\pi}{4}(0.240^2 - 0.038^2) = 0.044 \text{ m}^2$$

$$A_F = \frac{\pi}{4}(0.240^2 - 0.121^2) = 0.034 \text{ m}^2$$

$$A_h = \frac{\pi}{4} \times 0.038^2 = 0.001 \text{ m}^2$$

$$F_A = (H_{MAX} \times g \times \rho) \times [(\overline{K_B}A_B) - (\overline{K_F}A_F)] - (1000\ p_s A_h)$$

$$F_A = (114 \times 9.81 \times 1000) \times [(0.68 \times 0.044) - (0.70 \times 0.034)] - (1000 \times 0 \times 0.001)$$

F_A = 6800 N towards suction

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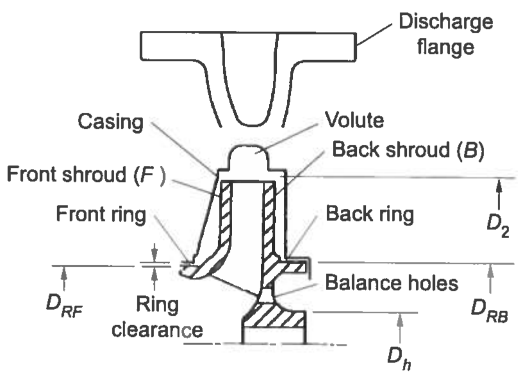

Figure 1.3.5.2.1d — Impeller with back ring

Thrust calculation for impeller with back ring (see Figure 1.3.5.2.1d) in metric units

Where:

Q = 145 m^3/h

H_{BEP} = 100 m

H_{MAX} = 114 m

p_s = 345 kPa

N = 3550 rpm

D_2 = 0.240 m

D_{RF} = 0.121 m

D_{RB} = 0.095 m

D_h = 0.038 m

ρ = 600 kg/m^3

$$n_s = \frac{3550\sqrt{\frac{145}{3600}}}{100^{0.75}} = 22.5$$ From Figure 1.3.5.2.1b, $K_{A_2} = 0.75$

$$\frac{D_{RF}}{D_2} = \frac{0.121}{0.24} = 0.504$$ From Figure 1.3.5.2.1b, $K_{A_{RF}} = 0.64$

$$\overline{K_F} = \frac{K_{A_2} + K_{A_{RF}}}{2} = \frac{0.75 + 0.64}{2} = 0.70$$

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$$\frac{D_{RB}}{D_2} = \frac{0.095}{0.240} = 0.396 \quad \text{From Figure 1.3.5.2.1b, } K_{A_{RB}} = 0.62$$

$$\overline{K_B} = \frac{K_{A_2} + K_{A_{RB}}}{2} = \frac{0.75 + 0.62}{2} = 0.69$$

$$A_B = \frac{\pi}{4}(0.240^2 - 0.095^2) = 0.038 \text{ m}^2$$

$$A_{BAL} = \frac{\pi}{4}(0.095^2 - 0.038^2) = 0.006 \text{ m}^2$$

$$A_F = \frac{\pi}{4}(0.240^2 - 0.121^2) = 0.034 \text{ m}^2$$

$$A_h = \frac{\pi}{4} \times 0.038^2 = 0.001 \text{ m}^2$$

$$F_A = (H_{MAX} \times g \times \rho) \times [(\overline{K_B} A_B) - (\overline{K_F} \times A_F)] - 1000\ p_s A_h + 70{,}000\ A_{BAL}$$ [1]

$$F_A = (114 \times 9.81 \times 600) \times [(0.69 \times 0.038) - (0.70 \times 0.034)] - (1000 \times 345 \times 0.001) + 70{,}000 \times 0.006$$ [1]

F_A = 1699 *N* towards suction

Thrust calculation for enclosed impeller with plain back shroud (see Figure 1.3.5.2.1c) in US customary units

Where:

Q	=	640 gpm
H_{BEP}	=	325 ft
H_{MAX}	=	375 ft
p_s	=	0 psi
N	=	3550 rpm
D_2	=	9.44 in
D_{RF}	=	4.75 in
D_h	=	1.5 in
s	=	1.0

[1] The average pressure behind the back ring is 70 kPa greater than suction pressure when the area of the balance holes equals three times the area of the clearance of the back ring.

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$$N_s = \frac{3550(\sqrt{640})}{325^{0.75}} = 1170 \quad \text{From Figure 1.3.5.2.1b, } K_{A_2} = 0.75$$

$$\frac{D_{RF}}{D_2} = \frac{4.75}{9.44} = 0.503 \quad \text{From Figure 1.3.5.2.1b, } K_{A_{RF}} = 0.64$$

$$\overline{K_F} = \frac{K_{A_2} + K_{A_{RF}}}{2} = \frac{0.75 + 0.64}{2} = 0.70$$

$$\frac{D_h}{D_2} = \frac{1.5}{9.44} = 0.16 \quad \text{From Figure 1.3.5.2.1b, } K_{A_H} = 0.60$$

$$\overline{K_B} = \frac{K_{A_2} + K_{A_H}}{2} = \frac{0.75 + 0.60}{2} = 0.68$$

$$A_B = \frac{\pi}{4}(9.44^2 - 1.5^2) = 68.2 \text{ in}^2$$

$$A_F = \frac{\pi}{4}(9.44^2 - 4.75^2) = 52.3 \text{ in}^2$$

$$A_h = \frac{\pi}{4} \times 1.5^2 = 1.8 \text{ in}^2$$

$$F_A = \left(\frac{H_{MAX} \times s}{2.31}\right) \times [(\overline{K_B}A_B) - (\overline{K_F}A_F)] - (p_s A_h)$$

$$F_A = \left(\frac{375 \times 1.0}{2.31}\right) \times [(0.68 \times 68.2) - (0.70 \times 52.3)] - (0 \times 1.8)$$

F_A = 1600 lbf toward suction

Thrust calculation for impeller with back ring (see Figure 1.3.5.2.1d) in US customary units

Where:

Q = 640 gpm

H_{BEP} = 325 ft

H_{MAX} = 375 ft

p_s = 50 psi

N = 3550 rpm

D_2 = 9.44 in

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D_{RF} = 4.75 in

D_{RB} = 3.75 in

D_h = 1.5 in

s = 0.6

$$N_s = \frac{3550(\sqrt{640})}{325^{0.75}} = 1170 \quad \text{From Figure 1.3.5.2.1b, } K_{A_2} = 0.75$$

$$\frac{D_{RF}}{D_2} = \frac{4.75}{9.44} = 0.503 \quad \text{From Figure 1.3.5.2.1b, } K_{A_{RF}} = 0.64$$

$$\overline{K_F} = \frac{K_{A_2} + K_{A_{RF}}}{2} = \frac{0.75 + 0.63}{2} = 0.70$$

$$\frac{D_{RB}}{D_2} = \frac{3.75}{9.44} = 0.397 \quad \text{From Figure 1.3.5.2.1b, } K_{A_{RB}} = 0.62$$

$$\overline{K_B} = \frac{K_{A_2} + K_{A_{RB}}}{2} = \frac{0.75 + 0.62}{2} = 0.69$$

$$A_2 = \frac{\pi}{4}(9.44^2) = 70.0 \text{ in}^2$$

$$A_B = \frac{\pi}{4}(9.44^2 - 3.75^2) = 58.9 \text{ in}^2$$

$$A_{B1} = \frac{\pi}{4}(3.75^2 - 1.5^2) = 9.28 \text{ in}^2$$

$$A_F = \frac{\pi}{4}(9.44^2 - 4.75^2) = 52.3 \text{ in}^2$$

$$A_h = \frac{\pi}{4} \times 1.5^2 = 1.8 \text{ in}^2$$

$$F_A = \left(\frac{H_{MAX} \times s}{2.31}\right) \times [(\overline{K_B} \times A_B) - (\overline{K_F} \times A_F)] - A_h p_s + (10 A_{B1})^1$$

$$F_A = \left(\frac{375 \times 0.6}{2.31}\right) \times [(0.69 \times 58.9) - (0.70 \times 52.3)] - (1.8 \times 50) + (10 \times 9.28)^1$$

F_A = 395 lbf towards suction

---

[1] The average pressure behind the back ring is 10 psi greater than suction pressure when the area of the balance holes equals three times the area of the clearance of the back ring.

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1.3.5.2.2 Calculation of axial thrust for semi-open impellers

The semi-open impeller (Figure 1.3.5.2.2a) has only one shroud. The difference in pressure distributions along both the outer side and vane side of the shroud is typically greater than between front and back shrouds of an enclosed impeller. Axial thrust on a semi-open impeller is higher than that of an enclosed impeller. This can result in short bearing life or need of an extra-large thrust bearing.

Because most semi-open impellers employ an open front design, this discussion will be about that type of design only.

To reduce the imbalance, the back shroud area may be decreased by removing material. This is called *scalloping*. The amount of scalloping will reduce the thrust accordingly. The scalloping also causes some internal recirculation within the impeller passages. This will result in a reduction in overall efficiency; it can also result in a gain of developed head from the same recirculation.

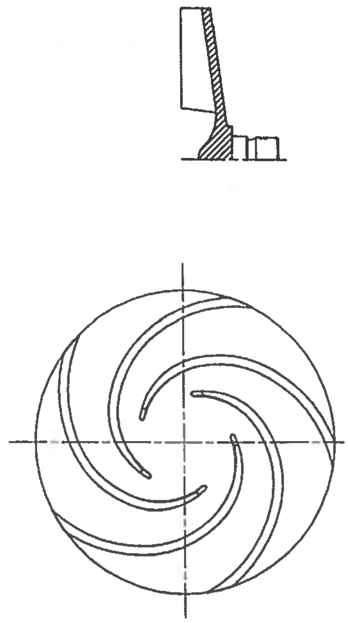

Figure 1.3.5.2.2a — Semi-open impeller with a full back shroud

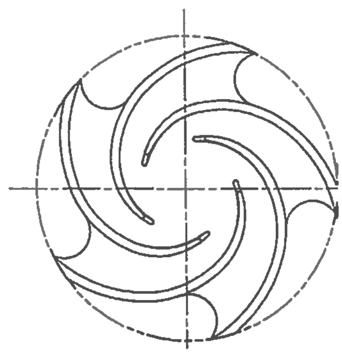

Figure 1.3.5.2.2b — Semi-open impeller with a scalloped back shroud

The axial thrust is calculated as follows:

Thrust toward the pump suction

Metric units

$$F_A = (Hgp)\left\{\left(\frac{K_{A2}+K_h}{2}\right)(A_2 - A_h) - \left(\frac{K_2}{2}\right)(A_2 - A_1)\right\}$$

US customary units

$$F_A = \left(\frac{Hs}{2.31}\right)\left\{\left(\frac{K_{A2}+K_h}{2}\right)(A_2 - A_h) - \left(\frac{K_2}{2}\right)(A_2 - A_1)\right\}$$

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Where:

D_1 = eye diameter of the impeller, in mm (in)

A_1 = eye area of the impeller, in mm^2 (in^2)

D_2 = outside diameter of the impeller for full shrouds, in mm (in)

A_2 = area covered by the full diameter of the impeller, in mm^2 (in^2)

K_2 = pressure gradient factor for the hub shroud (Refer to OEM.)

A_h = area enclosed by the shaft sleeve when packing is used or at the mean face of the mechanical seal, in mm^2 (in^2)

K_h = pressure gradient factor

H = developed head of the pump, in m (ft)

s = specific gravity

g = gravitational constant, in m/s^2

K_{A2} = per Figure 1.3.5.2.1b for the appropriate specific speed

K_1 = zero for open or semi-open impellers

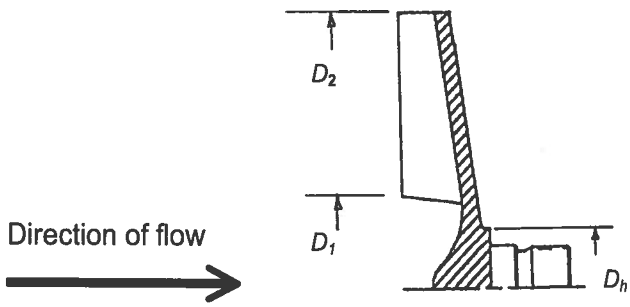

Figure 1.3.5.2.2c — Cross section of semi-open impeller showing diameter locations

K_h changes with the shroud configuration. K_h factors apply when the shroud or pump-out vanes are within 0.25 to 0.40 mm (0.010 to 0.015 in) running clearance to the mating face (usually the casing cover or casing wall). Excessive clearance can result in higher and additional thrust loads.

For a full shroud with no pump-out vanes, K_h = 0.6.

For a plain shroud that has 25% of the shroud area scalloped or removed (average value), K_h = 0.4.

For a full shroud with full diameter pump-out vanes, K_h = 0.25.

For a shroud that is scalloped 25% and has pump-out vanes, K_h = 0.15.

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When the shroud is scalloped the value of A_2 becomes A_{De}. The diameter of A_{De} is located at De, which depends on the amount of area removal or scalloping. This should be the diameter used to determine K_2.

$$De = \left(\frac{Full\ shroud\ area - Scalloped\ area}{0.785}\right)^{0.5}$$

Similar to the enclosed impeller, a reversal of axial thrust has to be accounted for if there is significant suction pressure.

Thrust from suction pressure = $T_s = A_h \times$ suction pressure.

1.3.5.2.3 Calculation of axial thrust for axial flow pump impellers

The axial flow pump impeller (propeller) has no front or back 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.

1.3.5.2.4 Calculation of axial force due to momentum change

In single suction impellers the axial force related to the momentum of the incoming fluid should be considered, particularly at operating points beyond BEP. This only applies to single suction impellers with radial or mixed flow discharges. The incoming liquid has mass and is traveling at a relatively high velocity. The momentum change as the flow changes direction in the impeller creates an axial force that should be taken into account in overall axial thrust calculation.

Metric units

$$F_A = \frac{98 \times s \times Q^2}{{D_1}^2}$$

US customary units

$$F_A = \frac{s \times Q^2}{567 \times {D_1}^2}$$

Where:

F_A = axial thrust, in N (lbf)

s = specific gravity

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

D_1 = impeller eye diameter, in mm (in)

1.3.5.3 Types of bearing arrangements

The bearing arrangement in a rotodynamic pump is normally one of two types:

a) Impeller overhung from bearings (Type OH).

b) Impeller mounted between bearings (Type BB).

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Close-coupled and frame-mounted single suction pumps are common examples of an overhung bearing load. Single-stage double suction and multistage horizontal shaft pumps are common examples of the "between-bearing impeller" load. Both arrangements can provide acceptable bearing life.

1.3.5.3.1 Impeller overhung from bearings (Type OH)

Pumps of simplest design have single-stage suction impellers and use an overhung shaft supported by two bearings (see Figure 1.3.5.3.1). The bearings are mounted in a cast housing. The inboard bearing, or bearing closest to the impeller, is usually a pure radial bearing. The outboard bearing, however, absorbs all the axial loads and some radial load. The radial load on the inboard bearing is approximately twice that being generated by the impeller. The radial load on the outboard bearing is approximately that produced by the impeller. As the required developed pressure and flow for the pumps increase, it can become economically impractical to remain with a single-stage, end-suction impeller design. (The exceptions are slurry pumps handling large solid particles and sludge.) The options are to utilize single-stage double suction pumps for high flows and multistage pumps for high developed pressure.

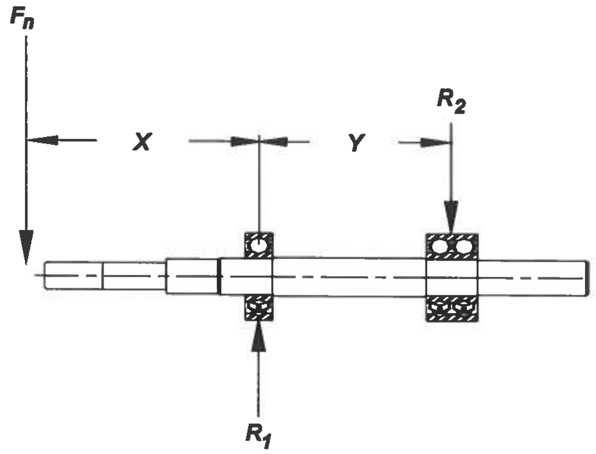

Figure 1.3.5.3.1 — Overhung impeller pump

$$R_1 = \frac{F_n \times (X + Y)}{Y}$$

$$R_2 = \frac{F_n \times X}{Y}$$

Where:

R_1 = inboard bearing reaction load

R_2 = outboard bearing reaction load

F_n = F_R + W, where:

- F_n = net force typically applied at the impeller centerline
- F_R = radial hydraulic thrust applied at impeller centerline
- W = impeller weight

Assume F_R + W are acting in the same direction for maximum load.

X = distance from applied load to the centerline of the inboard bearing

Y = distance between inboard and outboard bearing

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1.3.5.3.2 Impeller mounted between bearings (Type BB)

This arrangement (see Figure 1.3.5.3.2) can be applied to single suction, double suction, and multiple impeller configurations. Typically it can be found on double suction and multiple stage pumps.

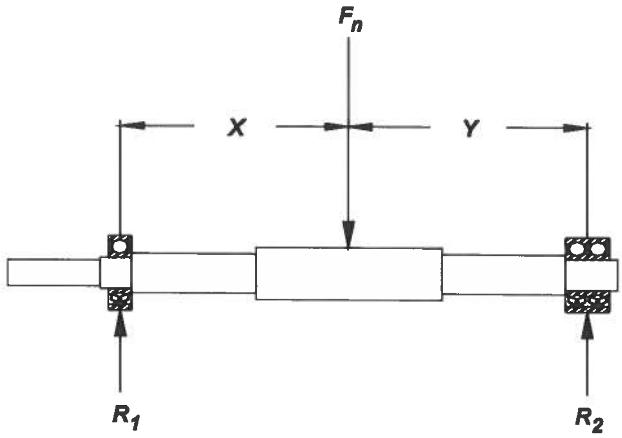

Figure 1.3.5.3.2 — Impeller between bearings

$$R_1 = \frac{F_n \times Y}{X + Y}$$

$$R_2 = \frac{F_n \times X}{X + Y}$$

Where:

R_1 = inboard bearing reaction load

R_2 = outboard bearing reaction load

F_n = F_R + W, where:

F_n = net force typically applied at the impeller centerline

F_R = radial hydraulic thrust applied at impeller centerline

W = impeller weight

Assume F_R + W are acting in the same direction for maximum load.

X = distance from applied load to the centerline of the inboard bearing

Y = distance from applied load to the centerline of the outboard bearing

1.3.5.3.2.1 Impellers between bearings, double suction pumps

Because a double suction impeller has two opposed inlets, it is typical to use a shaft with bearings at each end (see Figure 1.3.5.3.2.1). In theory, because the impeller is symmetrical, the axial loads should be negligible; however, in practice the piping and flow conditions can create an imbalanced pressure distribution. Recognizing this possibility the outboard bearing selection may include axial thrust capability.

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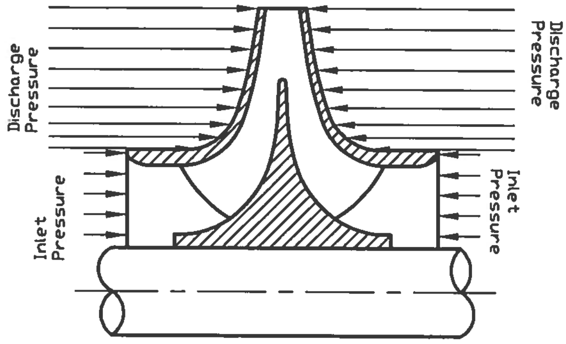

Figure 1.3.5.3.2.1 — Pressure distribution on between-bearings double suction enclosed impeller

1.3.5.3.2.2 Impellers between bearings, multistage pumps

With the increase of required developed pressure, there is an increase in impeller diameter. For a given flow it may become impractical to cast impellers with the required impeller-width-to-diameter ratio. Also the efficiency is drastically reduced from the hydraulic friction of the large-diameter impeller shrouds. To obtain good castings and maintain high hydraulic efficiency, pumps use multiple impellers on the shaft. These are called *multistage pumps*. These designs also require bearings on both ends of the shaft.

Radial load: If the volutes for the multistage were not aligned to oppose the radial load of an adjacent stage, there would be large radial loads on the bearings. Therefore, the adjacent stage's volute cutwater is placed at 180 degrees or the casings are cast as dual volutes for each stage to allow a counteraction and reduction of the radial thrust. Another method of eliminating radial thrust is to use diffusers around each impeller.

Axial load: When all the stages are mounted on the shaft so the suction of all the stages face the same direction, the thrust of each stage is additive and results in a large axial thrust in one direction. To counteract this total thrust, a balancing drum with a return line back to the pump suction is used.

Another method is to mount half of the impeller stages facing one direction on the shaft and the other half in the opposite direction to balance out each other. With either method there is usually a residual axial thrust that has to be absorbed by the bearings. The magnitude of the thrust depends on what portion of the head versus rate of flow (HQ) curve the pump is operating.

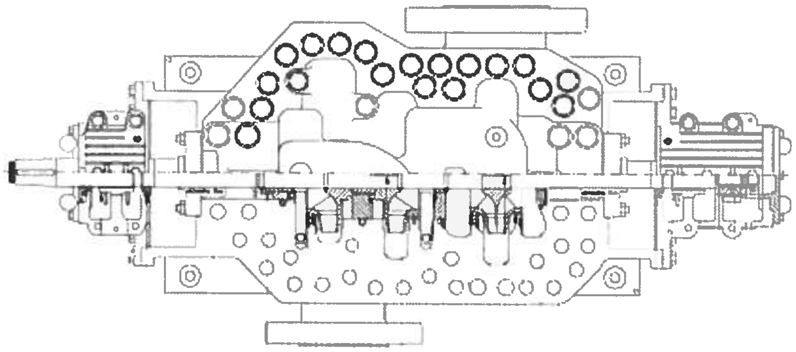

Figure 1.3.5.3.2.2 — Horizontal between-bearings axially split multistage pump

Shaft support: In simple beam theory, shaft deflection at the center of the bearing span is often greater than the available internal bushing clearances. Additionally, using basic calculation methods the first dry critical speed of the shaft system will be calculated to be within the operating range. However, during operation (and in all appropriate rotordynamic analysis models), the liquid film existing in the close running clearance of internal wear part labyrinths at each stage is known to produce a dynamic bearing stiffness at each of these locations. This provides internal support of the system and essentially eliminates radial hydraulic loads from the external bearings. The selection of the external bearings can therefore be based on the weight of the rotating element and the residual axial thrust.

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1.3.5.4 Shaft deflection

1.3.5.4.1 Description

Shaft deflection is a design criterion that greatly influences pump performance due to its effect on the mechanical seal, internal clearances, and bearings.

Radial loads acting on the rotating impeller(s) are transmitted directly to the pump shaft. This force will deflect the shaft where it is applied, irrespective of the bearing configuration. The direction of the hydraulic radial load, at a given operating point, remains constant with respect to the pump casing. However, it is seen as cyclic stress reversal with respect to the rotating shaft and has a dynamic effect on the mechanical seal. The shaft must be designed to accommodate this hydraulic radial load in conjunction with the additional radial load imposed due to the mass of the impeller(s) and other rotating components. Under these conditions the rotor must be stiff enough to limit the resulting deflection to within limits of the internal clearances and mechanical seal requirements. Adequate clearance must be verified at critical locations by using the shaft deflection equations.

Dynamic deflection of the pump shaft changes the relative location of the mechanical seal faces and thus has a large impact on the overall seal life. For a mechanical seal to reach its design life, a number of requirements have to be met. From a static or dimensional point of view, the relative locations of the primary, stationary, and rotating faces must be held within control limits. These limits can be met by using a combination of flexibility within the seal assembly and by using appropriate manufacturing procedures and processes.

Limiting the shaft deflection will also improve packing life in arrangements where a packed stuffing box is used as the method of shaft sealing.

Shaft deflection causes angular misalignment of the shaft at the bearings, which may affect bearing wear and life depending on degree of deflection, bearing type, and bearing retainer design.

External loads such as coupling misalignment should also be considered.

1.3.5.4.2 Typical industry standards

1.3.5.4.2.1 Overhung impeller pumps, ASME B73.1

In accordance with ASME B73.1, dynamic shaft deflection at the impeller centerline shall not exceed 0.125 mm (0.005 in) anywhere within the design region. Hydraulic loads and shaft deflection shall be calculated in accordance with ANSI/HI 1.3. The design region is defined in ASME B73.1 as 110% of best efficiency flow and the minimum flows shown in ASME B73.1, Table 5, unless specifically noted otherwise by the manufacturer.

1.3.5.4.2.2 Overhung impeller pumps, API Standard 610

In accordance with API 610, the maximum shaft deflection at the location of the primary seal faces is 0.05 mm (0.002 in) at the most severe dynamic conditions over the complete head versus rate-of-flow curve, with a maximum diameter impeller operating at specified speed.

1.3.5.4.2.3 Overhung impeller pumps, ISO 5199

In accordance with ISO 5199, maximum shaft deflection is 0.05 mm (0.002 in) at the face of the seal chamber within the allowable operating range of the pump at operating speed.

Pump shafts should be designed to meet or exceed the appropriate specification to ensure satisfactory performance of the pump and sealing mechanisms.

If hydraulic radial load test data are not available, then hydraulic radial loads can be calculated in accordance with Section 1.3.5.1, Calculation of radial thrust for volute pumps.

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1.3.5.4.2.4 Shaft deflection calculation methods

Overhung impeller method of calculating shaft deflection (neglecting coupling weight)

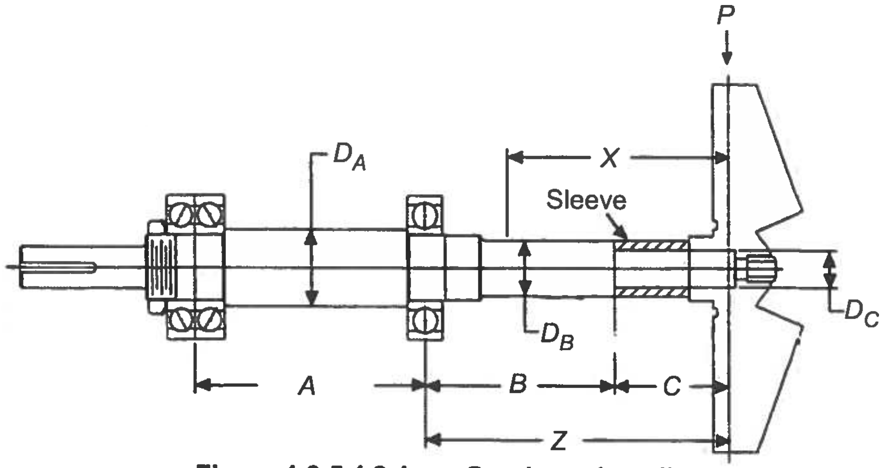

Figure 1.3.5.4.2.4a — Overhung impeller

For $0 \le X \le C$ $$\delta_X = \frac{P}{3E}\left\{\frac{ZA(Z-X)}{I_A} + C^3\left(\frac{1}{I_C}-\frac{1}{I_B}\right) + \frac{Z^3}{I_B} + \frac{X^3}{2I_C} - \frac{3X}{2}\left[C^2\left(\frac{1}{I_C}-\frac{1}{I_B}\right) + \frac{Z^2}{I_B}\right]\right\}$$

For $X > C$ $$\delta_X = \frac{P}{3E}\left[\frac{ZA(Z-X)}{I_A} + \frac{Z^3}{I_B} + \frac{X^3}{2I_B} - \frac{3XZ^2}{2I_B}\right]$$

Where:

$P = F_R$ = radial force acting at impeller location (I: consideration of radial thrust only), in N

$P = F_R + M_I \times g$ = radial force acting at impeller location (II: consideration of radial thrust and impeller weight), in N

M_I = impeller mass, in kg

X, Z, A, C = dimensions per Figure 1.3.5.4.2.4a, in mm (in)

D_B, D_C = dimensions per Figure 1.3.5.4.2.4a, in mm (in)

I_A, I_B, I_C = area moment of inertia ($I = \left(\frac{\pi}{64}\right)D^4$), in mm^4 (in^4)

E = Young's Modulus of Elasticity of shaft material, in N/mm^2 (psi)

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1.3.5.4.2.5 Overhung impeller method of calculating dry critical speed (neglecting coupling weight)

Shaft deflection at impeller location ($X = 0$) δ_X:

$$\delta_X = \frac{M_I g}{3E}\left\{\frac{Z^2 A}{I_A} + C^3\left(\frac{1}{I_C} - \frac{1}{I_B}\right) + \frac{Z^3}{I_B}\right\} \text{ mm (in)}$$

Where:

M_I = impeller mass, in kg (lbm)

g = 9.81 m/s^2 (32.2 ft/s^2) gravitational constant

E = Young's Modulus of Elasticity of shaft material, in N/mm^2 (psi)

D_A, A, B, C, Z = dimensions per illustration above, in mm (in)

I_A, I_B, I_C = polar area moment of inertia (e.g., $I_A = \frac{\pi D_A^4}{64}$), in mm^4 (in^4)

Shaft stiffness at impeller location K_S:

$$P = M_I g = K_S \delta_X$$

$$K_S = \frac{M_I g}{\delta_X} = \frac{3E}{\left\{\frac{Z^2 A}{I_A} + C^3\left(\frac{1}{I_C} - \frac{1}{I_B}\right) + \frac{Z^3}{I_B}\right\}} \text{ N/mm (lbf/in)}$$

Dry critical speed f_{DRY}:

$$f_{DRY} = \frac{1}{2\pi}\left(\frac{K_S}{M_I}\right)^{0.5} \text{ Hz}$$

Sample calculation – (metric units) (see Figure 1.3.5.4.2.4a)

Where:

A = 186.5 mm

B = 216 mm

C = 43 mm

D_A = 65 mm

D_B = 48 mm

D_C = 28 mm

M_I = 4.8 kg

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$E = 2.069 \times 10^5$ N/mm^2

$g = 9.81$ m/s^2

$Z = B + C = 216 + 43 = 259$ mm

$I_A = \frac{\pi D_A^4}{64} = \frac{\pi \times 65^4}{64} = 8.762 \times 10^5 \text{ mm}^4$ Similarly $I_B = 2.606 \times 10^5$ mm^4 and $I_C = 3.017 \times 10^4$ mm^4

Shaft deflection at the impeller ($X = 0$) is

$$\delta_X = \frac{4.8 \times 9.81}{3 \times 2.069 \times 10^5}\left\{\frac{259^2 \times 186.5}{8.762 \times 10^5} + 43^3\left(\frac{1}{3.017 \times 10^4} - \frac{1}{2.606 \times 10^5}\right) + \frac{259^3}{2.606 \times 10^5}\right\}$$

$$\delta_X = 6.318 \times 10^{-3} \text{ mm}$$

Shaft stiffness at the impeller location is

$$K_S = \frac{M_I g}{\delta_X} = \frac{4.8 \times 9.81}{6.319 \times 10^{-3}} = 7.453 \times 10^3 \text{ N/mm}$$

Dry critical speed is

$$f_{DRY} = \frac{1}{2\pi}\left(\frac{7.453 \times 10^3 \times 1000}{4.8}\right)^{0.5} = 198.3 \text{ Hz}$$

Sample calculation – (US customary units) (see Figure 1.3.5.4.2.4a)

Where:

$A = 7.34$ in

$B = 8.50$ in

$C = 1.69$ in

$D_A = 2.56$ in

$D_B = 1.89$ in

$D_C = 1.10$ in

$M_I = 10.582$ lb

$E = 3.000 \times 10^7$ psi

$Z = B + C = 8.50 + 1.69 = 10.20$ in

$I_A = \frac{\pi D_A^4}{64} = \frac{\pi \times 2.56^4}{64} = 2.108 \text{ in}^4$ Similarly $I_B = 6.264 \times 10^{-1}$ in^4 and $I_C = 7.187 \times 10^{-2}$ in^4

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Shaft deflection at the impeller ($X = 0$) is

$$\delta_X = \frac{10.582 \times 32.2}{3 \times 3.000 \times 10^7}\left\{\frac{10.20^2 \times 7.34}{2.108} + 1.69^3\left(\frac{1}{7.187 \times 10^{-2}} - \frac{1}{6.264 \times 10^{-1}}\right) + \frac{10.20^3}{6.264 \times 10^{-1}}\right\}$$

$$\delta_X = 8.011 \times 10^{-3} \text{ in}$$

Shaft stiffness at the impeller location is

$$K_S = \frac{M_I g}{\delta_X} = \frac{10.582 \times 32.2}{8.011 \times 10^{-3}} = 4.255 \times 10^4 \text{ lbf/in}$$

Dry critical speed is

$$f_{DRY} = \frac{1}{2\pi}\left(\frac{4.255 \times 10^4 \times 386.22}{10.582}\right)^{0.5} = 198.3 \text{ Hz}$$

Impeller Between-Bearings Method

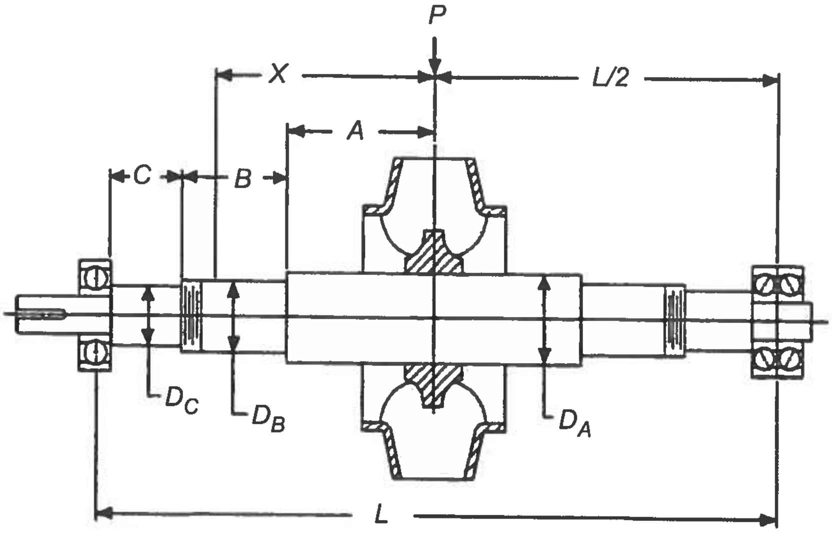

Figure 1.3.5.4.2.4b — Between-bearings, single-stage pump - method of calculating shaft deflection neglecting coupling weight

1.3.5.4.2.6 Between-bearing, single-stage pump calculation

For $X = 0$; 0 = Impeller centerline

$$\delta_0 = \frac{P}{6EI_C}\left\{C^3 + \frac{(C+B)^3 - C^3}{K_2} + \frac{(A+B+C)^3 - (B+C)^3}{K_3}\right\}$$

For $0 < X \leq A$

$$\delta_X = \delta_0 + \frac{P}{6EI_C}\left\{\frac{X^3 - 3X^2(A+B+C)}{2K_3}\right\}$$

For $A < X \leq (A + B)$

$$\delta_X = \delta_0 + \frac{P}{6EI_C}\left\{\frac{(A+B+C-X)^3 - (B+C-X)^3 - 3A(B+C)^2}{2}\left(\frac{K_2 - K_3}{K_2K_3}\right) - \frac{3X^2}{2}\left(\frac{A}{K_3} + \frac{B+C}{K_2}\right) + \frac{X^3}{2K_2}\right\}$$

For $(A + B) < X \leq (A + B + C)$

$$\delta_X = \frac{P(A+B+C-X)}{4EI_C}\left\{C^2 + \frac{(B+C)^2 - C^2}{K_2} + \frac{(A+B+C)^2 - (B+C)^2}{K_3} - \frac{(A+B+C-X)^2}{3}\right\}$$

Where:

$P = F_R$	=	radial force acting at impeller location (I: consideration of radial thrust only), in N
$P = F_R + M_I{}^*g$	=	radial force acting at impeller location (II: consideration of radial thrust and impeller weight), in N
M_I	=	impeller mass, in kg
A, B, C, X	=	dimensions per Figure 1.3.4.2.4b, mm (in)
D_A, D_B, D_C	=	dimensions per Figure 1.3.4.2.4b, mm (in)
I_A, I_B, I_C	=	area moment of inertia [$I = \frac{\pi}{64}D^4$], mm^4 (in^4)
E	=	Young's Modulus of Elasticity of shaft material, in N/mm^2 (psi)
K_2	=	$\frac{I_B}{I_C}$ = shaft area moment ratio, B to C (dimensionless)
K_3	=	$\frac{I_A}{I_C}$ = shaft area moment ratio, A to C (dimensionless)

NOTES:

1) These calculations do not consider contributions of any shaft sleeve to the stiffness of the shaft, and the additional mass has negligible effect on deflection.

2) Equations do not account for any support the pump shaft might receive from hydrostatic stiffness at the impeller wear rings and seal chamber throat bushing.

3) Equations do not account for internal looseness in the bearings, dynamic radial loads caused by impeller imbalance, or shaft runout.

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1.3.5.4.2.7 Between-bearings impeller method of calculating dry critical speed (neglecting coupling weight)

Shaft deflection at impeller location (X = 0) δ_0

$$\delta_0 = \frac{M_I g}{6EI_C}\left\{C^3 + \frac{(C+B)^3 - C^3}{K_2} + \frac{(A+B+C)^3 - (B+C)^3}{K_3}\right\} \text{ mm (in)}$$

Where:

M_I = impeller mass, in kg (lb)

g = 9.81 m/s^2 (32.2 ft/s^2) gravitational constant

E = Young's Modulus of Elasticity of shaft material, in N/mm^2 (psi)

A, B, C = dimensions per illustration above, in mm (in)

I_A, I_B, I_C = polar area moment of inertia (e.g., $I_A = \frac{\pi D_A^4}{64}$), in mm^4 (in^4)

Shaft stiffness at impeller location K_S:

$$P = M_I g = K_S \delta_X$$

$$K_S = \frac{M_I g}{\delta_X} = \frac{3E}{\left\{\frac{Z^2 A}{I_A} + C^3\left(\frac{1}{I_C} - \frac{1}{I_B}\right) + \frac{Z^3}{I_B}\right\}} \text{ N/mm (lbf/in)}$$

$$K_S = \frac{M_I g}{\delta_X} = \frac{6EI_C}{\left\{C^3 + \frac{(C+B)^3 - C^3}{K_2} + \frac{(A+B+C)^3 - (B+C)^3}{K_3}\right\}} \text{ N/mm (lbf/in)}$$

Dry critical speed f_{DRY}:

$$f_{DRY} = \frac{1}{2\pi}\left(\frac{K_S}{M_I}\right)^{0.5} \text{ Hz}$$

Sample calculation – (metric units) (see Figure 1.3.5.4.2.4b)

Where:

A = 240 mm

B = 170 mm

C = 140 mm

D_A = 90 mm

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D_B = 80 mm

D_C = 75 mm

M_I = 100 kg

E = 2.10×10^5 N/mm^2

Shaft seal location = 320 mm from impeller centerline

$$I_A = \frac{\pi D_A^{\ 4}}{64} = \frac{\pi \times 90^4}{64} = 3.221 \times 10^6 \text{ mm}^4 \quad \text{Similarly, } I_B = 2.011 \times 10^6 \text{ mm}^4 \text{ and } I_C = 1.553 \times 10^6 \text{ mm}^4$$

Shaft area moment ratio, from B to C is

$$K_2 = \frac{I_B}{I_C} = \frac{2.011 \times 10^6}{1.553 \times 10^6} = 1.295$$

Similarly, shaft area moment ratio, from A to C is

$$K_3 = 2.074$$

Shaft deflection at the impeller centerline ($X = 0$) is

$$\delta_0 = \frac{100 \times 9.81}{6 \times 2.10 \times 10^5 \times 1.553 \times 10^6}\left\{140^3 + \frac{(140+170)^3 - 140^3}{1.295} + \frac{(240+170+140)^3 - (170+140)^3}{2.074}\right\}$$

$$\delta_0 = 4.487 \times 10^{-2} \text{ mm}$$

Shaft stiffness at the impeller centerline ($X = 0$) is

$$K_S = \frac{M_I g}{\delta_0} = \frac{100 \times 9.81}{4.487 \times 10^{-2}} = 2.186 \times 10^4 \text{ N/mm}$$

Dry critical speed is

$$f_{DRY} = \frac{1}{2\pi}\left(\frac{2.186 \times 10^4 \times 1000}{100}\right)^{0.5} = 74.4 \text{ Hz}$$

Sample calculation – (US customary units) (see Figure 1.3.5.4.2.4b)

Where:

A = 9.449 in

B = 6.693 in

C = 5.512 in

D_A = 3.543 in

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D_B = 3.150 in

D_C = 2.953 in

M_I = 220.5 lb

E = 3.05×10^7 psi

Shaft seal location = 12.599 in from impeller centerline

$$I_A = \frac{\pi D_A^4}{64} = \frac{\pi \times 3.543^4}{64} = 7.735\ \text{in}^4 \quad \text{Similarly, } I_B = 4.833\ \text{in}^4 \text{ and } I_C = 3.733\ \text{in}^4$$

$$K_2 = \frac{I_B}{I_C} = \frac{4.833}{3.733} = 1.295 \quad \text{Similarly, } K_3 = 2.072$$

Shaft deflection at the impeller centerline (X = 0) is

$$\delta_0 = \frac{220.5 \times 32.2}{6 \times 3.05 \times 10^7 \times 3.731}\left\{5.512^3 + \frac{(5.512 + 6.693)^3 - 5.512^3}{1.295} + \frac{(9.449 + 6.693 + 5.512)^3 - (6.693 + 5.512)^3}{2.074}\right\}$$

$$\delta_0 = 0.058\ \text{in}$$

Shaft stiffness at the impeller centerline (X = 0) is

$$K_S = \frac{M_I g}{\delta_0} = \frac{220.5 \times 32.2}{0.058} = 1.225 \times 10^5\ \text{lbf/in}$$

Dry critical speed is

$$f_{DRY} = \frac{1}{2\pi}\left(\frac{1.225 \times 10^5 \times 386.22}{220.5}\right)^{0.5} = 73.7\ \text{Hz}$$

As an alternative to the hand calculations, finite element analysis (FEA) can be used to analyze the complete shaft loading. This includes not only the deflection due to bending but torsional deflection as well. Care must be taken to accurately mesh the shaft model to fully capture the affects of the shaft features such as keyways, snap ring grooves, fillets, and threaded portions. These can create regions of stress concentration that can reduce the overall load capacity of the shaft. These not only affect the shaft stress in a steady state condition, but often more importantly, in the fatigue life of the shaft.

1.3.5.5 Guidelines for bearings and lubrication methods

1.3.5.5.1 Introduction

The type of bearings and lubrication system selected for a given pump application depends on the magnitude and direction of radial and axial loads acting on the rotor. Factors affecting loading are rotative speed, pump casing design, tolerance, misalignment, rotor unbalance, service conditions, pump operating point, and type of pump supplied.

Rolling element bearings will typically have associated with them an L_{10} lifetime based on speed, operating loads, and manufacturer's rated capacity. See Section 1.3.5.5.3. Nominally, this means that 90% of all pump bearings

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should still be serviceable after this lifetime has elapsed. The lifetime (fatigue failure point) is rarely realized because premature failure occurs as a result of static overload, corrosion, lubricant failure, contamination, or overheating. Selection and maintenance of the lubrication system is a critical factor in improving actual bearing life.

Selection of bearing type and lubrication method are usually a part of the pump design process. The bearing type and lube system design become an integral part of the overall pump design. The end user may not have a choice, or options may be limited, as to alternative bearing types and lubrication methods.

1.3.5.5.2 Bearing types

1.3.5.5.2.1 Rolling element bearings

Various rolling element bearing designs are shown in Table 1.3.5.5.2.1. This table is not meant to be all-inclusive, but to represent the most common designs used in pumps. Generally, ball bearing designs will tolerate misalignment better than cylindrical or tapered roller bearing designs. The ball bearing designs tend to run cooler than roller bearings due to lower friction but demonstrate lower load capacities. These designs are found on many chemical, petroleum, food, and other process applications. Cylindrical roller bearings are typically used for slower speed, high radial load applications, such as slurry transfer service. The use of spherical roller bearings provides for higher load capacity than with the cylindrical roller bearing while also allowing greater misalignment tolerances and some thrust loading. The tapered roller bearing exhibits the highest thrust and radial load ratings but is more sensitive to misalignment and poor lubrication.

Table 1.3.5.5.2.1 — Rolling element bearing type

Bearing Type	Description
Single row deep groove ball bearing	Accept low axial loads in either direction and radial loads.
Double row angular contact ball bearing (Conrad style)	Accept high axial loads in either direction and radial loads.
Duplex (2) angular contact ball bearings	Designed for improved higher axial loads capacity than double row with good radial loads.
Triplex (3) angular contact ball bearings	Greater axial loads rating, but cooling and lubrication more critical due to higher bearing friction.
Cylindrical roller bearings	Designed specifically for very high radial loads. Alignment tolerances critical.
Spherical roller bearings	Excellent for high radial loads, some axial loads capacity. Spherical outer race design increases alignment tolerances.
Duplex (2) tapered roller bearings	Excellent axial load and radial load capacity. Alignment, cooling, and lubrication very critical.

1.3.5.5.2.2 Sleeve bearings

Sleeve bearings operate on a hydrodynamic principle. The proper application of this principle depends on the maintenance of a lubricating film between wear surfaces. This film is dependent on lubricant viscosity, shaft speed, bearing running clearance, length, and radial pressure on the bearing. Heat buildup and wear can be appreciable if there is insufficient lubrication. Subsynchronous shaft whirling (orbiting) in the bearing occurs when the lubrication film viscosity and thickness are not adequate to dampen the dynamic action of the shaft. Material contact can occur when the lubrication film viscosity and thickness are not adequate to dampen the dynamic action of the shaft. Shaft and bearing materials must be carefully chosen to minimize the coefficient of friction and optimize wear properties.

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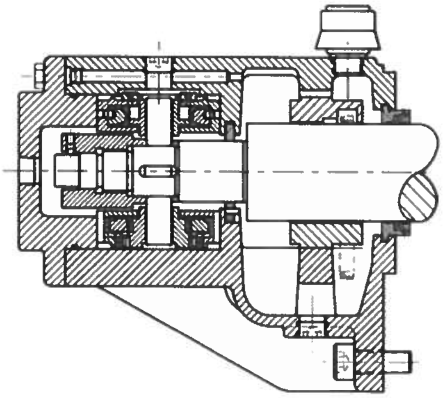
Sleeve bearings can be cylindrical, grooved, elliptical, lobed, or tilting-pad designs. Cylindrical offers the advantage of simple design. The groove design facilitates the expulsion of solids and contaminants from the bearing. Elliptical, lobed, and tilting-pad designs have lower load capacities but demonstrate greater shaft stability. They are particularly useful for high-speed, low-load applications. An exception to this is the pivot shoe or tilting-pad-type journal bearing, which exhibits very high radial thrust load capacity.

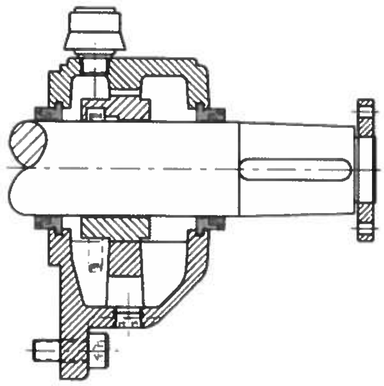
In some cases, sleeve bearings are used both on the inboard and outboard for the radial thrust with an antifriction bearing outboard of the outboard sleeve bearing to absorb the axial thrust.

Sleeve with pivot-shoe-type thrust bearing

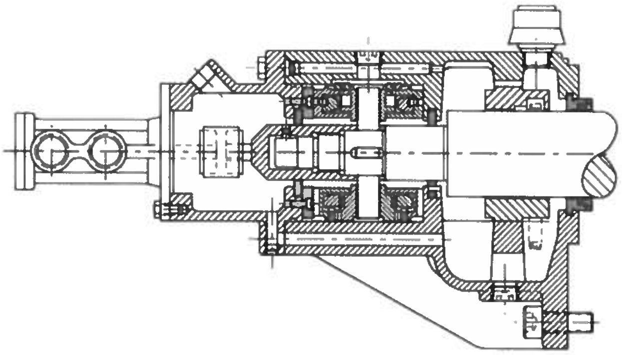
The sleeve bearing is comprised of a journal, usually the shaft itself, and a bearing bushing that is horizontally split and lined with a suitable material with the proven tribological properties of low friction, high lubricity, and acceptable wear resistance (such as babbitt). Outboard of the outboard sleeve bearing is a pivot-shoe-type thrust bearing. Pumps with this bearing arrangement usually have a separate lubrication system with its own reservoir, oil pump, and cooler.



Pivot-shoe-type thrust bearing



Cylindrical-type radial journal (sleeve) bearing



Pivot-shoe-type thrust bearing – complete with shaft-driven oil pump arranged for force feed lubrication

Figure 1.3.5.5.2.2a — Sleeve bearings

Sealless magnetic drive and canned motor type pumps also use cylindrical or grooved type product-lubricated bearings. Common materials for these applications are carbon, silicon carbide, and graphite.

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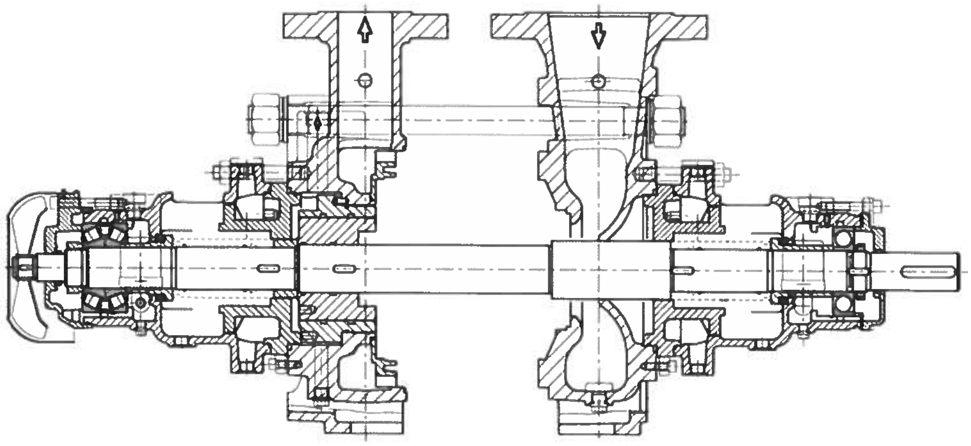

Sectional ring casing pumps (type BB4 multistage radial split single casing) have many types of bearings: ball bearings, roller bearings, and sleeve bearings. Some designs offer product-lubricated bearings in the suction nozzle.

Figure 1.3.5.5.2.2b — Type BB4 multistage radial split single casing pump with ball and roller bearings

1.3.5.5.3 Bearing life (horizontal pumps)

1.3.5.5.3.1 Introduction

Bearing minimum load

All rolling element bearings require a minimum applied load. This is necessary to ensure that normal elasto-hydrodynamic operation occurs. If too small a load is applied, then the bearing rolling elements (balls or rollers) can skid as opposed to rolling. When skidding occurs, wiping of the lubrication film typically results due to the less favorable surface motion. This in turn can cause metal-to-metal contact and consequent reduction in the life of the bearing.

The minimum load requirements vary depending on the bearing design.

As a general rule, the larger the area of contact between the rolling element and the raceway, the greater the minimum load required. Thus a cylindrical roller bearing (which has a line contact) has a higher minimum load requirement than a deep-groove ball bearing (which has a point contact).

Variations in bearing internal geometry, tolerances, and surface finish can also affect the minimum load requirements to a lesser extent.

Rolling element bearing manufacturers normally provide data tables or equations to allow the designer to calculate the minimum load required.

The practical consequence of bearing minimum load requirements is that if the result of the L_{10} life calculation of a bearing is very high, the bearing may fail prematurely due to insufficient bearing load being present to prevent skidding.

This stands in contradiction to the tendency of users to specify high L_{10} life requirements. As a general guide it is recommended that bearing life requirements not be set higher than 100,000 hours under the worst-case combination of loading.

Pumps with single volutes have a very large variation between the hydraulic radial thrust at minimum flow compared to BEP flow. There is a considerable risk that in designing for a high L_{10} life at minimum flow, the minimum

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load requirements are not met at BEP. These pumps should always be checked against the minimum bearing load requirements.

If the bearing minimum load requirement cannot be met, the designer can either select a bearing with a lower minimum load requirement or find a way to increase the load on the bearing. For example, the bearing load can be increased by changing the pump hydraulic design (increased hydraulic load) or by applying mechanical loading (often using springs to generate preload).

1.3.5.5.3.2 Bearing life (horizontal pumps)

This specification covers all rolling element bearings used to support rotodynamic pump rotors. It does not cover plain bearings or bearings of any type used in accessory equipment, such as speed reducers or drivers.

The following paragraphs present some of the basic methods currently used for calculating theoretical bearing life. Methods 1, 2, and 3 are presented in increasing order of sophistication and are termed *basic rating life, adjusted rating life*, and *modified rating life.*

With the increasing degree of sophistication comes a reduced level of uncertainty and risk.

It becomes incumbent on the user to decide which method is best suited to the evaluation being performed considering the desired reliability and with the operational information available.

1.3.5.5.3.3 Definitions

Adjusted rating life, L_{na}, millions of revolutions. A rating life obtained by adjustment of the basic rating life for different reliability levels, special bearing materials, and specific operating conditions. Adjusted rating life is rating life adjusted for 90% or other reliability, and/or special bearing properties, and/or nonconventional operating conditions.

Axial load, F_a, newtons (pounds). The total purely axial load imposed on a bearing. Included is any assembly preload and the external axial loads carried by the bearing.

Basic dynamic radial load rating, C_r, newtons (pounds). The constant stationary radial load a rolling element bearing could theoretically endure for a basic rating life of one million revolutions and 90% reliability. Care must be taken to establish the correct value to be used for C_r because this value may vary between bearings of the same size and type supplied by different manufacturers. Values may be taken from the published vendor catalogs or alternately may be calculated following the equations provided in ISO 281 (1990) or ABMA Standards 9 and 11 (1990).

Basic rating life, L_{10}, million of revolutions. Rating life associated with 90% reliability for bearings manufactured with commonly used high-quality bearing steel material, of good manufacturing quality, and operating under conventional operating conditions assuming proper mounting and lubrication. It should be noted that the L_{10} basic rating life is a rated fatigue life. Actual service life may differ due to the effects of wear, noise, corrosion, etc. Some of these affects can also be estimated. (See theory on adjusted rating life.)

Dynamic equivalent radial load, P_r, newtons (pounds). An equivalent stationary radial load, calculated by combining the actual radial and axial loads, under the influence of which a rolling element bearing would have the same rating life as it will attain under the actual load condition.

Radial load, F_r, newtons (pounds). The purely radial external load imposed on a bearing.

Reliability. For a group of apparently identical rolling element bearings, operating under the same conditions, the percentage of the group expected to attain or exceed a specified life.

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Life. The number of revolutions a bearing makes before the first visible evidence of fatigue develops in the material of one of the races or rolling elements.

Modified rating life, L_{nm}, millions of revolutions. A method of life calculation with reference to ISO 281: 1990 (Amendment 2: 2000). This method adjusts the traditional L_{10} rating with a reliability factor, a_1, and a life modification factor, a_{xyz}, to account for the influence of material, lubrication, environment, contamination particles, internal stresses in the bearing rings, mounting, and bearing load.

Factors

Factor a_1 = Reliability adjustment factor. A factor varying between 0.21 for 99% reliability and 1.0 for 90% reliability. Defined in ISO 281 (1990) and ANSI/ABMA Standards 9 and 11 (1990). According to Table 1.3.5.5.3.5, such that for 90% reliability a_1 = 1.0. Other common factors are for 95% reliability a_1 = 0.66 and for 99% reliability a_1 = 0.21.

Factor a_2 = Material adjustment factor. The factor a_2 is the material adjustment factor. A bearing may require special properties as regards to life by the use of a special type and quality of material and/or special manufacturing processes or design. Such special life properties are taken into account by the application of the life adjustment factor a_2. The present state of knowledge does not make it possible to define relationships between the values of a_2 and quantifiable characteristics of the material or bearing raceway geometry, for example. The values of a_2 have to be based on experience and are usually obtained from the specific manufacturer of the bearing. The use of a certain steel analysis and/or process is not sufficient justification for the use of an a_2 greater than 1.0. Values of a_2 greater than 1.0, however, may be applicable to bearings made of steel of particularly low impurity or of special analysis as long as other environmental conditions are satisfactory.

Factor a_3 = Operating conditions adjustment factor. Considered to include the adequacy of the lubrication condition (at the operating speed and temperature), and operating conditions causing changes in the material properties. Manufacturers of bearings are expected to supply recommendations regarding appropriate values of a_3 to be used with their bearings. Where specific system requirements of lubrication and cleanliness are fulfilled, a_3 is equal to 1.0.

Discussion

The factor a_3 is the operating conditions adjustment factor. Operating conditions to be considered here include the adequacy of the lubrication (at the operating speed and temperature), presence of foreign matter, and conditions causing changes in the material properties (for example, high temperature causing reduced raceway hardness). The calculation of the basic rating life assumes the lubrication is adequate, i.e., that the lubricant film in the rolling element/raceway contacts has a thickness equal to or slightly greater than the composite roughness of the contact surfaces. Where this requirement is fulfilled, a_3 is equal to 1.0. Usually values of less than 1.0 should be considered where the kinematic viscosity of the lubricant at the operating temperature is less than 13 mm^2/s (13 cSt) for ball bearings or 20 mm^2/s (20 cSt) for roller bearings or where the rotational speed is very low. Lubricant films greater than the composite surface roughness can result in values of a_3 greater than 1.0. Manufacturers of bearings are expected to supply recommendations regarding appropriate values of a_3 to be used with their bearings because it may not be assumed that the use of a special material, process, or design will overcome a deficiency in lubrication and the limitation of not using an a_2 greater than 1.0 with an a_3 less than 1.0. Similarly, the basic rating life assumes no foreign particles are present in the lubricant. For pumps installed in dusty environments, or where large temperature swings may cause condensation in the bearing cavity, a value of a_3 less than 1.0 should be considered. For extremely clean environments, such as may be obtained with an oil mist lubrication, it may be justifiable to select a value of a_3 greater than 1.0.

It is common to have manufacturers combine factors a_2 and a_3 into a single factor a_{23} recognizing the interaction of material properties and lubrication conditions.

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Factor a_{xyz} = Life modification factor (used in modified rating life theory). Based on a system's approach according to ISO 281/Amendment 2 (February 2000). Parameters having a significant influence on the life of rolling bearings are:

- Contamination level
- Particle size
- Particle shape
- Particle hardness
- Operational lubrication condition
- Bearing load
- Material fatigue stress limit
- Bearing type

From these variables three factors are derived: contamination index, fatigue load limit, and equivalent bearing load.

1.3.5.5.3.4 Method 1 – Basic rating life L_{10}

Loads on all bearings are calculated. Sources of loads that must be accounted for are: radial thrust due to hydraulic force on the impeller and axial thrust due to hydraulic force on the impeller and the shaft sealing element.

a) Static weight of the shaft, impeller, and coupling.

b) Loads due to belt drives attached directly to the pump shaft.

c) Bearing mounted preloads.

Loads due to coupling misalignment, rotor unbalance, or driver thrust (unless quantifiable) in properly installed pumps are normally not significant and are not included.

The dynamic equivalent radial load (P_r) is calculated as:

$$P_r = XF_r + YF_a$$

where values for X and Y can be found in ISO 281 or ANSI/ABMA Standard 9 for ball bearings, and Standard 11 for roller bearings. X is the coefficient of radial dynamic flow, Y is the coefficient of axial dynamic flow, F_r is radial thrust in newtons (pounds force), and F_a is axial thrust in newtons (pounds force).

The basic rating life in operating hours is calculated from:

$$L_{10} = \left(\frac{C_r}{P_r}\right)^3 \times \frac{10^6}{n \times 60} (\textit{Ball bearings})$$

$$L_{10} = \left(\frac{C_r}{P_r}\right)^{10/3} \times \frac{10^6}{n \times 60} (\textit{Roller bearings})$$

C_r is the single bearing basic radial load rating and P_r is the single bearing equivalent dynamic radial load. For paired bearings, a multiplication factor must be used to obtain the paired bearing load rating or system life. Consult individual manufacturer's catalogs or ANSI/ABMA Standard 9 or 11.

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Stacking bearings together to improve load-carrying capacity and life does not necessarily increase the capacity proportional to the number of bearings stacked. The reader is referred to ANSI/ABMA Standard 9 or 11 (or ISO 281) for further details of calculating the load-carrying capacity and life of paired or stacked bearings. Some manufacturers are now using different methods for calculating stacked bearings based on the bearing type and on their ability to precision grind and obtain proper mounted load sharing.

1.3.5.5.3.5 Method 2 – Adjusted rating life L_{na}

The basic rating life is associated with a 90% reliability for contemporary and commonly used bearing materials, manufacturing quality, and for conventional operating conditions. Some manufacturers may offer materials with a higher fatigue life or modified designs that will result in a greater bearing life. Also, many applications involve lubricating conditions that may deviate from basic assumed practice. It also may be desirable to calculate life for different reliability. ISO 281 and ABMA Standards 9 and 11 provide for a more accurate life calculation by adjusting the basic rating life by means of three factors: Factor a_1, Factor a_2, and Factor a_3.

Table 1.3.5.5.3.5 — Life adjustment factors

$L_{na} = a_1 a_2 a_3 L_{10}$	Reliability, %	L_{na}	a_1
	90	L_{10}	1
	95	L_5	0.66
	96	L_4	0.53
	97	L_3	0.44
	98	L_2	0.33
	99	L_1	0.21

1.3.5.5.3.6 Method 3 – Modified rating life L_{nm}

This is the method of life calculation with reference to ISO 281: 1990 (Amendment 2: 2000, Life modifications factor a_{xyz}).

This method adjusts the traditional L_{10} rating with a reliability factor, a_1, and a factor to account for the influence of material, lubrication, environment, contamination particles, internal stresses in the bearing rings, mounting, and bearing load, a_{xyz}.

The a_1 factor allows you to calculate a rating life for higher reliability than the traditional 90% provided with the L_{10}. ISO introduced the a_2 and a_3 factors in 1977 to include the effect of material and lubrication on bearing life. Since these factors are interdependent, they were later combined into the a_{23} factor. ISO 281, Amendment 2, introduces the factor a_{xyz}, which further extends outside effects to include environment, contamination, internal stresses, mounting, and bearing load.

$$L_{nm} = a_1 \times a_{xyz} \times L_{10}$$

Where:

L_{nm} = modified rating life, in millions of revolutions

a_1 = life adjustment factor for reliability

a_{xyz} = life modification factor

L_{10} = basic rating life, in millions of revolutions

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This standard identifies factors other than bearing load that will affect the bearing service life. However, the standard does not quantify the effects of the factors and leaves it up to the individual to determine them or contact the specific manufacturer for recommendations.

Discussion

It is possible to perform calculations with an anticipated life based on established fatigue stress limits. ISO 281, Amendment 2, recognizes that bearing fatigue life is the result of the interaction of the combined internal stresses generated from all of the various environmental factors and the material strength. While there is general agreement as to material stress fatigue limits, internal stresses are greatly affected by bearing design, manufacturing processing, and hardening processes. Therefore it is required to consult with individual manufacturers for specific values of bearing fatigue load or stress limits.

The calculation methods pertain to the fatigue life of the bearing. Other mechanisms like wear, corrosion, or grey-staining (micropitting) are not considered. Operating conditions deviating from those described in ISO 281 (1990), such as misalignment and operating clearance, are taken into account separately.

Under favorable operating conditions, modern bearings of high quality can achieve lives far exceeding the calculated rating life according to ISO 281 (1990). Under ideal conditions, i.e., with complete separation of the contacting surfaces by a lubricant film and no surface damage caused by contamination, virtually infinite life can be achieved. In contrast the previous (historically-used) calculation method described in ISO 281 (1990) always gives a finite life.

State-of-the-art calculation models for the fatigue life in rolling contacts are based on a fatigue limit of the bearing material. The classic calculation model of Lundberg and Palmgren is extended by a fatigue limit σ_u below which, under ideal conditions, no fatigue of the material will occur.

Parameters having a significant influence on the life of rolling bearings are

- Bearing load
- Lubrication regime (viscosity and type of lubricant, speed, bearing size, additives)
- Material properties (cleanliness, hardness, fatigue limit, surface structure, response to temperature)
- Bearing design (internal load distribution, friction conditions within the bearing, raceway profiles)
- Residual stresses in the material (from manufacturing, heat treatment, and from ring interference after mounting)
- Environment (contamination of the lubricant, humidity)

A number of additional terms are needed in the calculation of life modification factor a_{xyz} in order to introduce an estimation of bearing fatigue limits. These terms include:

σ_u = endurance stress limit used in fatigue criterion, in N/mm^2

C_u = fatigue limit load

κ = kappa – viscosity ratio = oil viscosity actual/minimum required oil viscosity

η_c = eta$_c$ – contamination factor

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Table 1.3.5.5.3.6 — Oil contamination factor η_c

Range of η_c Contamination factor	Description
0.01 to 0.1	**Heavily contaminated** Typical for open bearings in gearboxes with no filtration, or greased open bearings where the housing seals are inadequate for the contaminant levels in the environment.
0.1 to 0.3	**Contaminated** Typical of bearings without integral seals, coarse lubricant filters, and/or particle ingress from the surroundings, where lubrication intervals are inadequate.
0.3 to 0.5	**Normal** Typical for bearings greased for life and shielded. Normal workshop cleanliness. Particle size typically up to two orders of magnitude larger than the film thickness. Continuous bypass filtration of the oil.
0.5 to 0.8	**Clean** Typical for bearings greased for life and sealed. Particle size typically up to one order of magnitude larger than the film thickness. Continuous fine filtration of the oil.
0.8 to 1.0	**Very clean** Particle size the same order of magnitude as the lubricant film thickness. Obtained with the continuous mainstream ultrafine filtration of circulating oil. Typical of laboratory test conditions.

Modern computing technology makes it possible to take these parameters into account in the rating life calculation. For a practical calculation method, a life adjustment factor can be derived from

$$a_{xyz} = f\left[\frac{\eta_c \times C_u}{P_r}, \kappa\right]$$

The fatigue limit load C_u takes into account the fatigue limit of the raceway material while the contamination factor η_c describes the stress increase caused by solid contaminants in the bearing.

According to ISO 281/Amendment 2 (Feb. 2000), the life modification a_{xyz} depends on the ratio of the fatigue limit σ_u of the bearing material and the real subsurface stress σ used in the fatigue criterion.

$$a_{xyz} = f\left[\frac{\sigma_u}{\sigma}\right]$$

For rolling bearings of modern high-quality manufacture and material, the fatigue limit is reached at a contact stress of about 1500 N/mm^2 (22,000 lbf/in^2). The fatigue limit is lower in the case of reduced manufacturing or material quality and can be increased under ideal contact conditions.

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The fatigue-determining subsurface stress in the raceway mainly depends on the bearing internal load distribution and the distribution of subsurface stress in the most heavily loaded contact. To facilitate the practical calculation, a fatigue load limit C_u is introduced. Values of the fatigue limit load should be requested from the bearing manufacturer.

1.3.5.5.4 Bearing temperature

Bearings have upper temperature limits for dimensional stability. The upper temperature limit is sometimes referred to as the *heat stabilization temperature*. It is usually 121 °C (250 °F) for off-the-shelf ball bearings. The operating temperature of a bearing housing and bearing lubricant will vary greatly with the design and application. In many cases, churning of the lubricant is the greatest contributor to bearing heating. Exceeding the design oil level or overgreasing a bearing can raise the bearing operating temperature and adversely affect the viscosity of the lubricant. Bearing temperature increases of more than 20 °C (38 °F) due to this effect are quite typical. (Note also that immediate signs of high temperatures after start-up can often be attributable to misalignment of bearing assembly parts.)

Achieving the design bearing life is contingent on maintaining an adequate quality and viscosity of the lubricant. In general, the minimum lubricant viscosity at the actual bearing operating temperature should be 13 cSt (70 SSU). For greases this should be the viscosity of the base oil. Specific bearing manufacturing quality may allow for lower viscosity oils, and the use of synthetic oils can either raise or lower the required viscosity. In greases, the NLGI Consistency Grade is equally important.

Mineral oils degrade rapidly at elevated temperatures. A rule of thumb is that mineral oil life decreases by half for every 15 °C above 70 °C. For this reason, it is often recommended that a maximum lubricant temperature of 82 °C (180 °F) be observed when using mineral oils.

This maximum lubricant temperature depends on the viscosity grade and minimum required viscosity for adequate bearing lubrication. The 82 °C (180 °F) maximum stated above is currently valid for ISO VG 68 with a Viscosity Index (VI) of 85 because that is the temperature at which it stops providing the minimum 13 cSt required by ball bearings.

Lubricant temperature should be directly measured in the sump as the oil drains off the bearings. If that is not practical, the lubricant temperature can be estimated at 11 °C (20 °F) above the outside temperature of the bearing housing at the bearing. However, this will be invalidated if the bearing housing has an internal water jacket or fan-cooling feature.

1.3.5.6 Bearing lubrication

1.3.5.6.1 Grease

Grease lubrication is often used with rolling element bearings because this design facilitates the thorough dissemination of lubricant over the wear surfaces. Internal grease is heated to semiliquid while cooler grease at the bearing–housing interface remains solid and acts as a barrier to external contaminants as well as providing a reservoir of lubricant between regrease intervals. Overheating due to inadequate circulation can occur if too much grease is injected into the bearing. A rule of thumb is to fill the bearing housing two-thirds full of grease. This allows room for circulation and expansion of grease.

Grease formulations are currently available that are optimized for specific types of environment. The proper grease should be chosen based on such factors as moisture, bearing load, speed, and other conditions. Consult a grease manufacturer for selecting the correct lubricant for special environments.

Commonly available grease typically has a maximum operating temperature of 121 °C (250 °F), but it is usually limited to a service temperature of 93 °C (200 °F). Shielded bearings are usually limited to a maximum operating temperature of 52 °C (125 °F).

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Grease is used on tapered roller bearings except at low speeds.

Shielded/sealed rolling element bearings provide economical low maintenance or maintenance-free service on some limited applications. The bearing is pregreased and permanently shielded or sealed. Shielding is accomplished by using rigid covers attached to the stationary race with a close clearance to the rotating race. Sealing is accomplished by a flexible element attached to the stationary race and rubbing against the rotating race. Care must be taken in applying shielded/sealed bearings because there may be no provision to renew the lubricant. However, provisions can be made on shielded bearings to regrease through the running clearance between the shield and inner bearing race. Care must be taken in the application of single shielded bearings. The shield must be toward the grease source to prevent overgreasing the bearing.

1.3.5.6.2 Oil lubrication

Various forms of oil lubrication exist. The most common form of bearing lubrication is direct contact. As the shaft rotates, the rolling elements in the bearing make contact with a level of oil. If the level of lubricant is too high or too low, excessive heat will be generated accelerating the degradation of the oil and shortening the life of the bearing.

Oil rings or flingers are used either to augment flood oil circulation or as a separate lubrication design entirely. For the latter case, the oil level remains below the lowermost bearing ball and the rings are the sole source of lubricant dispersion. Heat buildup due to the oil being churned by the bearing balls is eliminated while the ring action substantially improves circulation cooling. Oil rings also deliver a finite supply of oil to the bearings. Oil rings and flingers are superior to flood lubrication, particularly at high speeds. Flood lubrication should not be used above established limits, i.e.; $Dmn > 300{,}000$, where $Dm = 0.5\ (ID + OD)$ in mm, and n = operating speed, in rpm; and at loads where oil churning can significantly increase heat buildup. However, speed limits exist with oil rings to ensure proper ring rotation on the shaft without slippage. An evaluation must be made for high-temperature services, as oil cooling may be required to keep bearing temperature to an acceptable level.

Oil mist lubrication is a centralized system providing a continuous pressurized feed of atomized oil throughout the bearing housing. Purge oil mist uses a conventional oil sump, with the mist being used to purge the housing and replenish nominal oil loss. This action facilitates dissemination of oil over the bearings, improves cooling, and prevents the intrusion of ambient air and moisture. Oil in the mist will condense in the bearing housing and increase the level in the sump. Provision must be made to drain off this excess oil.

Purge oil mist does not eliminate the potential for bearing oxidation and wear from oil additive breakdown and wear particle contamination. The oil in the sump must still be changed periodically.

With pure oil mist, no lubricating oil sump is necessary. The bearing is lubricated by a continuous supply of fresh oil. Turbulence generated by bearing rotation causes oil particles suspended in the oil mist stream to condense on the rolling elements.

Pure oil mist prevents oil sump contamination from wear particles and the breakdown of oxidation preventive additives. As a result, higher permissible bearing temperatures are possible. Additionally, wear particles are continuously washed away. No periodic oil changes are necessary, although proper handling of the condensate/draining must be considered.

Oil mist systems will be greater in capital cost but lower in maintenance costs. A fail-safe system is desirable to shut the pump down in the event of lubrication system failure.

Force feed lubrication is used to augment circulation of oil throughout the bearing. Horizontal sleeve and tilting-pad (pivot-shoe-type thrust bearings, see Figure 1.3.5.5.2.2a) bearings may require this feature to ensure that oil is properly dispersed over the bearing pad or load area. Oil may be pressurized by a shaft-driven pump or independently driven pump. Various filters, pressure and temperature gauges, and heat exchangers can be installed to monitor the oil. The determining parameters for using a force feed system with sleeve bearings are loading, oil temperature, and bearing wear surface speed. It is critical that a fail-safe system be employed to shut the pump down upon a feed system failure.

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1.3.5.6.3 Product lubrication

Product-lubricated bearings are usually of the sleeve design. They are typically found in sealless magnetic drive and canned motor pumps. Spring-loaded, grooved, and conical designs are all used. Certain material combinations have proven success in the right applications, including

a) Carbon versus silicon carbide.

b) Silicon carbide versus silicon carbide.

c) Silicon carbide versus chrome oxide.

Product lubrication is introduced to the bearing either as a result of product flow through the pump and thus directly through the bearing or through a bypass from the main flow. Various types of external coolers, strainers, orifices, booster pumps, etc., may be used to treat the product before injection into the bearings.

The sleeve bearing design is very sensitive to solids content. Various options may be employed in lieu of direct or treated product flush. Where abrasive solids cannot be avoided or for extremely abrasive services, two commonly used solutions are

a) Extremely hard materials, such as ceramics.

b) A supply of clear liquid that isolates the bearings from the pumpage, extending life but at a greater initial cost.

Caution is necessary to ensure that product-lubricated bearings are not allowed to run dry.

1.3.5.6.4 Quality of lubrication (oil flood, ring oil circulating and splash systems)

The most critical elements of lubrication are quality and quantity. Without one the other is significantly affected. Having the proper quantity of poor-quality oil is no better than having an insufficient quantity of high-quality oil.

Having the proper quantity of oil may even be more important than maintaining the quality of the oil. Oil sump lubrication does not require that a specific level be maintained for proper bearing load, only that oil levels do not reach critically low or high points.

1.3.5.6.4.1 Low level (oil sump)

In a low-level operating condition, the bearing will not receive enough lubricant necessary for proper film strength – a precursor to surface contact, skidding, and possible catastrophic failure. As the temperature of the bearing increases, the ball and race both expand, which creates an even tighter fit. This increases the temperature even more, and the cycle continues to a rapid, catastrophic failure.

1.3.5.6.4.2 High level

In a high-level operating condition, churning of the lubricant will occur, accelerating the oxidation rate due to excessive air and elevated temperatures. Too much oil can affect the operation of flinger rings, slingers, and direct bearing contact.

1.3.5.6.4.3 Quality of oil

Degradation and contamination affect the quality of lubrication. Although contamination is widely recognized for its effect on the quality of oil, degradation can be just as damaging to equipment. The leading causes of contamination are particulate, moisture, incompatible fluids, and air entrainment. The leading causes of degradation are oxidation, heat, and use.

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1.3.5.6.4.4 Particle contamination

Particle contamination is the most well-known form of lubricant contamination. This form is considered the cause of wear of component parts, silting, and surface fatigue. A common method used to quantify particulate cleanliness involves the ISO 4406:1999 codes. The standard provides a three-part code to represent the number of particles per milliliter (ml) of fluid greater than or equal to 4 μm, 6 μm, and 14 μm, respectively.

1.3.5.6.4.5 Water contamination

Water contamination of oil can cause several problems relative to oil degradation. Since each type of oil has its own safe level of water before damage can occur, the common practice of measuring parts-per-million (ppm) is not conclusive. There are significant differences between oils with mineral and synthetic bases. Additive packages, referred to as *ad-pacs*, can also make a difference in how much water an oil can hold before phase separation occurs and free water forms. Temperature also plays a major role in how much water oil can hold. By the time water becomes visible, damage is already occurring to both the oil and the surfaces of the equipment and components.

1.3.5.6.4.6 Heat

Elevated operating temperatures are a major contributor of oil oxidation. Combined with air, particulate, and water contamination, the chain reaction of oil oxidation and degradation begins. (See Section 1.3.5.6.5.2.)

1.3.5.6.4.7 Air entrainment

Air entrainment is the primary source of oxygen in the oxidation failure of oil. New oil can contain as much as 10% air at atmospheric pressure. Splash-type bearing housings utilizing flinger rings or slingers are all aeration-prone applications. In addition, air entrainment can lead to accelerated surface corrosion, higher operating temperatures, and oil varnishing.

1.3.5.6.5 Improving quantity in oil flood, ring oil circulating and splash systems

A general guideline is to maintain minimal contact with the lubricating element. Rolling element bearings should not be submerged more than one half the diameter of the rolling element (ball) at the deepest point of submersion in static condition. Flinger rings are more dependent on the shaft speed relative to the depth of submersion. Slinger disks are less susceptible to problems of overlubrication since they are attached directly to the rotating shaft.

One of the most widely used methods of maintaining the proper level lubricant in a bearing housing is the constant level oiler. The constant level oiler replenishes oil lost by leakage through seals, vents, and various connections and plugs in the bearing housing. View ports (bull's-eyes) can also be used to verify proper oil level.

1.3.5.6.5.1 Improving quality - contamination

Housing components, including oilers, seals, and vents, can lead to ingression contamination. Ingression of the housing sump can be reduced by using nonvented oilers and eliminating vent plugs.

Bearing isolators are used to prevent lubricant leakage and contaminant ingress. Labyrinth-type bearing isolators are the most widely used on horizontal pumps. Bearing isolators allow increased pressure created by normal pump operation to vent through the seal and have proven to be very effective at reducing contamination.

Lip seals can also be very good at preventing contamination, although as a contacting-type design, eventual wear to the seal allows for contamination ingression and oil leakage.

Face seals are used to prevent damage to bearings due to contamination and lubricant leakage. Such seals are characterized by the axial loading of the sealing faces provided either by magnetic force or springs.

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1.3.5.6.5.2 Improving quality – degradation

The life of a lubricant is significantly reduced when exposed to high operating temperature conditions. The oxidation rate of oil may double with every 10 or 15 °C (18 or 27 °F) temperature increase. If, by lowering the operating temperature of the oil from, for example, 60 °C (140 °F) to 50 °C (122 °F), a 50% reduction in the rate of oxidation can be realized, thus doubling the effective life of the oil. By this evaluation oil operating at 75 °C (167 °F) will last 100 times longer than at 130 °C (266 °F).

1.3.5.6.5.3 Improving quality – moisture

Pressure differential between the equipment housing and surrounding atmosphere is a leading cause of moisture ingression. When moisture is introduced into the housing, the oil absorbs it at a variable rate depending on temperature, type of oil, and lubricant agitation.

To eliminate water ingression, a closed-system-type constant level oiler should be used and any bearing housing vents plugged. Some seals are not capable of handling the pressures due to equalization and require an expansion chamber.

If moisture/water is a known problem, there are various products commercially available to aid in the removal. Desiccant-type dryers remove moisture over a period of time indicated by a change in color when maximum absorption has occurred. Filtration is another way of removing water from the oil.

Establishing an oil monitoring system is another positive step to achieving proper lubrication. Using condition-monitoring devices are another method. There are numerous commercially available devices that provide moisture/water levels, viscosity levels, and vibration analysis.

1.3.5.7 Type of couplings

The component connecting the driver with the pump is referred to as the *coupling*. The coupling transmits the torque required to turn the pump rotor at speed and can range from rigid to extremely flexible. A few parameters for selection are shaft alignment, torque load, balance requirements, and vibration isolation. In this section are descriptions of the various types of couplings and their typical applications.

The coupling can be selected based on its ability to perform numerous functions. Therefore correct coupling selection requires a review of the application parameters. The primary function of any coupling is to transmit a given torque load from the driver (motor, turbine, engine, gear, or other) to the pump at a given speed (or speeds). The coupling must be sized to transmit all anticipated torque loads including start-up and continuous operation. Additionally the coupling must be sized to fit both the driver shaft and pump shaft.

The need for spacer-type designs and the required shaft separation is to be considered. The inclusion of a spacer-type transmission unit is considered desirable in many instances since (with the correct amount of shaft end separation) the removal of the spacer will allow pump maintenance (including the removal of mechanical seals and bearings) without the need to disturb the driver or pump casing. Most couplings allow some degree of angular misalignment between pump and driveshafts. With the use of a spacer coupling this angular offset translates to a greater degree of lateral offset between shaft ends than can be accommodated with a nonspacer design.

1.3.5.7.1 Offset (flexible or driveshaft)

This type of coupling is more often called a *driveshaft* and has a universal joint at each end plus a slip joint at one end to permit free axial movement. A large-diameter tube connects the two universal joints. The assembly has flanged yokes at each end to connect to the flanged hubs mounted on the driving shaft and the driven shaft. These "couplings" are best used to advantage when the driver and driven units are installed on separate foundations. The maximum parallel shaft offset amount depends on the speed and is obtained from the coupling manufacturer. These couplings are often used with gas or diesel engine drives and may require a damper coupling in addition to the driveshaft.

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Table 1.3.5.7 — Shaft couplings - functions and parameters for selection

Functions of a coupling	Parameters for selection
Transmit required torque at given speed(s)	Torque loads (transient and continuous) Speeds Service factors
Fit to pump and driver shaft	Driver type Pump type Shaft size and design Required hub retention Shaft orientation
Provide shaft end separation to: Permit removal of mechanical seal Permit removal of pump bearings Permit lateral offset of pump and driver shaft	Customer specification Pump design Seal length Bearing length Potential for thermal growth
Permit angular misalignment	Possible degree of alignment accuracy Potential for shaft end movement after initial alignment Potential for thermal growth
Provide end float limitation	Driver type and design
Maintain required degree of unbalance at given speed (speeds)	Customer specification Speed Limits of vibration Precision of manufacture Potential methods of balance correction Lubrication method
Dampen torsional vibrations	Drive system type and arrangement
Prevent the dispersion of stray eddy currents	Driver type and design
Resist corrosive environments	Customer specification Atmosphere
Prevent the occurrence of sparks	Customer specification Area classification

1.3.5.7.2 Limited end float

The most common use of limited end float couplings is with drivers having no axial thrust bearing, such as sleeve bearing motors. Such motors will have rotor end play limits provided on both their outline drawing and on the motor shaft itself. The coupling must be selected to permit the free axial movement of the driver rotor during operation while restricting axial movement within limits established by the driver manufacturer. Coupling manufacturers can supply buttons or disks to restrict driver shaft end float within the necessary limits. This type of coupling is normally applied with a rotodynamic pump that has a rotor fixed in the axial position by a thrust bearing. Should both the pump and driver rotors be free to move axially, the type of coupling must be investigated thoroughly with both the pump and the driver manufacturer. There must be a thrust bearing somewhere in the drivetrain used in combination with a limited end float coupling. This is critical since full axial movement of the motor shaft during operation will result in damage to the motor bearings.

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1.3.5.7.3 Gear

The gear coupling consists of two hubs with external gear teeth and a floating sleeve assembly with internal teeth. The hubs are fitted to the ends of the driver and driven shafts and, as the driving shaft rotates, the driving hub teeth engage the internal teeth so that the entire assembly rotates as a unit. Sliding motion occurs between the teeth, so a supply of lubricant is necessary. A gear coupling wears over time so should be inspected during the pump maintenance schedule. This type of coupling, when properly lubricated, is a long-life product. Gear couplings are used where minimum coupling size is important, where the coupling must perform functions other than power transmission, or where elastomer materials are incompatible with ambient conditions. This type coupling is also applicable for higher-than-usual operating speeds. It is often also applied when a pump has a balance disk and where the pump shaft must move axially without excessive restraint. Note that gear couplings are often chosen for their axial movement capability (end float or slide). One example is when it is necessary to facilitate impeller axial position in compensating for wear.

A thrust button and plate would convert a common gear coupling to a limited end float gear coupling. Limited end float (LEF) couplings are specified for applications that use sleeve bearing motors. The sleeve bearing motor does not have a thrust bearing so thrust is transferred through the coupling to the pump bearing. Because the motor rotor must float a little to accommodate its magnetic center while operational, the coupling has a "limited" axial movement from the end of the shaft to the button. A similar device would be used to prevent end float all together, there would be no initial clearance from the end of the driver shaft to the thrust button. Note that gear couplings are often chosen for their axial movement capability (end float or slide). Nongear couplings also pass on the thrust load to a bearing, but may flex as is common for a disk coupling.

1.3.5.7.4 Disk

This is an all-metal coupling with replaceable flexing element. No lubrication is required, and inspection may be made without disassembly. The coupling normally tolerates a wide temperature range but limits angular misalignment. Some couplings of this type will provide limited end float without modification or float-restricting devices. This type of coupling when properly sized and applied is considered to be an infinite-life coupling. It is used for the highest-speed applications because the highest levels of precision can be guaranteed both in manufacture and assembly. This is particularly important for high-speed operation since residual unbalance levels and the impact on machine vibration can be minimized. Disk couplings provide many of the functional capabilities of a gear coupling without the requirement for lubrication.

1.3.5.7.5 Elastomeric

In this type of coupling, torque is transmitted by compression or shear of a flexible elastomeric element between the driving and driven hubs. No lubrication is required, however, the coupling wears in operation so should be inspected when the pump is maintained. The coupling life, ambient conditions, and range of temperatures are typically restricted by the characteristics of the elastomer. The physical size of couplings in this category may be larger than for all-metal designs. Speed limitations for this type of coupling are typically more stringent than those for metal couplings. Elastomeric couplings are used when damping or resilience is needed in the power transmission system. Such couplings can be a fusible link or a fail-safe design.

1.3.5.7.6 Speed limitations

The maximum speed for a given coupling varies with the type of coupling and the materials from which it is constructed. The limiting speed is always provided by the coupling manufacturer and will be checked by a pump manufacturer who supplies the coupling. Coupling balance may be required to meet the limits for speed.

1.3.5.7.7 Alignment

Irrespective of the type of coupling used, it is important that the two halves be properly aligned to ensure smooth operation. See ANSI/HI 1.4 *Rotodynamic (Centrifugal) Pumps for Installation, Operation, and Maintenance* for the proper procedure for coupling alignment. The coupling manufacturer's allowable misalignment tolerance often

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refers to the amount of misalignment that the couplings may sustain without damage. However, this value may be much greater than the alignment required for smooth operation. Allowable residual misalignment limits should be supplied by the pump manufacturer.

1.3.5.8 Shaft seals

In most rotodynamic pumps the fluid pumped and the bearing lubricant must be sealed around a rotating shaft. In the case of the pumped fluid, this can be accomplished with either a mechanical seal or packing. The pump bearing housing fluid can be sealed with closures, labyrinths, or magnetic face seals. This section describes the various types of rotating shaft seals available and their typical applications. Selection of the correct type and materials is based on service desired, leakage tolerable, liquid properties, and temperature. A qualified seal manufacturer should be consulted for final selection for critical or special services.

1.3.5.8.1 Packed stuffing boxes

For compliance with current environmental and safety regulations, packed stuffing boxes are normally limited to water services since leakage of the fluid to atmosphere is unavoidable and uncontained.

The two most common packed stuffing boxes are those with lantern rings and without lantern rings. The two figures below show these two constructions. For slurry type pumps, please refer to ANSI/HI 12.1–12.6 *Rotodynamic (Centrifugal) Slurry Pump for Nomenclature, Definitions, Applications, and Operations.*

The construction in Figure 1.3.5.8.1a is used where suction pressure is above atmospheric and where pumped fluid is clean-clear and its leakage to atmosphere is acceptable. This arrangement is often used on multistage pumps at the inlet of the second (or higher) stage impeller.

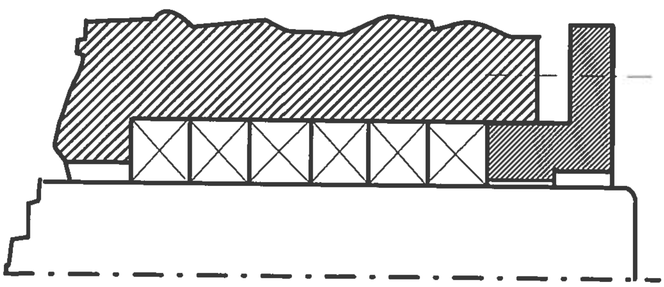

Figure 1.3.5.8.1a — Stuffing box without lantern ring

The type of stuffing box in Figure 1.3.5.8.1b is used where suction pressure is below atmospheric or at a low positive pressure with the possibility that pressure may be below atmospheric. Often a clean-clear pumped liquid is piped from the first-stage discharge to provide the liquid for sealing and lubrication. When the pumped liquid is dirty or over 93 °C (200 °F), an external clean-clear, cool liquid is connected to the tapped opening over the lantern ring of sufficient pressure to provide slight flow into the pump.

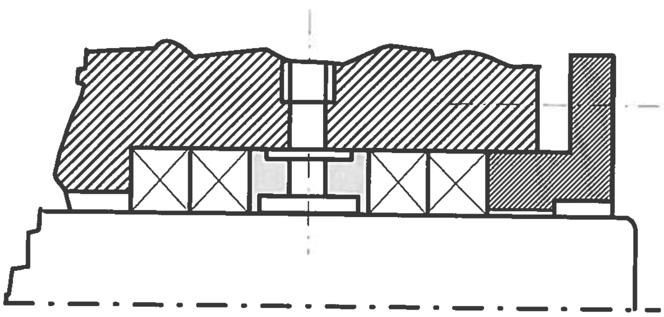

Figure 1.3.5.8.1b — Stuffing box with lantern ring

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

1.3.5.8.2 Mechanical seals

A diagrammatic description of the more popular types of mechanical seals available is shown in Figures 1.3.5.8.2.1a and 1.3.5.8.2.1b. The pump manufacturer will supply mechanical seal details for the specified service conditions. Additional information on mechanical seals can be found it the HI publication, *Mechanical Seals for Pumps: Application Guideline*.

1.3.5.8.2.1 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 be more costly and will require a much more complete disclosure of the liquid handled. (Always give the pump manufacturer complete information on the liquid being handled, including the liquid description, suction and discharge pressure, temperature and condition of the liquids [such as the type of solids contained, if any]. This can normally be done by the completion of the appropriate datasheets that are supplied as part of other ANSI/HI/API/ISO standards.)

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 the 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 (ISO 21049).

The user/specifier must work closely with the pump manufacturer as new pump specifications are developed. Some important points follow:

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

b) The manufacturer can best advise if the specified seal will fit his seal chamber.

c) The manufacturer can best advise if the metallurgy of the seal specified will be compatible with the product pumped. With the many mechanical seal types available, it is recommended that full disclosure of operating conditions be given to the pump manufacturer rather than writing specific seal construction specifications.

1.3.5.8.2.2 Mechanical seal definitions and nomenclature

Back-to-back – One of several dual seal arrangements in which both of the axially flexible assemblies are mounted such that the sealing faces are located at each end. Back-to-back seals can be separate assemblies or one assembly utilizing common spring-loading elements.

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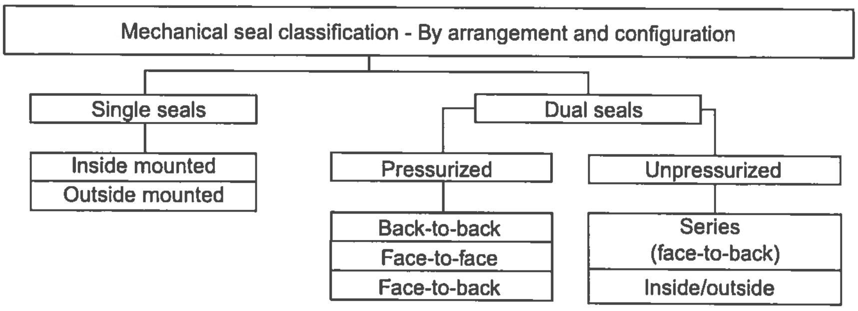

Figure 1.3.5.8.2.1a — Mechanical seal classification by arrangement

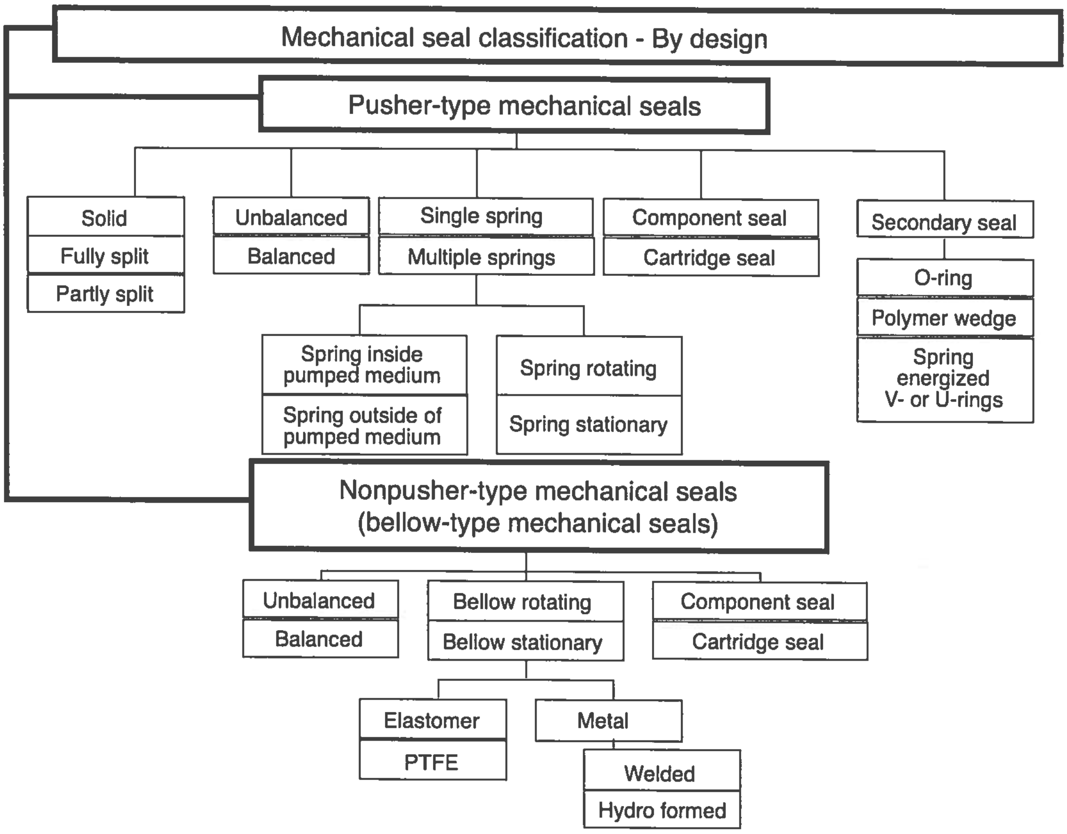

Figure 1.3.5.8.2.1b — Mechanical seal classification by design

Balance ratio – The ratio of the hydraulic closing area to the hydraulic seal face area, typically expressed as a percentage.

Balanced seal – A mechanical seal configuration in which the fluid closing forces have been modified through seal design so that the balance ratio is less than 1, or 100%.

Contacting wet seal – Seal design where the mating faces are lubricated by a liquid and are not designed to intentionally create a hydrodynamic force to sustain a specific face separation gap.

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Containment seal – A seal assembly located on the outboard or atmospheric end of a dual unpressurized seal assembly that is used to contain leakage in the event of a primary seal failure. Typically, the containment seal is designed to run dry during normal pump operation.

Dual pressurized – Dual seal arrangement that has a secondary fluid, termed a *barrier fluid*, at a pressure greater than the product pressure in the pump seal chamber. Dual pressurized seals were previously called *double seals.*

Dual unpressurized – Dual seal arrangement that has a secondary fluid, termed a *buffer fluid*, at a pressure lower than product pressure in the pump seal chamber, typically atmospheric. Dual unpressurized seals were previously called *tandem seals.*

Dynamic seal – A combination of expelling vanes on the back shroud of the impeller and separate expellers located in chambers behind the impeller that generates a pressure to offset the pump casing pressure so that the pump operates without leakage.

Face-to-back – One of several dual seal arrangements in which one mating ring is mounted between two flexible elements and one flexible element is mounted between two mating rings.

Face-to-face – One of several dual seal arrangements in which a single mating ring or both mating rings are mounted between the flexible elements.

Flexible rotor – The portion of a mechanical seal that contains a spring element or elements that rotates with the shaft and moves axially relative to the pump shaft.

Flexible stator – The portion of a mechanical seal that contains a spring element or elements, which is stationary relative to the shaft and moves axially relative to the pump shaft.

Mating ring – A disk or ring-shaped member, mounted on either a shaft or in a housing, such that it does not move axially relative to the shaft or housing and which provides the mating seal face for the flexible element sealing face.

Mechanical seal – A device that prevents the leakage of fluids along rotating shafts. Sealing is accomplished by a primary seal face bearing against the face of a mating ring in conjunction with static secondary seals. The sealing faces are perpendicular to the shaft axis. Axial mechanical force and fluid pressure maintain the contact between the wearing seal faces.

Nonpusher – A type of mechanical seal where the secondary seal is an elastomeric or metal bellows in which part of the secondary seal is fixed to, and does not have to contact, the shaft to compensate for shaft movement or seal face wear.

Pusher – A type of mechanical seal in which the secondary seal is pushed by mechanical or hydraulic means along the shaft to compensate for shaft movement or seal face wear.

Quench gland – A plate providing an enclosed space on the atmospheric side of a mechanical seal or packing to which a neutral fluid is introduced to control the environment. Typically used to provide cooling or prevent coking, crystallization, or icing of the product sealed.

Secondary seal – A device, such as an O-ring, elastomeric or metal bellows, that prevents leakage around the primary sealing faces of a mechanical seal. The term *secondary seal* also refers to static seals, such as O-rings or gaskets, used in ancillary components to prevent leakage from a high-pressure area to a low-pressure area.

Single seal – A seal configuration with only one mechanical seal per seal chamber.

Throat bushing – A device that forms a restrictive close clearance around the shaft, or shaft sleeve, between the seal or packing, and the impeller.

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Unbalanced seal – A mechanical seal configuration in which the fluid closing forces have been modified through seal design so that the balance ratio is greater than 1, or 100%.

1.3.5.8.3 Seal chambers

The standard seal chamber with a throat bushing is also called a *small-bore seal chamber* when mechanical seals are supplied instead of packing. This design can accept either packing or mechanical seals.

There are specific designs for mechanical seals. One design is an enlarged version of the seal chamber and is typically called a *large cylindrical bore seal chamber*. This design has variations such as no throat bushing, a reduced diameter towards the impeller to mount the inboard mating ring, partial throat bushings, and cooling jackets.

In the large bore seal chamber designs there is also a tapered bore design expanding outward towards the impeller side of the chamber. This design has no restrictions like the cylindrical bore seal chamber. The main advantage of this chamber is that it is self-venting. Some tapered bore chambers have flow modifiers of various designs that improve the product flow pattern in the seal chamber and enhance the seals environment when solids or vapors are present in the product being pumped.

The above designs are covered in ASME B73.1, ASME B73.2, and ISO 3069 (Type C seal chambers) standards.

1.3.5.8.4 Other sealing means

Alternatives to mechanical seals or packed seal chambers are available. Limited-leakage labyrinths or bushings are used for some large high-speed boiler feed pumps. Dynamic seals use an auxiliary impeller (also known as an *expeller*) in the seal chamber area to prevent leakage in severe corrosive or abrasive sealing applications in the process industries. In most applications these dynamic seals eliminate the need for seal water and therefore eliminate dilution of the product being pumped. Dynamic seals can be considered whenever a seal water supply is unavailable or normal seal chamber leakage of the pumpage is unacceptable. Dynamic seals need to have a backup or a static sealing device when the pump is not running. The additional sealing device must seal statically when the pump is shut down and it usually runs dry during operation unless a separate flush is provided. Seal chamber maintenance is also greatly reduced with this type of seal.

1.3.5.8.5 Sealless pumps

There are also alternative pump configurations designed to contain liquids without the need for any type of dynamic shaft sealing or leak-off device. These pump configurations are collectively known as sealless pumps. Additional information on *sealless pumps* can be found in ANSI/HI 5.1-5.6 *Sealless Pumps for Nomenclature, Definitions, Application, Operation and Test.*

1.3.5.8.6 Bearing housing sealing

1.3.5.8.6.1 Definition and function

Bearing housing shaft seals create a seal between the rotating shaft and the stationary bearing housing. The function of the bearing housing shaft seal is to both retain lubricant in and exclude contaminants from the bearing housing. Proper selection and application of bearing housing shaft seals has a significant impact on the reliability of the bearing housing and thus the reliability of the entire pump assembly.

Bearing housing shaft seals are divided into the two primary categories of contact and noncontact seals. Contact seals may be further subcategorized as either lip seals or face seals. Noncontact seals may be further subcategorized as gap seals, labyrinth seals, or bearing isolators. Bearing isolators and lip seals are the most commonly used seals in pump bearing housings. The following chart illustrates the various types of commonly available bearing housing shaft seals.

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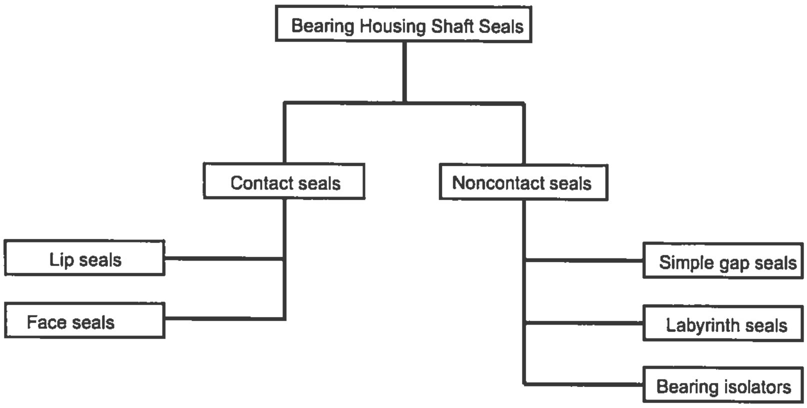

Figure 1.3.5.8.6.1 — Types of bearing housing shaft seals

1.3.5.8.6.2 Contact seals

Contact seals include all designs that have dynamic contact as a requirement for proper function. Contact seals are mostly recommended for applications where the seal must retain or exclude a static level or pressure differential, such as a horizontal bearing housing with a lubricant level above the shaft seal surface.

Contact seals have speed and life limitations and are power-consuming devices. Contact seals are not recommended for use unless retention or exclusion of a static head is required. The most common contact-type seals used in pump bearing housings are lip seals.

1.3.5.8.6.3 Lip seals

Lip seals are characterized by a static, spring-loaded elastomeric element that makes positive, dynamic contact with the shaft. Lip seal designs often include a secondary dirt exclusion lip and are sometimes used in conjunction with a separate, secondary elastomeric V-ring to augment contaminant exclusion. Life expectancy of lip seals is typically 3000 hours or less, depending on many operational factors.

Lip seals manufactured from Buna-N (Nitrile) have a maximum service temperature of 120 °C (250 °F), while lip seals of fluoroelastomers have limits closer to 200 °C (392 °F).

Lip seals are directional and must therefore be installed in the proper orientation. All lip seals have specific shaft surface finish and hardness requirements. Consultation with the lip seal manufacturer is necessary to ensure these requirements are met.

1.3.5.8.6.4 Face seals

Face seals are characterized by flat stationary and rotating faces axially loaded by either magnetic or spring force. Face seals are manufactured in a wide variety of configurations. The magnets or springs may be located in various positions and the faces may be manufactured from different materials. Their primary benefit over lip seals is a typically longer life expectancy, which may offset their higher initial cost.

1.3.5.8.6.5 Noncontact seals

Noncontact seals consist of all seal designs that lack dynamic contact as a requirement for proper function. Noncontact seals, as opposed to contact seals, typically have no speed or life limitations and consume no power. Noncontact seals, however, are generally not suitable for sealing against static heads or pressure differentials.

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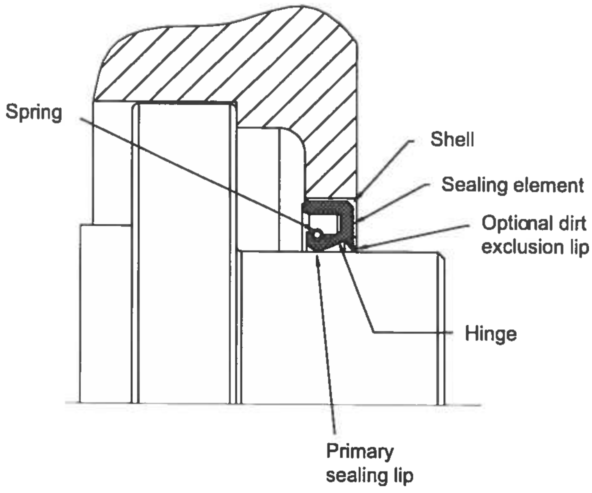

Figure 1.3.5.8.6.3a — Lip seal

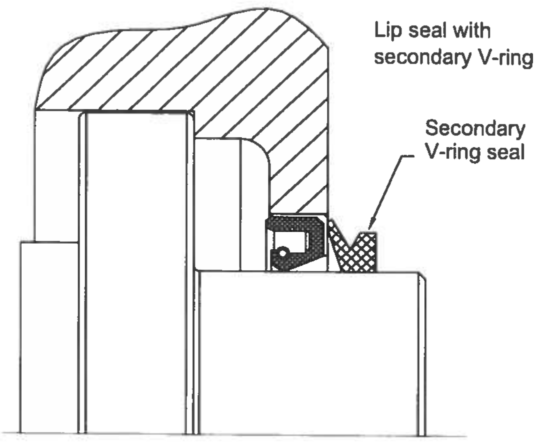

Figure 1.3.5.8.6.3b — Lip seal with secondary V-ring

1.3.5.8.6.6 Simple gap seals

The most basic of all noncontact seals is the simple gap seal. They consist of a singular stationary gap about the shaft and provide a barrier by consequence of a tight running clearance.

1.3.5.8.6.7 Labyrinth seals

A labyrinth is a device that relies on the close clearance path for the lubricant as it tries to leak out of the bearing housing. A return passage drains lubricant back to the housing.

1.3.5.8.6.8 Bearing isolators

Bearing isolators are composed of both a stationary and a rotating component that act in concert to retain lubricant and exclude contaminants from the bearing housing. Since bearing isolators have an infinite life and do not consume power, they are usually preferred over contact seals unless there is a specific requirement to retain a static level, head, or pressure differential.

Bearing isolators typically use a labyrinth pattern in the stator to retain lubricant in the 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 out through an expulsion port by a combination of centrifugal force and gravity drain.

1.3.5.8.6.9 Lubricant return

Bearing housings, regardless of seal type and design, should provide for adequate lubricant drain in the area between the seal and the bearing. The drain should be constructed so that the lubricant level between the bearing and seal shall not be substantially higher than the normal operational lubricant sump level. A machined, rather than integral cast, drain to sump is recommended.

1.3.6 Performance, selection criteria

1.3.6.1 Determination of operating duty point

A pumping system is comprised of the piping and fittings, valves, vessels, flow measuring equipment, and any other conduits through which the liquid is flowing. For a successful pump application, the pump, the system components, and the system must be properly matched to each other.

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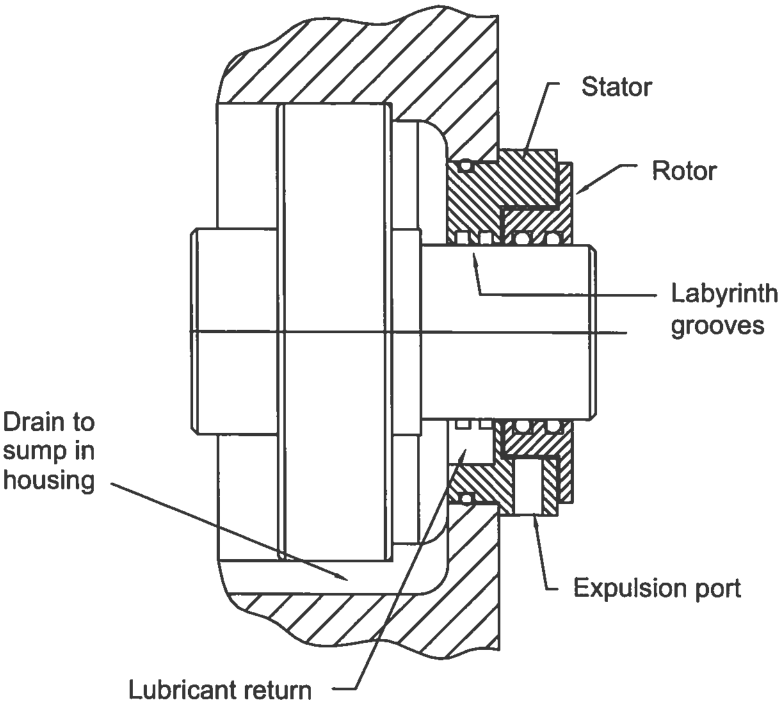

Figure 1.3.5.8.6.8 — Bearing isolator

The requirements for, and the characteristics of, the system must be determined before the pump can be selected. Modifications to the system may be needed for compatibility of all components. Consideration must be given to initial and future operating conditions. Some important features of the system are given in the following paragraphs.

1.3.6.1.1 Pump performance curve versus system curve

A typical simple system and pump curve is shown in Figure 1.3.6.1.1a. It will be noted the pump always operates at the intersection point of its head versus rate-of-flow curve with the system curve.

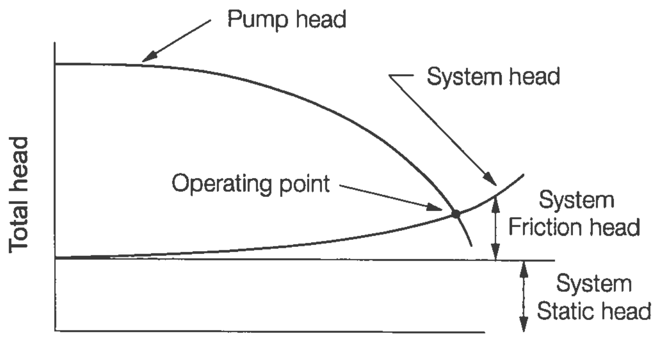

Figure 1.3.6.1.1a — Pump performance curve versus system curve

With more complex systems, the static head varies as the suction and discharge liquid levels, or pressure, change. Friction head is affected by changes in the piping, valve opening, or pipe condition.

Estimates of system friction losses are not normally precise because of the variability of pipe wall conditions and fitting resistances. Therefore, one should look at a range of friction losses when estimating system curves. However, the methodology and accuracy of calculating the system curve is beyond the scope of this standard.

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Similarly, the pump characteristics change if the pumps are operated at variable speed, if there is a lack of adequate suction pressure (NPSHA), or if several pumps are operated simultaneously (see Section 1.3.6.3).

All these changes 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 complete pump operating range.

In applications that require flow or pressure control, the use of variable-speed drive systems can provide excellent system control and much lower life-cycle cost. The variable-speed control system can eliminate costly and inefficient valve or bypass systems. Reducing the pump speed can save significant amounts of energy on variable-demand-type systems, particularly those with high system friction losses. The reduction in pump speed can also increase pump reliability and reduce maintenance costs. A detailed description of variable-speed pumping systems is provided by the Europump and Hydraulic Institute Guideline *Variable Speed Pumping - A Guide to Successful Applications.*

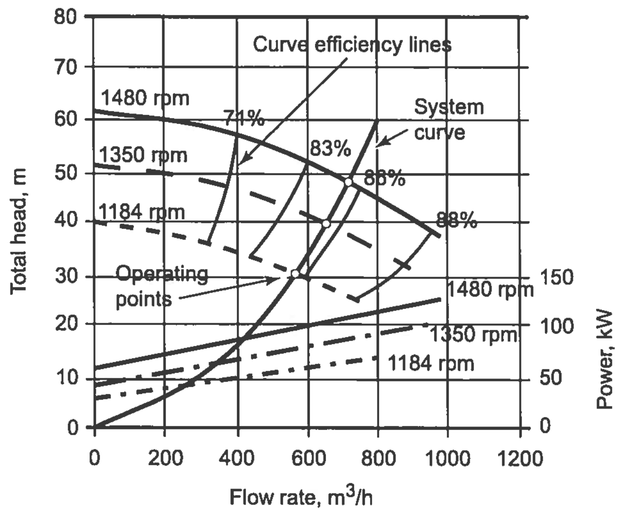

Figure 1.3.6.1.1b — Variable speed curve

1.3.6.1.2 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 1.3.6.1.2a.

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 1.3.6.1.2b).

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

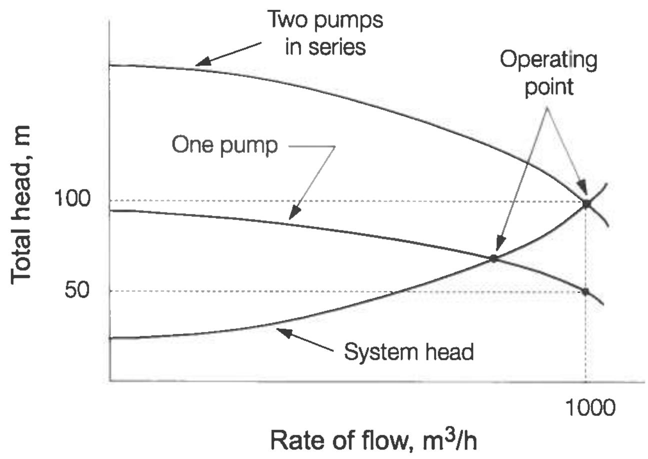

Figure 1.3.6.1.2a — Pumps operating in series

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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.

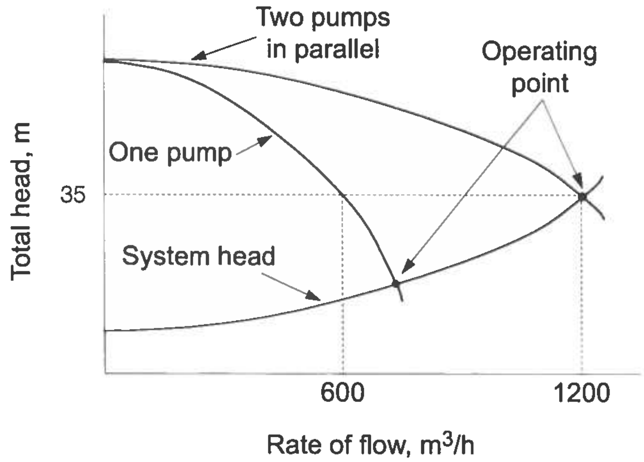

Figure 1.3.6.1.2b — Pumps operating in parallel

1.3.6.1.3 Continuous, intermittent, and cyclic service

Continuous service is often defined as operation for at least an eight-hour shift without stopping. In many cases, the pump operates 24 hours per day for prolonged periods.

Intermittent or cyclic service is often controlled by automatic controls; liquid level or pressure switches are the most common. The pump is started by some controller and, when a liquid level or pressure condition is satisfied, the pump is stopped. Precautions may be needed to avoid too many starts of an electric motor and prevent its overheating.

Pump construction is relatively insensitive to any of the services described above. As the pump increases in size for horsepower required, so does the need for full disclosure of operating conditions to the pump manufacturer. See ANSI/HI 1.4 for operating recommendations.

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

A characteristic of rotodynamic (vertical and centrifugal) pumps is that it is possible to determine the change in rate of flow, head, and power at any point on the pump characteristic curves by calculation when there is a change in the speed of rotation (frequency). 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, in rpm

Q_2 = rate of flow at desired speed, in m^3/h (gpm)

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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.

For combined motor pump units (OH8 and OH8A) or when reference is made to frequency instead of speed of rotation, the rate of flow, pump total head, and power input data are subject to the above-mentioned affinity rules, provided that n_2 is replaced by the frequency f_2 and n_1 by the frequency f_1.

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

The same affinity rules apply, within limits, 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

Normally the D diameter is the largest outside diameter (front shroud) of the impeller vanes (see Figure 1.3.6.1.4a).

However if the impeller exit is machined at an angle, then the D shall be the root mean squared (RMS) average value between the diameters (see Figure 1.3.6.1.4b). The RMS may be calculated as follows:

$$D = \sqrt{\frac{D_F^2 + D_B^2}{2}}$$

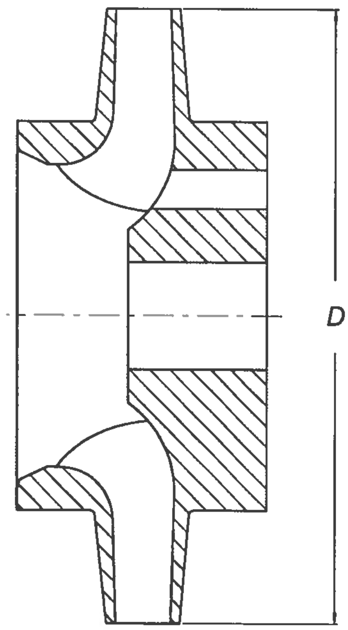

Figure 1.3.6.1.4a — Impeller with straight outside diameter

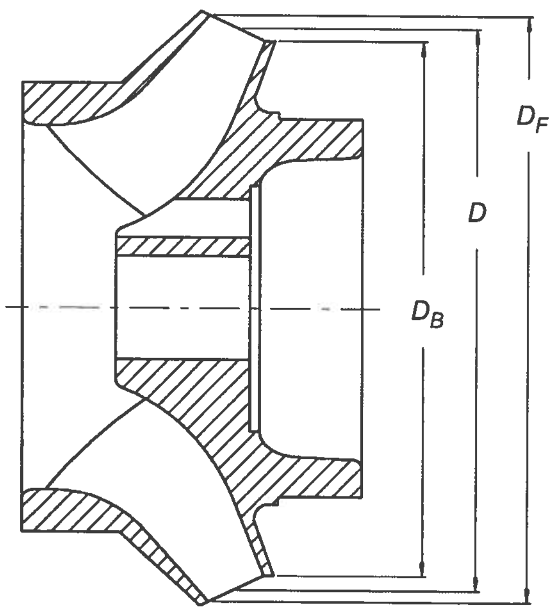

Figure 1.3.6.1.4b — Impeller with angled outside diameter

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

When changing impeller diameters more than 5%, the above equations are not recommended without consulting the pump manufacturer.

The manufacturer must be consulted to determine minimum acceptable impeller diameter.

The factors that limit the extent to which an impeller can be trimmed are:

The resultant efficiency. As an impeller is trimmed the efficiency of the pump will be reduced. Eventually the point will be reached whereby the efficiency reaches an unacceptably low level.

Vane overlap. Trimming the impeller reduces the overlap between impeller vanes. This in turn can cause undesirable flow patterns within the impeller at flows less than the BEP rate of flow. When such flow patterns occur, the result is unexpected changes in the shape of the head flow curve.

Impeller geometry. The location of features on the impeller, such as the wear ring, can limit the extent to which the impeller can be trimmed.

The type of pump casing construction also plays a major role in determining the minimum impeller diameter due to trimming. Typically pumps with a diffuser construction limit the impeller trim to 95% of full diameter. Volute design pumps can often allow impeller trims to 80% of the original impeller diameter. Greater trims may be possible with reference to the factors above.

A variation of normal trimming may be applied to pumps with diffuser construction. In this variation, only the vanes of the impeller are trimmed while the impeller shrouds remain at their original diameter. This trim may be applied to reduce the loss of pump efficiency for a given trim. It may also be employed to limit the possibility of pressure pulsations exciting the volumes of fluid between the impeller shrouds and the pump casing.

NOTE: The system requirements may limit pump performance so that the rate of flow change in the system will not follow the above calculation.

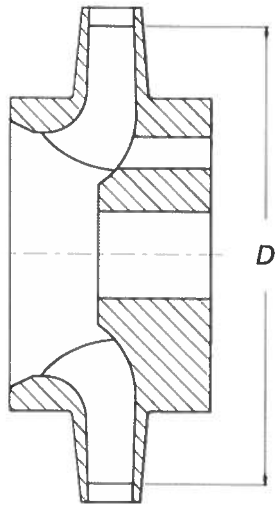

Figure 1.3.6.1.4c — Trimming only the impeller vane

1.3.6.1.5 System pressure limitation

The system must be capable of withstanding the pressures at the operating conditions, as well as at any other conditions that may be reasonably expected. If the system is equipped with a discharge shut-off valve, the piping should be designed for suction pressure plus 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 is discussed in more detail in Section 1.3.6.10.

1.3.6.1.6 Operation away from the best efficiency point

A rotodynamic pump is designed for optimum performance at one specific head and rate of flow for a given speed. This operating point is normally called the *best efficiency point* (BEP).

BEP is the most cost- and energy-efficient point at which to operate the pump.

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However, pumps often do not run continuously at their BEP due to changing system requirements and the difficulty of matching the selected operating conditions with the BEP of the pump. Operation at flow rates above or below the BEP imposes additional strain on some parts and can be damaging to the pump (see Section 1.3.6.2).

1.3.6.2 Minimum flow

Rotodynamic pumps have limitations on the minimum flow at which they should be operated either continuously or for an extended period of time. After starting a pump, do not operate with a closed discharge valve.

Operation of pumps at reduced rates of flow may lead to the following problems:

Temperature buildup: Absorption of the input power into the pumped liquid raises the liquid temperature. In general, temperature rise across the pump should be limited to manufacturer's recommendations (see Section 1.3.6.6).

Excessive radial thrust: Single volute pumps may have limitations in this regard. See Section 1.3.5.4. Double volute and diffuser pumps are usually not limited because of radial thrust.

Suction recirculation: Circulatory flow in the impeller eye can cause localized pitting and mechanical damage. Such flow depends on the impeller inlet design.

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.

Insufficient net positive suction head: In some designs, the NPSH required by the impeller can increase at low flows and noise, impeller pitting, and other symptoms can occur. The pump manufacturer's performance curve should be checked for NPSHR or minimum flow recommendations.

Consult the pump manufacturer for specific minimum flow recommendations or refer to ANSI/HI 9.6.3 *Centrifugal and Vertical Pumps for Allowable Operating Region* for additional information and general guidelines.

1.3.6.3 Net positive suction head

Net positive suction head available (NPSHA) is the total suction head of liquid absolute, determined at the first-stage impeller datum, less the absolute vapor pressure of the liquid:

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

Where:

h_{sa} = Total suction head in absolute = $h_{atm} + h_s$, in m (ft)

h_s = Suction head, in m (ft)

NPSH3 is the amount of suction head, over vapor pressure, required to prevent more than 3% loss in total head from the first stage of the pump at a specific rate of flow. (See Table 1.3.2.2 Principal symbols and Section 1.3.2.2 Preferred units for pump applications).

See ANSI/HI 1.1 and 2.1 *Types and Nomenclature* for further details on the definitions of NPSHA and NPSHR.

1.3.6.3.1 NPSHA corrections for temperature and elevation

NPSHA is a function of the absolute pressure, both static and dynamic, and the liquid vapor pressure. In an open system, the absolute pressure is in turn a function of the atmospheric pressure, static liquid elevation, suction

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piping resistance losses, and the vapor pressure. The correction for elevation is approximately 1 m per 1000 m (1 ft per 1000 ft) of elevation change. The vapor pressure of water at various temperatures can be found in the Hydraulic Institute *Engineering Data Book*.

1.3.6.3.2 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. NPSH3 is defined as the value of NPSHR at which the first-stage total head drops by 3% due to cavitation. See ANSI/HI 14.6 *Rotodynamic Pumps for Hydraulic Performance Acceptance Tests* for details. If the pump is allowed to operate with net positive suction head equal to NPSH3, the pump 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 usually 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.

The purchaser should consider an appropriate NPSH margin in addition to the NPSHR specified. An NPSH margin is the NPSH that exists in excess of the pump's NPSHR. It is usually desirable to have an operating NPSH margin that is sufficient at all flows (from minimum continuous stable flow to maximum expected operating flow) to protect the pump from damage caused by flow recirculation, separation, and cavitation. The vendor should be consulted about recommended NPSH margins for the specific pump type and intended service. In establishing NPSHA, the purchaser and the vendor should recognize the relationship between minimum continuous stable flow and the pump's suction specific speed.

In general, minimum continuous stable flow, expressed as a percentage of flow at the pump's best efficiency point, increases as suction specific speed increases. However, other factors such as the hydraulic design, the pumped liquid, and the NPSH margin, also affect the pump's ability to operate satisfactorily over a wide flow range. Pump design that addresses low-flow operation is an evolving technology, and selection of suction specific speed levels and NPSH margins should take into account current industry and vendor experience.

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

The NPSH requirements of rotodynamic 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 fluids, or water at significantly higher than room temperatures, will operate satisfactorily with less NPSHA than would be required for cold water.

Application considerations for petroleum process pumps

Pumps used for petroleum (hydrocarbon) services can usually survive with relatively small NPSHA margins for several reasons, including the following:

a) Processes are typically steady, with few system upsets (transients) or quick flow change demands.

b) Process requirements are typically well known and demands can be planned and predicted.

c) Most hydrocarbon liquids have relatively low vapor-volume-to-liquid-volume ratios. This means that, if the liquid should vaporize at or near the pump suction (impeller inlet), the volume of the resulting vapor does not choke the impeller inlet passages as severely as does water vapor during cavitation. This results in a smaller drop in developed head for the same NPSH margin.

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d) Less energy is released when hydrocarbon vapor bubbles collapse (velocity from implosion is less), and this means less damage occurs as a result of cavitation. It is, therefore, not as critical that cavitation be avoided, as might be the case with other liquids. Hydrocarbon liquids, especially mixtures of hydrocarbon liquids, because of their relatively low vapor volume, are sometimes associated with a so-called *hydrocarbon correction factor*. This correction factor is applied to the water NPSHR values to correct for the fact that the vapor volume of flashed hydrocarbon liquid is substantially less than that of flashed water and, thus, has the effect of reducing NPSH3. NPSH3 is defined as the amount of NPSH required by the pump at a given rate of flow before cavitation results in a 3% drop in the developed head (first-stage head) of the pump.

   It must be noted that ISO 13709 (API 610) requires that the NPSH required be based on water with no reduction allowed for other liquids. This thereby prohibits application of the hydrocarbon correction factor for API 610 applications. Instead it is required that the vendor shall specify on the data sheets the NPSHR based on water (at a temperature of less than 65 °C [150 °F]) at the rated flow and rated speed. A reduction or correction factor for liquids other than water (such as hydrocarbons) shall not be applied.

This favorable vapor bubble size situation with hydrocarbons should be taken into account when determining the NPSHA margin requirements for petroleum pumps. The margins can be lower than for other applications.

Application considerations – hot water pumps

Power plant pumps are water pumps. Cold water is one of the most difficult liquids to pump in that cavitation can cause severe damage. Unlike hydrocarbon liquids handled by petroleum pumps, water, when it vaporizes (flashes), expands tremendously. This results in higher impact velocities when the vapor bubbles implode, thus higher destructive energy. One pound of water at room temperature occupies 4.5×10^{-4} m^3 (0.016 ft^3), and will flash to over 34 m^3 (1200 ft^3) of vapor. This is a volume ratio of 75,000:1. For typical hydrocarbon liquids, this volume ratio is one half to one tenth that of water.

Hot water, on the other hand, can act similarly to hydrocarbon liquids. When water is heated to 120 – 150 °C (250 – 300 °F), the vapor volume characteristics become similar to that of a typical hydrocarbon. This means that the effects of flashing are diminished; however, the opportunities for system transients increase significantly with temperature.

Procedures for determining the effect of viscosity on NPSHR are also provided in ANSI/HI 9.6.7 *American National Standard (Guideline) for Effects of Liquid Viscosity on Rotodynamic (Centrifugal and Vertical) Pump Performance*.

Net positive suction head required (NPSHR) is influenced by the pressure distribution near the leading edge of impeller blades. The pressure distribution depends on both the Reynolds number and hydraulic losses between the pump suction flange and impeller inlet. These losses increase with viscosity and affect NPSHR. Other factors that influence NPSHR are liquid thermodynamic properties and the presence of entrained or dissolved gas. The interaction of these factors is discussed in ANSI/HI 9.6.7. A method of estimating the NPSHR on viscous liquids based on analytical considerations is also outlined in ANSI/HI 9.6.7. This generalized method should not be applied to hydrocarbons without consideration of thermal and elevation effects on the liquid properties. See Section 1.3.6.3.1.

Figures 1.3.6.3.5a and 1.3.6.3.5b are composite charts of NPSH reductions that may be expected for hydrocarbon liquids and high-temperature water, based on available laboratory data from tests conducted on the fluids shown, plotted as a function of fluid temperature and true vapor pressure at that temperature.

1.3.6.3.4 Limitations for use of chart for net positive suction head reduction (Figures 1.3.6.3.5a and 1.3.6.3.5b)

Until specific experience has been gained with operation of pumps under conditions where Figures 1.3.6.3.5a and 1.3.6.3.5b apply, NPSH reduction should be limited to 50% of the NPSH required by the pump for cold water.

Figures 1.3.6.3.5a and 1.3.6.3.5b are based on pumps handling pure liquids. When entrained air or other 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 air or other dissolved gases are present,

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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.

When using Figures 1.3.6.3.5a and 1.3.6.3.5b 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.

1.3.6.3.5 Instruction for using chart for net positive suction head reduction (Figures 1.3.6.3.5a and 1.3.6.3.5b)

Enter Figures 1.3.6.3.5a and 1.3.6.3.5b 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 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.

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.

1.3.6.3.6 Use of chart for net positive suction head reduction (Figures 1.3.6.3.5a and 1.3.6.3.5b) for liquids other than hydrocarbons or water

The consistency of results that have been obtained on tests conducted with both water and hydrocarbon fluids suggests that NPSHR (required) by a rotodynamic pump may be reduced when handling any liquid having relatively high true vapor pressure at pumping temperature. However, since available data are limited to the liquids for which temperature and vapor pressure relationships are shown on Figures 1.3.6.3.5a and 1.3.6.3.5b, application of this chart to liquids other than hydrocarbons and water is not recommended except where it is understood that such usage can be accepted on an experimental basis.

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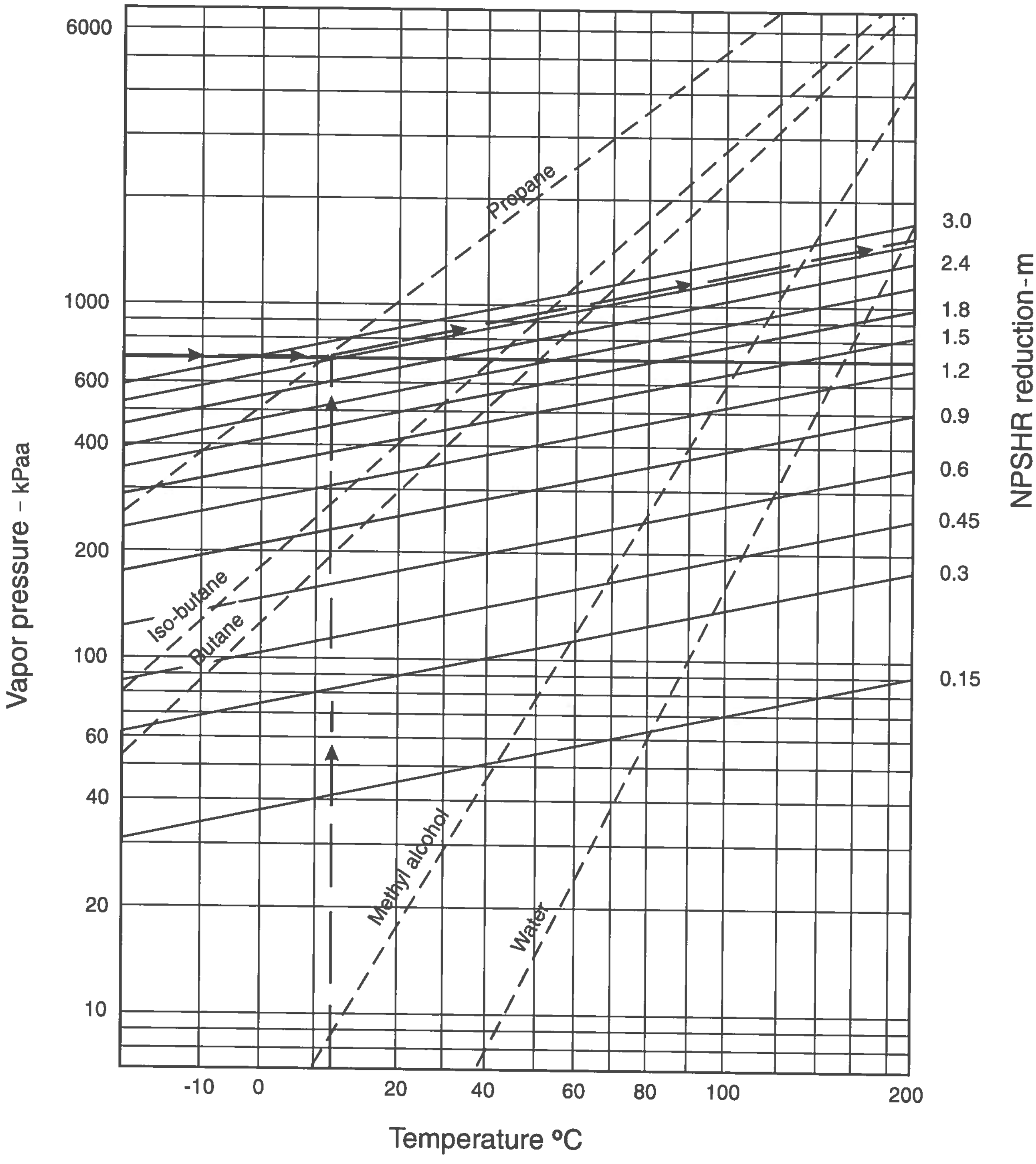

Figure 1.3.6.3.5a — NPSHR reduction for pumps handling hydrocarbon liquids and high-temperature water (metric units)

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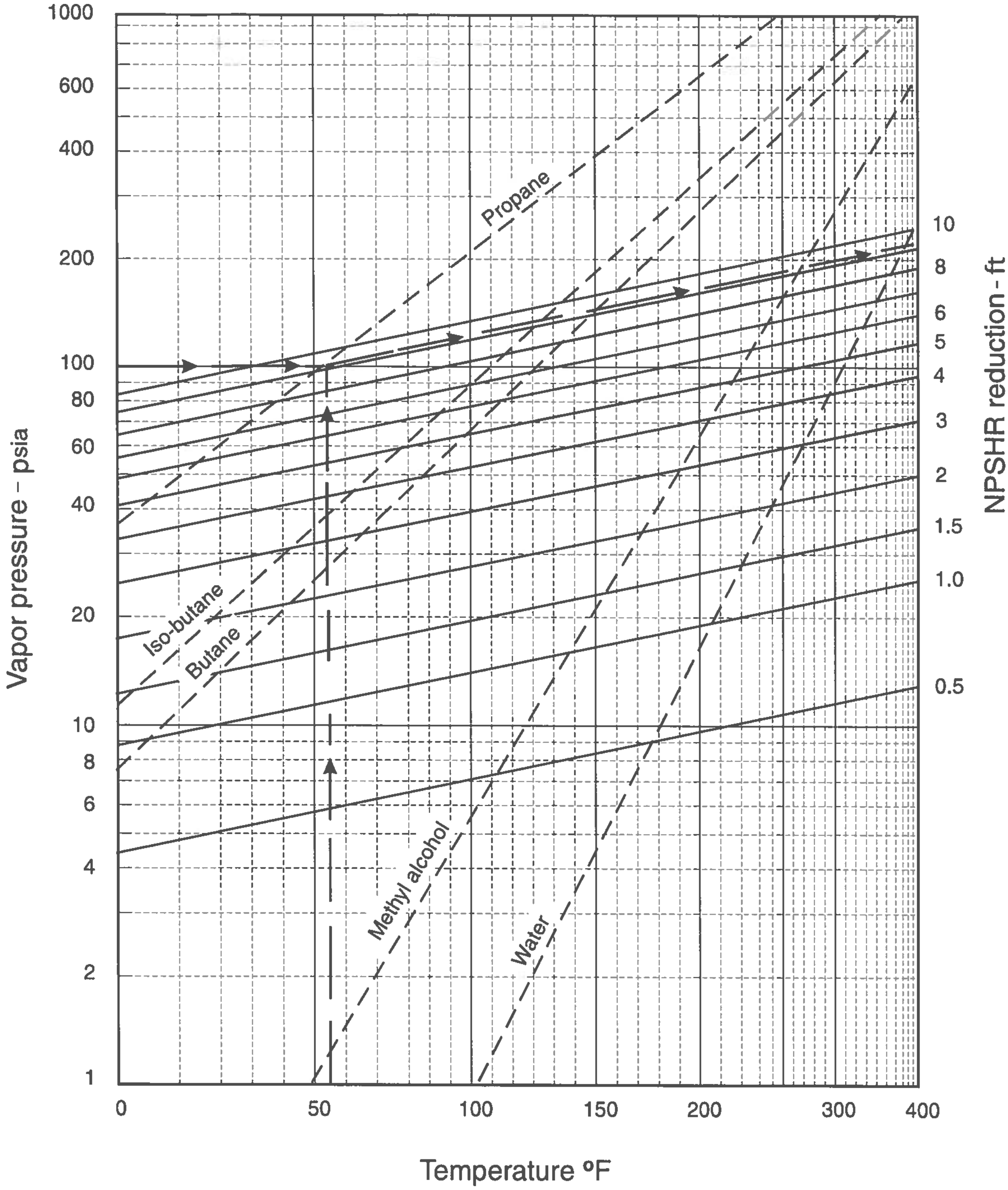

Figure 1.3.6.3.5b — NPSHR reduction for pumps handling hydrocarbon liquids and high-temperature water (US customary units)

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1.3.6.4 Suction performance considerations

Increased pump speeds without proper suction conditions can result in abnormal wear and possible failure from excessive vibration, noise, and cavitation damage.

1.3.6.5 Liquid temperature rise in a rotodynamic pump

1.3.6.5.1 Temperature rise calculation

Pump efficiency is the ratio of the energy imparted to the liquid by the pump (P_w) and the energy delivered to the pump shaft (P_p) expressed in percent. The difference between P_w and P_p represents the power losses within the pump itself, due to internal recirculation, friction, bearings, mechanical seal, etc. Except for small losses in the bearings, the power losses are converted into thermal energy (heat) and transferred to the liquid passing through the pump.

A convenient equation relates temperature rise to the total head and pump efficiency:

(Metric units) $$\Delta t = \frac{H}{102 \times C_p}\left(\frac{1}{\eta} - 1\right)$$

(US customary units) $$\Delta t = \frac{H}{778 \times C_p}\left(\frac{1}{\eta} - 1\right)$$

Where:

Δt = temperature rise through the pump, in °C (°F)

H = total developed head at flow being considered, in m (ft)

778 = constant

102 = constant

C_p = specific heat of the liquid at pumping temperature, in kJ/(kg K) [Btu/(lbm °F)]

η = efficiency of the pump at flow being considered, expressed as a decimal

Temperature rise through a pump increases as flow is decreased. When a pump is run at or near shutoff, the majority of the power input is converted to thermal energy, causing a rapid temperature rise.

For numerous and compelling reasons it is extremely important that heat rise within the pump be controlled and mitigated at all times.

All commonly handled fluids demonstrate predictable vapor pressure versus temperature properties.

If the pressure within the pump remains constant and yet temperature is allowed to increase, then the threshold at which the liquid changes to vapor will be crossed. Different materials used for the construction of various pump parts will expand at different rates. Stationary and rotating components may thereby become loose and tend to dislodge or may become tight and begin to rub and bind. In separately coupled pump applications, with increasing temperature the pump may expand while the driver remains in position causing coupling misalignment, excessive runout, and potential mechanical seal or bearing failure. Excessive heat buildup in the pump can migrate through the shaft and frame and exceed the operating limits of mechanical seals, bearings, or lubrication systems.

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All of these effects will potentially result in rapid seizure of the rotating parts, complete catastrophic pump failure, or destruction of the pump and associated equipment.

When working with liquids at elevated temperatures, proper safety measures must be in place.

An equation expressing the rate of temperature rise at shutoff follows:

(Metric units) $$T_{rso} = \frac{60 \times P_{pso}}{V \times C_p \times \rho}$$

(US customary units) $$T_{rso} = \frac{5.09 \times P_{pso}}{V \times C_p \times s}$$

Where:

T_{rso} = rate of temperature rise evaluated at shutoff, in °C/min (°F/min)

60 = constant for unit conversion

5.09 = constant for unit conversion

P_{pso} = input power at shutoff for the liquid pumped, in kW (hp)

V = casing internal volume, in m^3 (gal)

C_p = specific heat of the liquid at pumping temperature, in kJ/(kg K) [Btu/(lbm °F)]. For these calculations, a constant value is used.

ρ = density, in kg/m^3

s = specific gravity

This equation neglects, as does the first equation, the small effects of pump bearings, heat dissipation, and compressibility of the liquid, but is applicable to situations normally encountered.

NOTE: The liquid will first vaporize at the eye of the impeller where the local static pressure is the lowest.

1.3.6.5.2 Minimum flow in a pump due to temperature rise

This discussion focuses on minimum flow required through a pump to prevent excess temperature rise.

The methods presented in Sections 1.3.6.5.1 and 1.3.6.5.2 neglect the small effects of pump bearings, heat dissipation, and compressibility of the liquid, but are applicable to situations normally encountered. Also neglected is the effect of any volume of liquid between the pump discharge flange and the block valve.

A commonly accepted practice limits temperature rise through a pump to 8 °C (15 °F). For most installations, this is adequate and minimum flow may be calculated with equations. For 8 °C (15 °F) temperature rise through a pump:

(Metric units) $$Q = \frac{433 P_p}{C_p \times \rho}$$

(US customary units) $$Q = \frac{P_p}{2.95 \times C_p \times s}$$

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Where:

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

P_p = input power at the minimum flow, in kW (hp)

2.95 = constant

433 = constant

C_p = specific heat of the liquid at pumping temperature, in kJ/(kg K) [Btu/(lbm °F)]

ρ = density, in kg/m^3

s = specific gravity

At the minimum flows calculated using the above equation, the power input is approximately the same as at shutoff.

Temperature rise in excess of 8 °C (15 °F) may be permissible in certain circumstances. The pump manufacturer should be contacted if a temperature rise in excess of 8 °C (15 °F) is desired.

As previously discussed, catastrophic failure of the pump and associated equipment may result if the liquid within the pump casing is allowed to vaporize. To prevent flashing, a flow must be maintained through the pump that will keep the liquid below its saturation temperature.

Special situations occur when a relatively small margin exists between NPSHA and NPSHR by the pump, as in boiler feed services. A boiler feed pump normally takes suction from a deaerator or deaerating heater. A deaerator is a closed vessel in which the feedwater is heated by direct contact with steam to remove air, which could cause corrosion in the boiler. In a properly operated deaerator, the water is heated to boiling (i.e., vapor pressure is equal to the deaerator pressure).

To establish the maximum allowable temperature rise, first determine the:

a) Absolute pressure in the deaerator or other suction source.

b) Additional absolute pressure available at the pump suction nozzle above that required by the pump, by using:

(Metric units) $$P = \frac{(NPSHA - NPSHR) \times \rho \times g}{102}$$

(US customary units) $$P = \frac{(NPSHA - NPSHR) \times s}{2.31}$$

c) Sum of the values determined in a) and b).

The maximum allowable temperature rise is the difference between the saturation temperature corresponding to the pressure determined in c) and the temperature of the suction source.

C_p = specific heat of the liquid at pumping temperature, in kJ/(kg K) [Btu/(lbm °F)]

ρ = density, in kg/m^3

s = specific gravity

In multistage pumps, the allowable temperature rise corresponding to the difference between the NPSHA and NPSHR should be considered to occur in the first stage. Temperature rise through the entire pump typically should still be limited to 8 °C (15 °F).

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In the overall assessment of pump application, it must be recognized that temperature rise should not be the only consideration when determining minimum flow. Many pumps (in particular those of higher horsepower) will suffer an increase in vibration and decrease in reliability if operated at higher rates of flow above that limited by temperature rise. Refer to ANSI/HI 9.6.3 *Rotodynamic (Centrifugal and Vertical) Pumps – Guideline for Allowable Operating Region* for further details.

Minimum flow is guaranteed by installing a bypass from the discharge line to some low-pressure point in the system. The bypass must be back to a source that is capable of absorbing the total heat being generated during this process (over the total time of bypassing) without upsetting or damaging the system. The bypass should not lead directly back to the pump suction.

An orifice installed in the bypass line breaks down the differential pressure between the pump discharge and the low pressure point in the system.

The bypass may be manually or automatically operated but must be open during periods of light load or when starting or stopping the pump.

1.3.6.6 Efficiency prediction method for rotodynamic pumps

The major influences on rotodynamic pump efficiency are specific speed (n_S, N_S), pump size, NPSHA (or NPSHR), and the type of pump selected to meet the service conditions.

See the HI 20.3 *Efficiency Prediction Guideline* for an in-depth description of the recommended procedures for estimating these effects.

1.3.6.7 Effects of handling viscous liquids

The performance of rotodynamic pumps is affected when handling viscous liquids. A marked increase in brake horsepower, a reduction in head, and some reduction in rate of flow occur with moderate and high viscosities.

See ANSI/HI 9.6.7 *Effects of Liquid Viscosity on Rotodynamic (Centrifugal and Vertical) Pump Performance* for an in-depth description of the recommended procedures for estimating these effects.

1.3.6.8 Startup and shutdown

During startup, the pump torque and the driver torque vary as the pump is accelerating. The driver must be capable of supplying more torque at each speed than required by the pump. Following proper procedures reduces pump torque requirements and assists the driver during the starting process.

Users and operators must be made aware that for some rotodynamic centrifugal pump configurations, such as between-bearings multistage, and with some bearing lubrication systems, such as ring oil lubrication, it is possible that operation at a speed lower than the manufacturer's minimum allowable operating speed could cause serious damage. The user must fully investigate the suitability of the pump for operation at low speeds prior to such operation.

When designing the start-up system, due consideration must be made to the time required to accelerate the pump up to its rated speed. Normally the pump must be started and brought immediately up to operating speed (within 3 to 5 seconds). Operation below the specified minimum allowable operating speed must be avoided since this may potentially cause damage to the pump. The manufacturer should be consulted for situations requiring longer ramp-up times, "slow-rolling," or low-speed operation of the pump. Consideration must be taken during a slow start up or shutdown to avoid resonance and critical speeds that can cause high vibration in both the pump and mounting structures.

Refer to ANSI/HI 1.4 and 2.4 for further guidance.

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1.3.6.8.1 Starting with closed discharge valve

A low N_s rotodynamic pump that is primed and operating at full speed requires much less power when the discharge valve is nearly closed. For this reason, it may be advantageous to have the discharge valve nearly closed when the pump is being started. Operating the pump against a closed valve will cause rapid temperature rise and vaporization of the contained liquid (see Section 1.3.6.5). After starting the pump, do not operate with a closed or nearly closed discharge valve. Open the valve at a controlled rate to operate the pump in its allowable operating range. Note that this does not apply to higher N_s mixed flow and axial flow pumps.

1.3.6.8.2 Starting with open discharge valve (mixed flow and axial flow type pumps)

Pumps of the mixed flow type frequently require greater input power with the discharge valve closed than with it open. Axial flow type pumps nearly always require a great deal more power at shutoff than at rated conditions and must be started with the discharge valve open. The manufacturer's instructions should be consulted for the characteristic curve of such pumps.

1.3.6.8.3 Shutdown

Avoidance of water hammer (see Section 1.3.6.10) is a primary concern during the shutdown of a pump, especially in installations with long discharge piping or systems with quick closing valves. Gradually reducing the pump flow prior to shutdown by use of a variable-speed drive of a slow mechanical discharge valve is one way to eliminate or reduce the water hammer. For additional information, refer to ANSI/HI Standard 9.6.6 *Pump Piping for Rotodynamic (Centrifugal) Pumps.*

Operation at shut-off conditions (with the discharge valve closed) for extended periods is not recommended. This induces increased radial thrust in many pumps and overheats the liquid in the pump. Recirculation piping (back to the suction source) or reducing speed using a variable-speed controller will help increase pump and seal life, as recommended by the pump manufacturer.

See Section 1.3.6.2 regarding minimum flow.

1.3.6.9 Reverse runaway speed

A sudden power and check valve failure during pump operation against a static head will result in reverse pump rotation. If the pump is driven by a prime mover offering little resistance while running backwards, then the reverse speed may approach its maximum consistent with zero torque. This speed is called *reverse runaway speed.* Refer to ANSI/HI 1.4 *Rotodynamic (Centrifugal) Pumps for Installation, Operation, and Maintenance.*

1.3.6.10 Water hammer (hydraulic shock)

Water hammer or hydraulic shock is a condition that occurs when a column of water suddenly changes velocity. This condition can exist when the power to the driver is lost suddenly, air is completely expelled from the system piping, a valve is closed too quickly, or a check valve closes too slowly and allows backflow to occur, which causes the check valve to slam shut. When this occurs, the kinetic energy of the liquid is rapidly transformed into pressure energy, which will cause a sharp rise above normal system pressure. This transformation produces an acoustic pressure wave that propagates upstream within the pipe. The resulting peak pressure can be multiple (10 or more) times higher than normal working pressure. Failure of pressure-containing components will be a result if this is not mitigated by some protection device.

For additional information, refer to the ANSI/HI 9.6.6. 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.

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1.3.6.11 Pump liquid temperature limits on end suction pumps

Limits are placed on pumped liquid temperatures because of heat that travels from the pump casing, through the shaft, motor adaptor, or bearing frame. This heat raises the bearing lubricant temperature and can adversely affect internal bearing clearances due to differential expansion. For pumps using close-coupled motors, the motor winding temperature will increase.

Bearing lubricant temperatures above 80 °C (176 °F) can cause the lubricants to oxidize and lose their lubricating ability. The degradation of the lubricant will shorten bearing life. It is possible with special bearings and synthetic lubricants to operate above the 80 °C (176 °F) limit.

It is also possible to control pump lubricant and bearing temperatures by external cooling. This can be accomplished with either a cool liquid passed through a finned tube immersed in the bearing lubricant or through passageways designed into the bearing frame, seal chamber, or stuffing-box cover. The use of fins on the bearing frame exterior, with air blown over the bearing frame by a fan, is also an effective cooling method.

Temperature limits are also imposed by the materials of construction of the pump. For example, cast gray iron is limited to 175 °C (350 °F) due to its mechanical strength, whereas ductile iron has a higher limit of 340 °C (650 °F).

For high temperatures (greater than 175 °C [350 °F]) flexibly coupled arrangements with centerline mounting of the pump casing is beneficial. This eliminates the possibility of thermal growth of the casing (in the vertical plane) and thereby minimizes the impact of thermal growth on pump/driver alignment.

Many factors, including pump liquid temperature, ambient conditions, speed, bearing type, lubrication method, method of sealing, pump design, and cooling methods influence the final bearing lubricant temperature. The guidelines in Tables 1.3.6.11a and 1.3.6.11b are based on general experience and are commonly adopted in the pump industry. For temperatures beyond these limits, consult the pump manufacturer. Deviations can be justified based on special design, testing, and field experience. Flange pressure ratings are reduced at elevated temperatures and the adequacy of the flanges may need to be verified for the desired pressure, material, and temperature (see ANSI/ASME B16.5). Allowable nozzle loads are also reduced at elevated temperatures (see ANSI/HI 9.6.2 *Rotodynamic Pumps for Assessment of Applied Nozzle Loads)*.

1.3.6.12 Intake design

See ANSI/HI 9.8 *Pump Intake Design* for an in-depth discussion of this subject.

1.3.6.13 Pump and motor speed–torque curves

A plot of speed versus torque requirements during the starting (accelerating) phase of a rotodynamic pump (any pump type) is sometimes needed to check against the speed–torque curve of the driving motor. Rotodynamic pumps typically have a speed–torque curve characteristic where the torque varies as the square of the speed. The driver must be capable of supplying more torque than required by the pump along the entire curve in order to bring the pump up to rated speed under the conditions present during the starting phase. This is generally attainable for rotodynamic pumps with normal induction or synchronous motor performance characteristics. However, under certain conditions, such as with high specific speed pumps, or when motor terminal voltage is reduced below nominal tolerances, a motor with higher pull-in torque may be required to maintain adequate torque margins and ensure expected pump acceleration to operating speed. Reduced motor terminal voltage will result in reduced motor torque. For induction motors, torque is reduced in proportion to the square of the applied terminal voltage. Thus, a motor whose terminal voltage dips to 80% of nominal voltage during start-up will only produce accelerating torque equal to 64% of the torque under nominal full voltage conditions. Obviously, available torque is reduced significantly further when starting voltage defined by the end user is lower than 80%, and care should be taken to consider these situations.

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

Table 1.3.6.11a — Guidelines for minimum and maximum liquid temperature for gray iron, ductile iron, carbon steel, chrome steel, austenitic stainless and duplex stainless steel pumps (°C)

TUTORIAL NOTE: The maximum temperatures cited may be higher than the limits imposed by some user specifications. Materials selected for such applications must also be evaluated to match the requirements of the end user.

		Flexibly Coupled Pumps		Close-Coupled Pumps	
		Maximum Temperature		Maximum Temperature	
Material	Minimum Temperature[a]	Without Cooling	With Cooling[b,c,d]	Without Cooling	With Cooling[c]
Gray Cast Iron	-30	175	175	120	175[b]
Ductile Iron	-30	175	340	120	175–340[b]
Carbon Steel	-30	120	425	100	380
Chrome Steel	-100	120	425	100	380
Austenitic	-196	120	370	100	370
Duplex	-30	120	260	100	260

[a] Minimum temperature depends on pump configuration, sealing arrangement, and proven low-temperature ductility of the case material.
[b] Cooling is generally applied to the bearing assembly to prevent overheating of the lubricating oil.
[c] Cooling is also applied to the process fluid to prevent flashing at seal faces (flexibly coupled) or other areas of heat load, such as motor windings (close coupled).
[d] Recommendations for cooling vary with mechanical seal selection and seal flush piping arrangements.

Table 1.3.6.11b — Guidelines for minimum and maximum liquid temperature for gray iron, ductile iron, carbon steel, chrome steel, austenitic stainless and duplex stainless steel pumps (°F)

TUTORIAL NOTE: The maximum temperatures cited may be higher than the limits imposed by some user specifications. Materials selected for such applications must also be evaluated to match the requirements of the end user.

		Flexibly Coupled Pumps		Close-Coupled Pumps	
		Maximum Temperature		Maximum Temperature	
Material	Minimum Temperature[a]	Without Cooling	With Cooling[b,c,d]	Without Cooling	With Cooling[c]
Gray Cast Iron	-20	350	350	250	350[b]
Ductile Iron	-20	350	650	250	350–650[b]
Carbon Steel	-20	250	800	212	715
Chrome Steel	-150	250	800	212	715
Austenitic	-320	250	700	212	700
Duplex	-20	250	500	212	500

[a] Minimum temperature depends on pump configuration, sealing arrangement, and proven low-temperature ductility of the case material.
[b] Cooling is generally applied to the bearing assembly to prevent overheating of the lubricating oil.
[c] Cooling is also applied to the process fluid to prevent flashing at seal faces (flexibly coupled) or other areas of heat load, such as motor windings (close coupled).
[d] Recommendations for cooling vary with mechanical seal selection and seal flush piping arrangements.

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; see curve, Figure 1.3.6.13a.

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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 2 to 15% of the maximum torque at rated speed.

Speed torque requirements for starting conditions other than closed discharge will vary depending on the percentage of static head to total head; the volume content of the discharge line; the condition of the discharge line, whether 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.

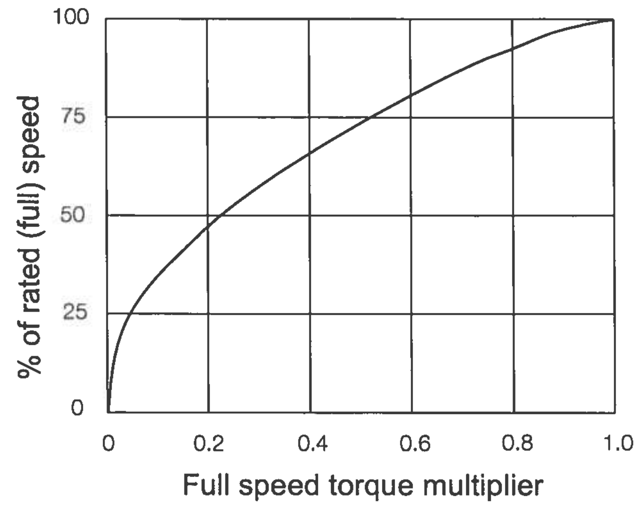

Figure 1.3.6.13a — Torque curve

The requirements of pump starting and running torque should be understood and transmitted to the driver manufacturer so that both the pump supplier and the driver manufacturer understand, accept, and account for the applicable conditions to ensure expected starting performance. 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 closed valve or a check valve, which may require lower torque from the driver, but have a limitation on operating time at this condition, due to temperature rise or excessive hydraulic loads.

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.

Large pumps and special starting requirements for the driver need to be reviewed on an individual basis by engineering personnel.

For high viscosities (greater than 500 cP) the breakaway torque becomes more significant. Additionally, torque increases, approaching a linear relationship with speed from near-zero to full speed.

The following curve (see Figures 1.3.6.13b and 1.3.6.13c) shows how pump starting and running torque requirements may be defined for transmittal to driver manufacturers so that they can confirm driver capability to meet the requirements, prior to ordering. In addition to defining the data shown in these figures, the pump supplier should also specify which starting condition(s) the driver is required to comply with, i.e., open valve and/or closed valve.

Curves may be completed using the examples and the WR^2 (mass-moment of inertia) 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. Pumps with high-viscosity fluids or limited motor inrush limits require review.

Small pumps (less than 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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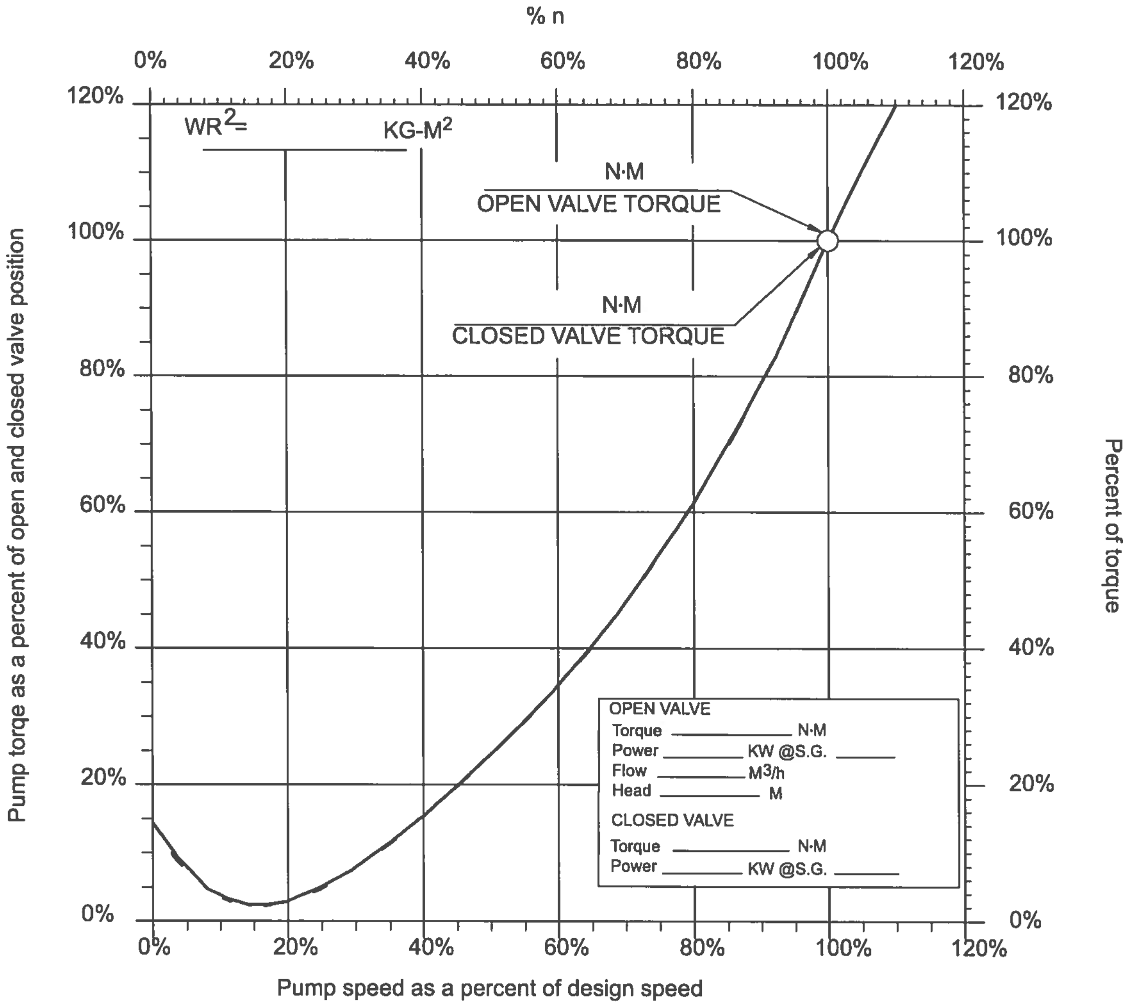

Figure 1.3.6.13b — Evaluation of pump torque versus speed (metric units)

The following examples illustrate how the speed–torque curves may be prepared to assist in the pump and driver selection process.

Example (metric units)

$$P(\text{kW}) = \frac{Q[m^3/h] \times H[m] \times s}{\eta_{pump} \times 367.1} \;\&\; \tau[\text{N•m}] = \frac{P(\text{kW}) \times 9549.3}{n_{rpm}} \Rightarrow \tau = \frac{Q[m^3/h] \times H[m] \times s \times 26}{\eta_{pump} \times n_{rpm}}$$

$$\tau_{OPEN-VALVE} = \frac{272.5 \times 1135.7 \times 0.96 \times 26}{0.803 \times 3580} = 2687 \text{ N·m}$$

$$\tau_{CLOSED-VALVE} = \frac{136.3 \times 1295 \times 0.96 \times 26}{0.65 \times 3580} = 1893 \text{ N·m}$$

Axial split between-bearings multistage pump type BB3 with eight stages.

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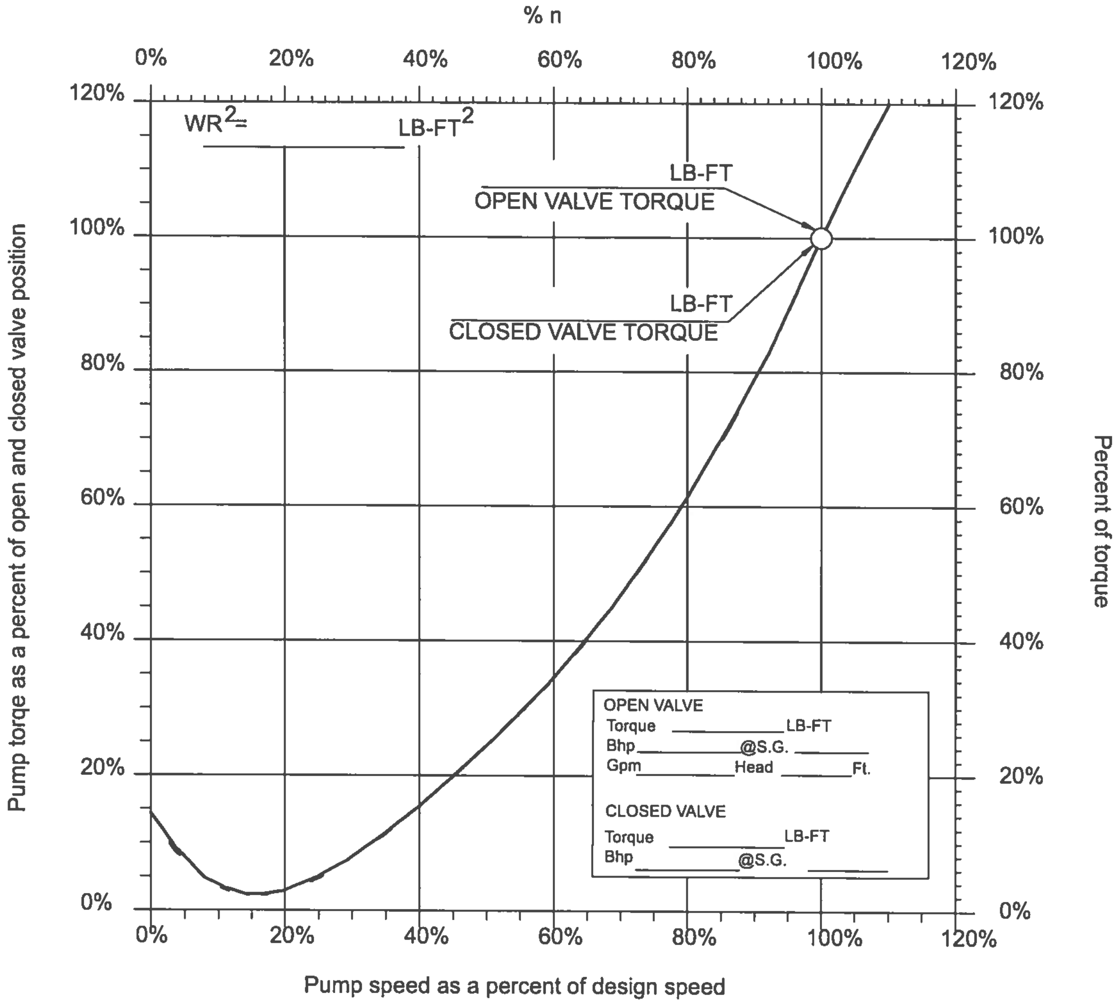

Figure 1.3.6.13c — Evaluation of pump torque versus speed (US customary units)

Rated power = 1007 kW at the design condition and 708 kW at minimum flow.

The customer would like to use an electric motor driver with 80% voltage applied to the motor terminals (either due to voltage dip during starting or by use of dedicated reduced voltage starting equipment). The following steps may be used to prepare the curve.

1) Select the driver size.

   Selection is a 1120-kW electric motor with a rated motor torque of 2983 N·m (information supplied by driver vendor).

2) Calculate pump torque at both open and closed valve conditions:

   WR^2 = 1.087 kg-m^2. This value is based on the manufacturer's data for pump type BB3 with eight stages.

3) Fill in the torque and mass inertia values of the pump.

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4) When all the above data are supplied to the motor vendor along with all other motor parameters, including voltage during start-up, they can plot motor torque against pump torque to determine if their standard motor design can start the pump and how long it will take to reach operating speed.

   Given that the torque of the motor is proportional to the square of the voltage, then the torque of the motor during acceleration will be reduced to 64% of nominal at all points along the motor speed–torque curve for the 80% voltage starting condition specified. Compromised torque margins typically appear at the speed associated with motor pull-in torque and/or at motor full load/rated power torque. The pump engineer can compare the rated speed torque requirements of the pump at open and closed valve versus the motor rated torque and reduced voltage torque at rated speed to determine if there is adequate margin. A margin of 10% could be considered adequate. Typically the motor engineer will attempt to ensure that there is adequate torque margin between pump requirements and motor torque under the reduced voltage/torque condition at all points along the curve.

   If the standard motor design is not adequate to ensure necessary torque margin throughout the acceleration period, alternate designs may be proposed, or the closed valve condition may be specified as the start-up condition instead of the open valve condition if it is only the open valve condition that is compromised.

   In this example, the motor rated (full voltage) torque of 2983 N·m provides 11% greater torque than the pump open valve rated condition of 2687 N·m and thus could be considered satisfactory.[1] However, at 80% voltage, motor rated torque would only be 1909 N·m and the motor could not start the pump. However, at the pump closed valve condition of 1893 N·m the motor does have enough torque at 80% voltage, but the margin would be considered too small and an adjustment would be needed either in the motor and/or the pump's defined closed valve condition.

5) See the curve in Figure 1.3.6.13d for an example of plotting actual values.

Another example of preparing a speed versus torque curve is when speed increasing or decreasing gears are involved. *Here the pump torque requirements must be translated to the output of the driving machine.*

1) If the pump in the previous example is going to be driven by a diesel engine rated 1305 kW at 1000 rpm, then a speed increasing gear would be required with a gear ratio of 3.58:1.

2) Since the power P in kW is the same whether it is through a gear or direct drive, the "n" or "rpm" component of the torque equation must be corrected.

3) Simply "n" must be divided by the speed increasing ratio

$$\tau = \frac{P \times 9549.3}{n_{pump}/r_{gear-ratio}} = \frac{1007 \times 9549.3}{3580/3.58} = 9616 \text{ N·m}$$

4) Also the pump inertia must be overcome. Previously the WR^2 value of 1.087 kg-m^2 was obtained from the pump vendor. The inertia that the driver undergoes is a square of the speed increasing ratio plus the inertia of the gear:

$$WR^2 = r^2_{gear-ratio} \times WR^2_{pump} + WR^2_{gear} = 3.58^2 \times 1.087 + 5.057 = 19.0 \text{ kg} \cdot \text{m}^2$$

NOTE: If step-down gearbox, then divide by the ratio squared.

When gears are used there is an efficiency reduction due to losses in the gear.

If an estimate of 2% losses in the gear is considered for a single step speed increaser, then there is a corresponding need to divide the power (kW) required by the pump by 0.98 to determine power required at driver output shaft.

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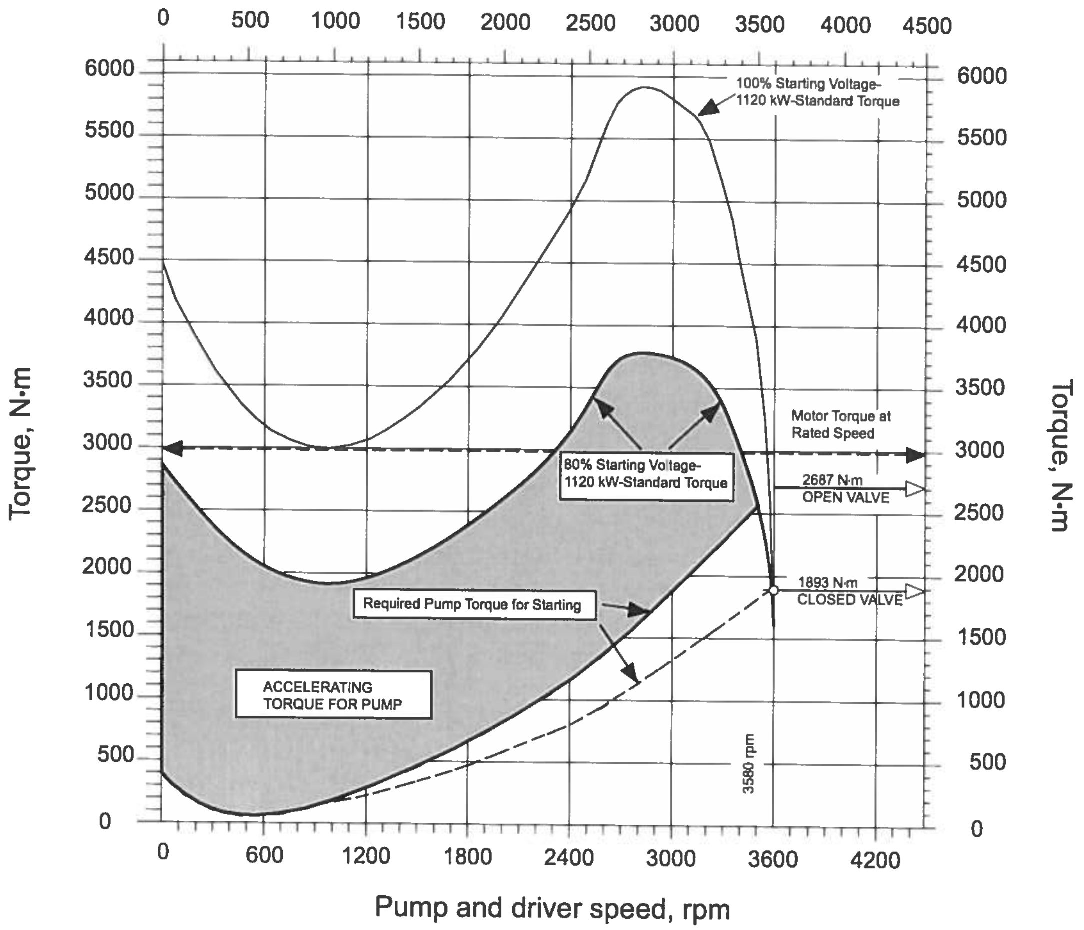

Figure 1.3.6.13d — Plotting torque versus speed (metric units)

The following examples illustrate how the speed–torque curves may be prepared to assist the pump and driver selection process.

Example (US customary units)

Axial split between-bearings multistage pump type BB3 with eight stages.

Rated power = 1350 hp at the design condition and 950 hp at minimum flow.

The customer would like to use an electric motor driver with 80% reduced voltage starting.

The following steps may be used to prepare the curve.

1) Select the driver size.

   Selection is a 1500-hp electric motor with a rated motor torque of 2200 lb-ft (information supplied by driver vendor).

2) Calculate pump torque at both open and closed valve conditions.

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$$P[BHP] = \frac{Q[US\ gpm] \times H[ft] \times s}{\eta_{pump} \times 3960} \ \& \tau_{Open\text{-}Valve}\ [\text{N•m}] = \frac{P[BHP] \times 5250}{n_{[\text{rpm}]}} \Rightarrow \tau_{Closed\text{-}Valve}\ [\text{N•m}]$$

$$= \frac{Q[US\ gpm] \times H[ft] \times s \times 5250}{\eta_{pump} \times 3960 \times n_{[\text{rpm}]}}$$

$$\tau_{Open\text{-}Valve} = \frac{1200 \times 3726 \times 0.96 \times 5250}{0.803 \times 3960 \times 3580} = 1980 \text{ lbf-ft}$$

$$\tau_{Closed\text{-}Valve} = \frac{600 \times 4248 \times 0.96 \times 5250}{0.65 \times 3960 \times 3580} = 1393 \text{ lbf-ft}$$

WR^2 = 25.8 lb-ft^2. This value is based on the manufacturer's data for pump type BB3 with eight stages.

3) Fill in the torque values and mass inertia of the pump.

4) When all the above data are supplied to the motor vendor along with all other motor parameters, including voltage during start-up, they can plot motor torque against pump torque to determine if their standard motor design can start the pump and how long it will take to reach operating speed.

   Given that the torque of the motor is proportional to the square of the voltage, then the torque of the motor during acceleration will be reduced to 64% of nominal at all points along the motor speed–torque curve for the 80% voltage starting condition specified.

   Compromised torque margins typically appear at the speed associated with motor pull-in torque and/or at motor full load/rated power torque. The pump engineer can compare the rated speed torque requirements of the pump at open and closed valve versus the motor rated torque and reduced voltage torque at rated speed to determine if there is adequate margin. A margin of 10% could be considered adequate. The motor engineer will usually attempt to ensure that there is adequate torque margin between pump requirements and motor torque under the reduced voltage/torque condition at all points along the curve.

   If the standard motor design is not adequate to ensure necessary torque margin throughout the acceleration period, alternate designs may be proposed, or the closed valve condition may be specified as the start-up condition instead of the open valve condition if it is only the open valve condition that is compromised.

   In this example, the motor rated (full voltage) torque of 2200 lbf-ft provides 11% greater torque than the pump open valve rated condition of 1980 lbf-ft and thus could be considered satisfactory.[1] However, at 80% voltage, motor rated torque would only be 1408 lbf-ft and the motor could not start the pump. However, at the pump closed valve condition of 1393 lbf-ft the motor does have enough torque at 80% voltage, but the margin would be considered too small and an adjustment would be needed either in the motor and/or the pump's defined closed valve condition.

5) See the curve in Figure 1.3.6.13e for an example of plotting actual values.

Another example of preparing a speed versus torque curve is when speed increasing or decreasing gears are involved. *Here the pump torque requirements must translated to the output of the driving machine.*

1) If the pump in the previous example is going to be driven by a diesel engine rated at 1750 hp at 1000 rpm, then a speed increasing gear would be required with a gear ratio of 3.58:1.

2) Since the BHP is the same whether it is through a gear or by direct drive, the "n" or "rpm" component of the torque equation must be corrected.

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[1] This example does not include the additional requirement from NEMA and IEC that the motor must be capable of operating at ±10% of specified voltage. The motor designer will consider this as well.

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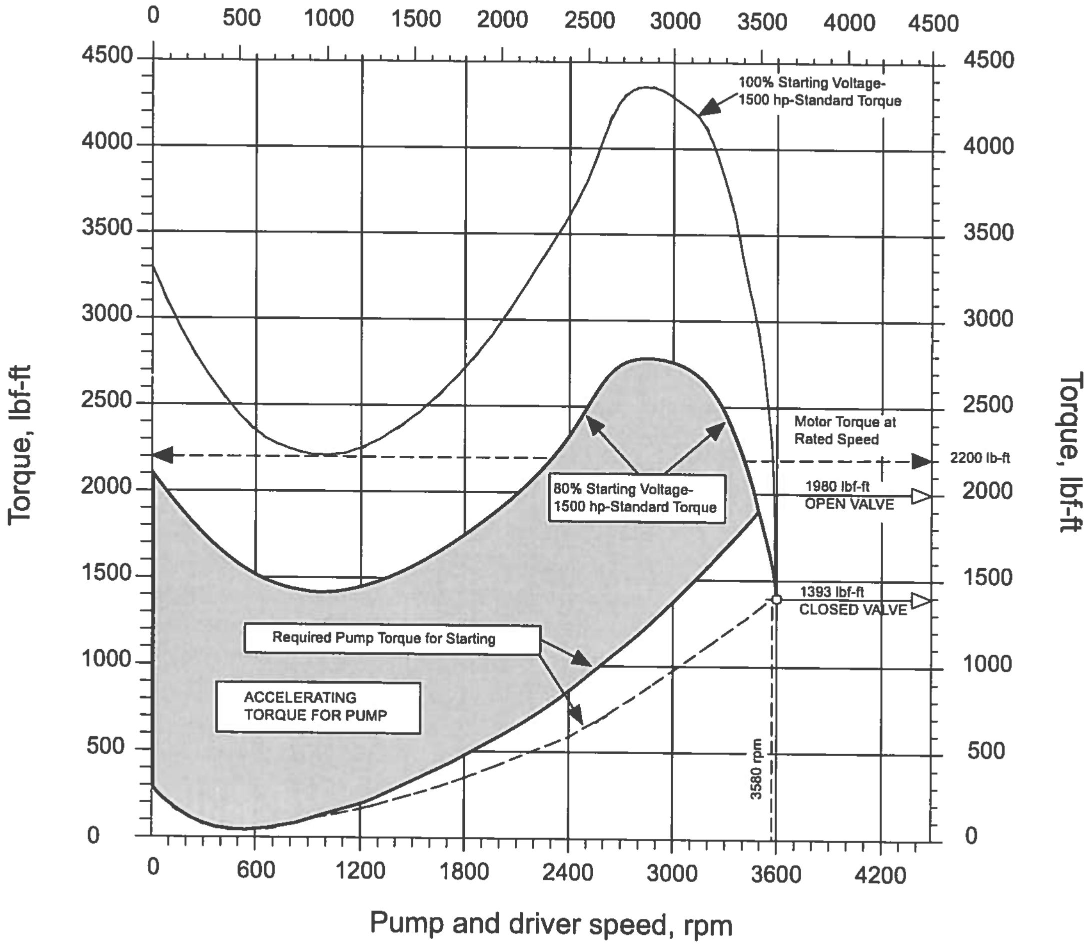

Figure 1.3.6.13e — Plotting pump torque versus speed (US customary units)

3) Simply "n" must be divided by the speed increasing ratio

$$\tau = \frac{Bhp \times 5250}{n_{pump}/r_{gear-ratio}} = \frac{1350 \times 5250}{3580/3.58} = 7088 \text{ lbf-ft}$$

4) Also the "pump and gear inertia" must be overcome. Previously the WR^2 value of 25.8 lb-ft^2 was obtained from the pump supplier data. Now the inertia that the driver undergoes is a square of the speed increasing ratio plus the inertia of the gear:

$$WR^2 = r^2_{gear-ratio} \times WR^2_{pump} + WR^2_{gear} = 3.58^2 \times 25.8 + 120 = 450 \text{ lb-ft}^2$$

If a step-down gearbox is used, then divide by the ratio squared.

When gears are used there is an efficiency reduction due to losses in the gear.

If an estimate of 2% losses in the gear is considered for a single step speed increaser, then there is a corresponding need to divide the power (hp) required by the pump by 0.98 to determine power required at driver output shaft.

Speed–torque curves will vary depending on pump specific speed (N_s), however, for most radial flow machines, the above speed–torque curve is representative.

The "closed valve" torque will vary depending on the system curve it will start against.

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Large pumps and special driver starting requirements need to be reviewed on an individual basis.

For high viscosities (greater than 500 cP) the breakaway torque becomes more significant.

1.3.7 Noise levels

The overall noise level of a rotodynamic pump set is the combination of the noise contribution of the individual components, i.e., pump, driver, gearbox, etc. In general, noise is generated in rotodynamic pumps by hydraulic effects transmitted to the pump case, which in turn generates airborne noise. Noise will also propagate through fluid passages (pump piping) and structural paths (equipment base).

Pump noise generation becomes more significant with increasing speed, power, and modern lightweight construction. Electric-motor-driven pumps can have a greater contribution from the motor to the overall noise level than that contributed by the pump, especially at lower power levels.

For those applications where minimum noise levels are required, the primary application rule is to select the pump at both a conservative rpm and liquid velocity level. This will often rule out the use of the smallest, most economical pump that will operate at the highest possible speed and with high liquid velocities. A quiet installation also demands complete freedom from possible cavitation, and this means a conservative NPSH margin and careful consideration of the suction piping layout.

Hydraulic noise typically increases with pump operation at flow rates well below or well above the BEP of the pump. Higher specific-speed pumps are more sensitive than lower specific-speed pumps in this regard. Pump oversizing with respect to purchaser-specified margin must be minimized to reduce pump noise levels.

Refer to the ANSI/HI 1.4 *Rotodynamic Pumps for Installation, Operation, and Maintenance* for additional details.

1.3.7.1 Estimation of sound pressure levels

In general noise is generated in rotodynamic pumps by hydraulic effects transmitted to the pump case, which in turn generates airborne noise. Noise will also propagate through fluid passages (pump piping) and structural paths (equipment base).

An estimation of pump sound pressure level can be calculated from the pump absorbed power and speed as follows:

$$L_{PA} = 63.3 + 13.8 * \log_{10}P - (23.0 + \log_{10}P - 3.0 * \log_{10}n)$$

Where:

L_{PA} = pump sound pressure level, in dBA

P = pump absorbed power, in kW

n = pump speed, in rpm

This equation is valid for a single-stage pump operating at or close to BEP.

As mentioned, the overall noise level of a rotodynamic pump set is the combination of the noise contribution of the individual components, i.e., pump, driver, gearbox, etc. Electric-motor-driven pumps can have a greater contribution from the motor to the overall noise level than that contributed by the pump, especially at lower power levels. The sound pressure level of the motor must be provided by the supplier under load condition.

The combined noise level of a pump and driver can be calculated using Figure 1.3.7.1a. The sound pressure increase should be added to the highest individual level, i.e., driver or pump.

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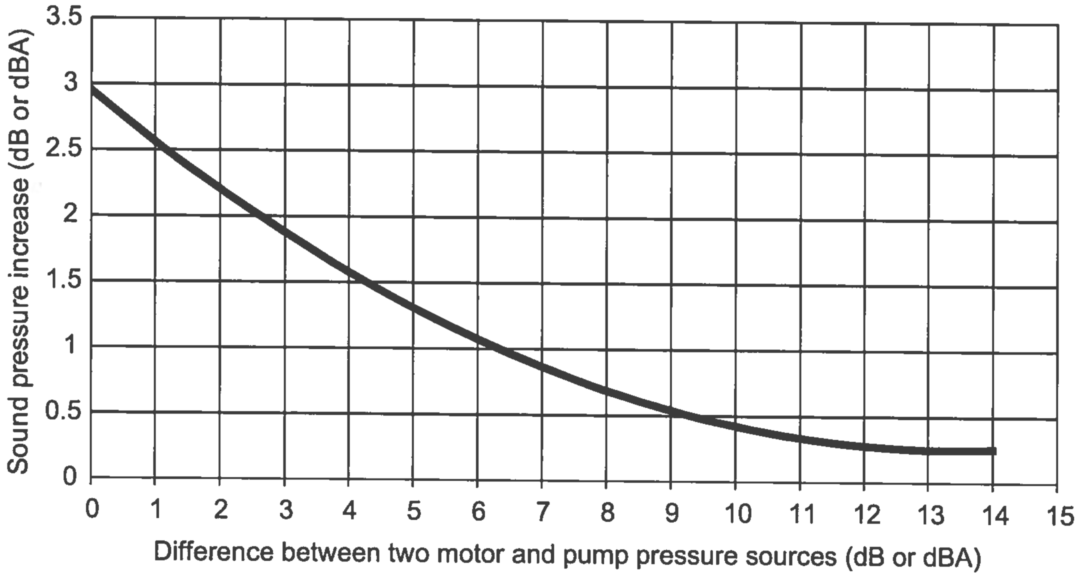

Figure 1.3.7.1a — Combining sound pressure

Example 1 per Figure 1.3.7.1a

Two separate sound sources have sound pressure levels of 85 dB each.
The difference between sources is 0.
Therefore the increase will be 3 dB, for a combined noise value of 88 dB.

Example 2 per Figure 1.3.7.1a

Two separate sound sources have sound pressure levels of 80 dB and 84 dB, respectively.
The difference between sources is 4 dB.
Therefore, the increase will be 1.5 dB over the higher sound level, for a total value of 85.5 dB.

The equipment sound pressure level calculated is based on using the method of prescribed surface technique for measurement as described in ANSI/HI 9.1–9.5 *Pumps - General Guidelines for Types, Applications, Definitions, Sound Measurement, and Decontamination* in Section 9.4.

Several correction factors can be applied to the pump noise level estimated using the equation above.

a) Pump stages

Noise measurements show that multistage pumps exhibit lower noise levels compared to single-stage pumps of the same power.

Table 1.3.7.1 — Multistage pump sound pressure reductions

Number of stages	Multistage dBA correction
2	-1.25
3	-1.95
4	-2.28
5	-2.48
6 and more	-2.55

b) Impeller trim and operating point

Hydraulic noise typically increases with pump operation at flow rates well below or well above the BEP of the pump, or with an increase in the impeller diameter as shown in Figure 1.3.7.1.b. This correction factor is highly dependent on pump specific speed and pump construction. As a result, Figure 1.3.7.1.b should only be used as a guide.

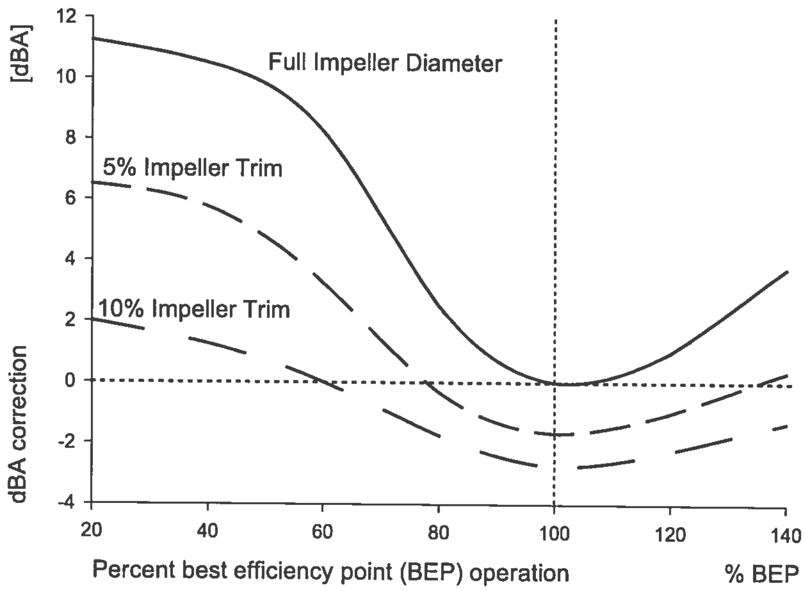

Figure 1.3.7.1b — dBA correction for impeller trim and percent BEP operation

c) Power cut-off

The equation in Section 1.3.7.1 predicts a very low noise level for low-power pumps. It is impractical to measure such low levels due to background noise. As a result, it is common for a cut-off point to be introduced by the pump manufacturer, e.g., 70 dBA minimum overall noise level.

d) Noise cladding or enclosure

Noise insulation in the form of cladding or enclosure will reduce the noise energy transmitted to the air by the equipment. The supplier of the noise insulation can estimate the appropriate reduction factor. Caution should be taken in the use of cladding or enclosures since these will also provide heat insulation with the net effect of increased air temperature within.

e) Barrel pumps (double case)

The outer case of a barrel pump acts similarly to a noise cladding on the internal pump cartridge. This noise reduction can be estimated at approximately –1.0 dBA.

Refer to ANSI/HI 1.4 *Centrifugal Pumps for Installation, Operation, and Maintenance* for additional details. Refer to ANSI/HI 9.1-9.5 *Pumps – General Guidelines* for details of measurement methods.

1.3.8 Baseplates – introduction

This section establishes guidance on technical criteria to be used in designing horizontal rotodynamic pump baseplates. Included within this section are explanations of typical baseplate types, analysis requirements, dimensional tolerancing, and alignment requirements.

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This standard is not intended to cover baseplates for pump designs in which the driver and pump are integral. Such machines do not have separate shafts, which require alignment. Examples of pumps with integral drives are close-coupled pumps and canned motor pumps.

1.3.8.1 Functional requirements

A baseplate is the structure to which the pump, motor, gearbox, and all auxiliary equipment are mounted. The purpose of a baseplate is to provide a foundation under a pump and its driver that maintains shaft alignment between the two. This baseplate must allow for initial mounting and alignment of equipment, survive handling during transportation to the installation site, be capable of being installed properly with minimum difficulty, allow final alignment of the mounted equipment, control spillage, and allow removal and reinstallation of equipment. It must be recognized that it is not necessary that an absolutely rigid baseplate be designed to meet these requirements. At the same time, the baseplate must not be permanently deformed after the equipment is mounted at the manufacturing facility. Compliance with these design criteria, in conjunction with proper installation procedure, will contribute significantly to meeting the functional requirements.

Any baseplate must be designed to satisfy numerous functional requirements.

To ensure correct design of the baseplate it is necessary to review all application parameters, including equipment selection, installation, and operational requirements. For standardized pump ranges with predefined and specific applications, the equipment manufacturer should have taken all of these factors into account. For customized pump applications it is necessary to review these fundamentals at the time of proposal.

Table 1.3.8.1 — Functions of a baseplate and parameters for selection

Functions of a baseplate	Parameters for selection
Support pump, driver, and auxiliary equipment.	Pump, driver, and equipment design.
Contain spillage leakage of process fluids. Drain to collection point(s).	Nature of process fluid. Nature of lubrication fluids. Environmental effects.
Resist corrosion caused by environment and/or service conditions.	Installation environment. Nature of process fluid. Nature of lubrication fluids.
Allow for lifting and transportation of assembled equipment.	Mass of pump, driver, and equipment. Permanency of installation.
Allow access to equipment for operation monitoring, service, and maintenance.	Pump, driver, and equipment design. Installation environment. Operation and maintenance capabilities.
Provide anchor to foundation.	Foundation type and design. Permanency of installation.

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Table 1.3.8.1 — Functions of a baseplate and parameters for selection (*continued*)

Functions of a baseplate	Parameters for selection
Withstand all combinations of static and dynamic loads caused by deadweight of equipment, piping (nozzle) loads, torque loads (startup, shutdown, and continuous operation), wind loads, seismic loads, transient thermal conditions.	Mass of pump, driver, and equipment. Size and design of attached piping. Driver power and torque characteristics. Environmental conditions. Safety regulations.
Maintain acceptable equipment alignment under all prescribed load conditions.	Pump, driver, and equipment design. Coupling selection. Customer specification.
Provide adaptation for anticipated movement of mounted equipment, i.e., thermal growth.	Pump application and operating conditions. Driver type.
Provide adaptation for anticipated movement of foundation support structure, i.e., offshore platform or ship - floating production storage and offloading platform (FPSO).	Installation location. Foundation type and design.
Provide smooth running operation. Stable structural operation at all defined operating speeds, confirmed by natural frequency analyses and validations to identify proven and 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.

1.3.8.2 Baseplate types

These include grout type, nongrout type, pregrout type, sole plate, and freestanding and are discussed in more detail in the following paragraphs.

Baseplate design standards include ASME B73.1, ISO 4664, API 610 (ISO13709), and API RP 686.

Baseplate styles include

ASME B73.1/ISO 2858
Nonmetallic
Offshore skid type
Oil pipeline skid type
Gimbal mounted
Fabricated steel
Cast iron
Pedestal or centerline mounted
Foot mounted
Water-cooled pedestal

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1.3.8.2.1 Grouted baseplate

The baseplate is designed to allow grout to be poured underneath the base. The grout placed inside the base contributes to the baseplate's installed rigidity and damping. See Figure 1.3.8.2.1a.

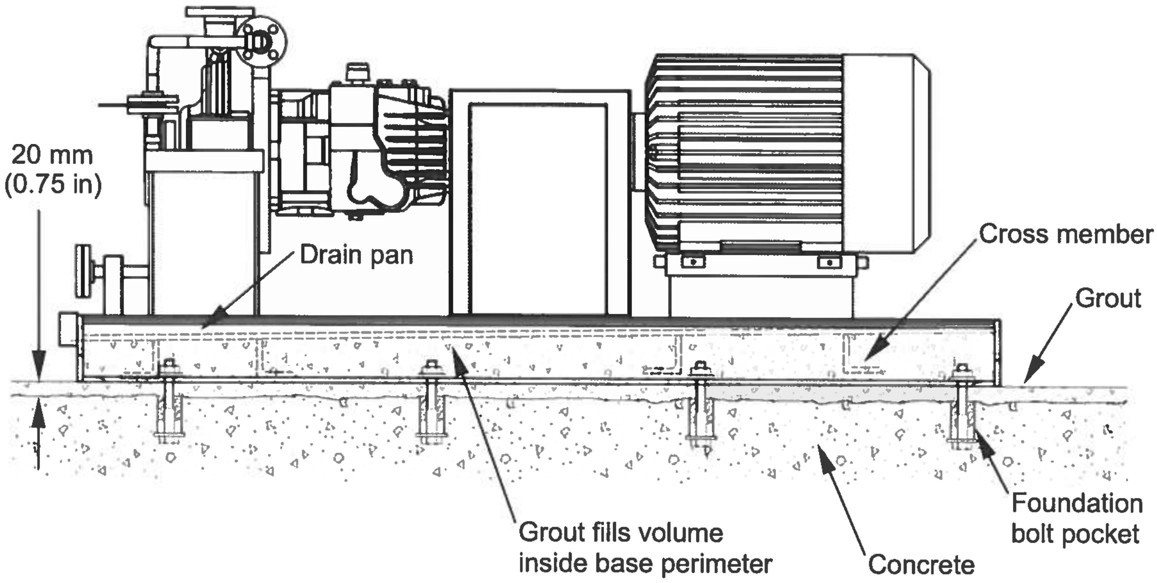

Figure 1.3.8.2.1a — Grouted baseplate, fabricated steel

The cross members used on this type of base are normally designed to lock into the grout and further resist deflection or vibration of the baseplate. Typically the cross member geometry chosen to achieve this is an L-section (shown), a T-section, or an I-section.

If the baseplate is a closed design (i.e., grout cannot be poured inside the baseplate perimeter due to the presence of a drain pan or deck plate), then grout holes must be provided to allow the grout to be placed inside the base. (See Section 1.3.8.8 for details of this requirement.)

The grout used may be either cementitious or epoxy-based. The surface preparation required for a baseplate to successfully bond to the grout is different depending on which grout will be used. It is therefore important that the vendor and customer agree in advance as to which type of grout will be used.

The baseplate described and shown in Figure 1.3.8.2.1a above is typical of a fabricated baseplate. Cast-iron baseplates are another type of grouted baseplate. The ability to integrally cast in features such as bracing, grout holes, and sloping surfaces provides a highly functional and economical solution for many applications.

1.3.8.2.2 Nongrout-type baseplate

This baseplate is placed directly on a foundation without the use of grout to fill the interior of the base to lock it to the foundation. Because of the loss of stiffening normally provided by the grout, nongrout bases must typically be structurally stiffer than comparable grouted bases.

Cross members do not need to lock into the grout and so may be selected on the basis of providing the best stiffening effect. For this reason hollow rectangular sections are often used on this design.

The advantage of this design over a grouted design is the simplified installation requirements.

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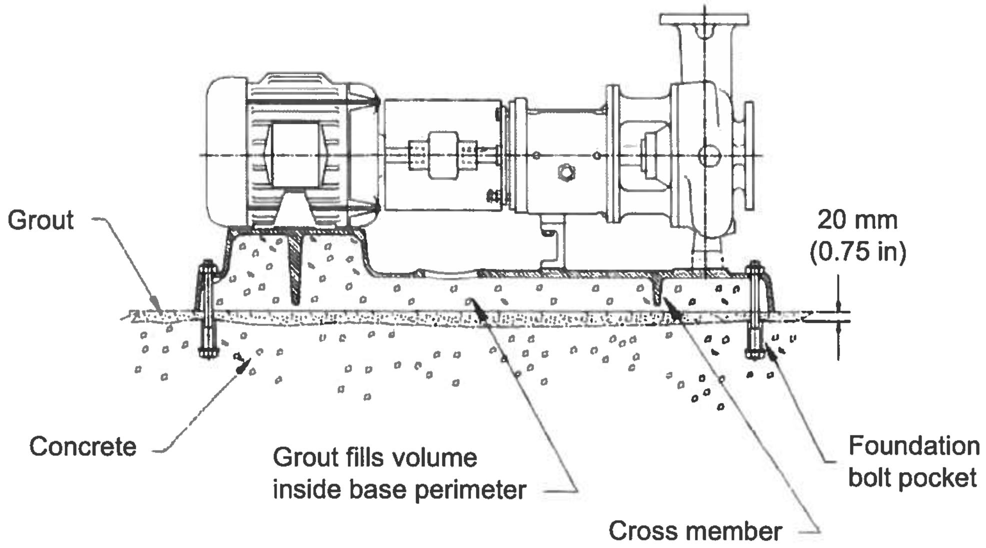

Figure 1.3.8.2.1b — Grouted baseplate, cast iron

The structural design must be stiffer than an equivalent grout-type base. Additionally the structural natural frequencies must be separated from any equipment operating frequencies. This is because the stiffening and dampening effects of the grout are not available.

The nongrout type is often used in offshore installations where the mass of grout and concrete must be avoided. This type can be provided for all types of rotodynamic pumps. (A multistage, axial split, between-bearings pump is illustrated).

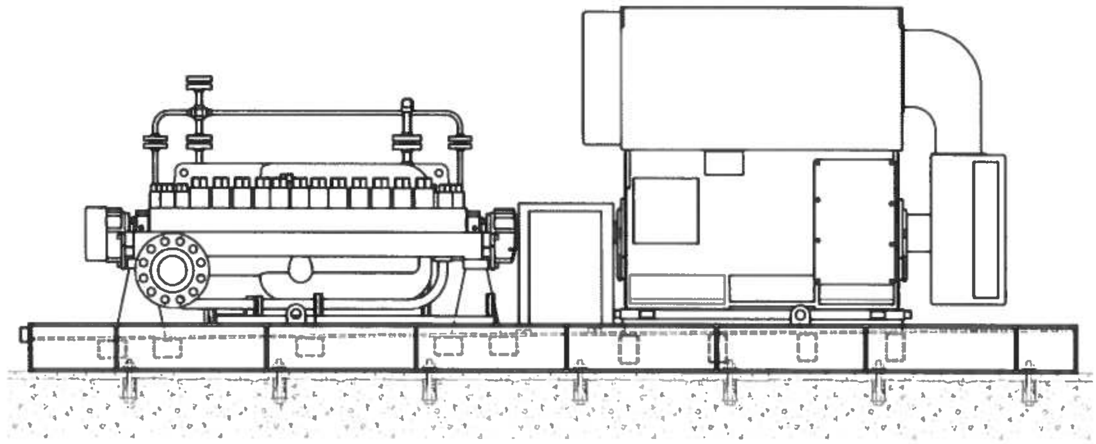

Figure 1.3.8.2.2 — Nongrout baseplate

1.3.8.2.3 Pregrouted baseplate

This baseplate is a variation of the grouted design. With this design, the baseplate interior is prefilled with an epoxy grout prior to final assembly of the pump and driver. To achieve this requires the base to be inverted and epoxy grout poured into the cavity until it is full. Only the closed type of baseplate (with a drain pan or deck plate) can be used with this process. Because the base is filled while inverted, no grout or vent holes are required.

Final grouting of the base to the foundation is still performed at site, but is a simplified procedure because the requirement to fill the interior of the baseplate has already been met.

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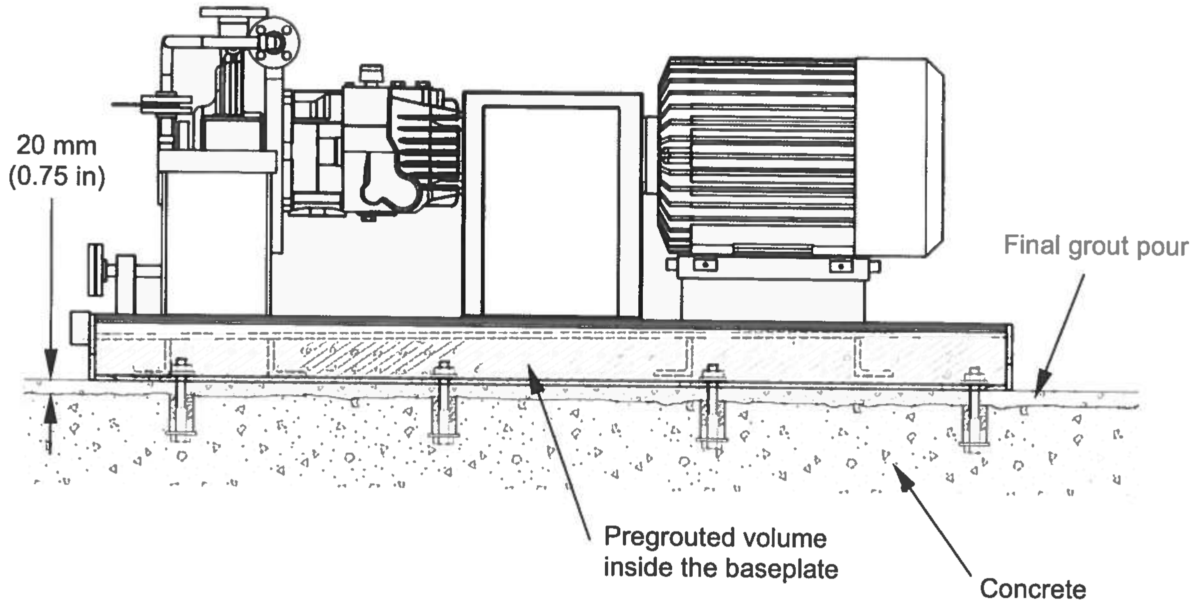

Figure 1.3.8.2.3 — Pregrouted baseplate

The advantages of pregrouting are:

- The grout inside the base will be free from voids and air pockets.
- Final grouting of the base at site is greatly simplified by eliminating the need to fill the interior of the base.
- The grout inside the base adds to stiffness. This will minimize the distortion of the mounting surfaces during installation.

Care and attention are required during the pregrouting process. It is normally advisable to perform final machining of the baseplate mounting surfaces after the pregrout has completely cured.

1.3.8.2.4 Soleplate

Soleplates are cast-iron or steel pads used in place of a bedplate to mount a large pump or a train of equipment. They are located under the feet of the pumps or its drivers and are grouted directly to the foundation. The equipment can be doweled or bolted to them. Other designs have the foundation bolt pass through the grouted soleplate.

On large horizontal units or equipment train with a pump, gear reducer, driver, and couplings, soleplates are used to save the cost of a large bedplate that would be otherwise required.

The corners of square or rectangular soleplates are rounded to prevent stress risers in the grouting.

1.3.8.2.5 Freestanding baseplate

This is a baseplate that is intended to be elevated off the floor, and supported by stilts, shims, or springs. All support types may additionally be used with slide bearing plates (see Figures 1.3.8.2.5a and 1.3.8.2.5b). This type of baseplate must be designed to provide its own rigidity, as there is no grout for additional support.

The features of this type of baseplate are

- The need for concrete foundation pads is removed

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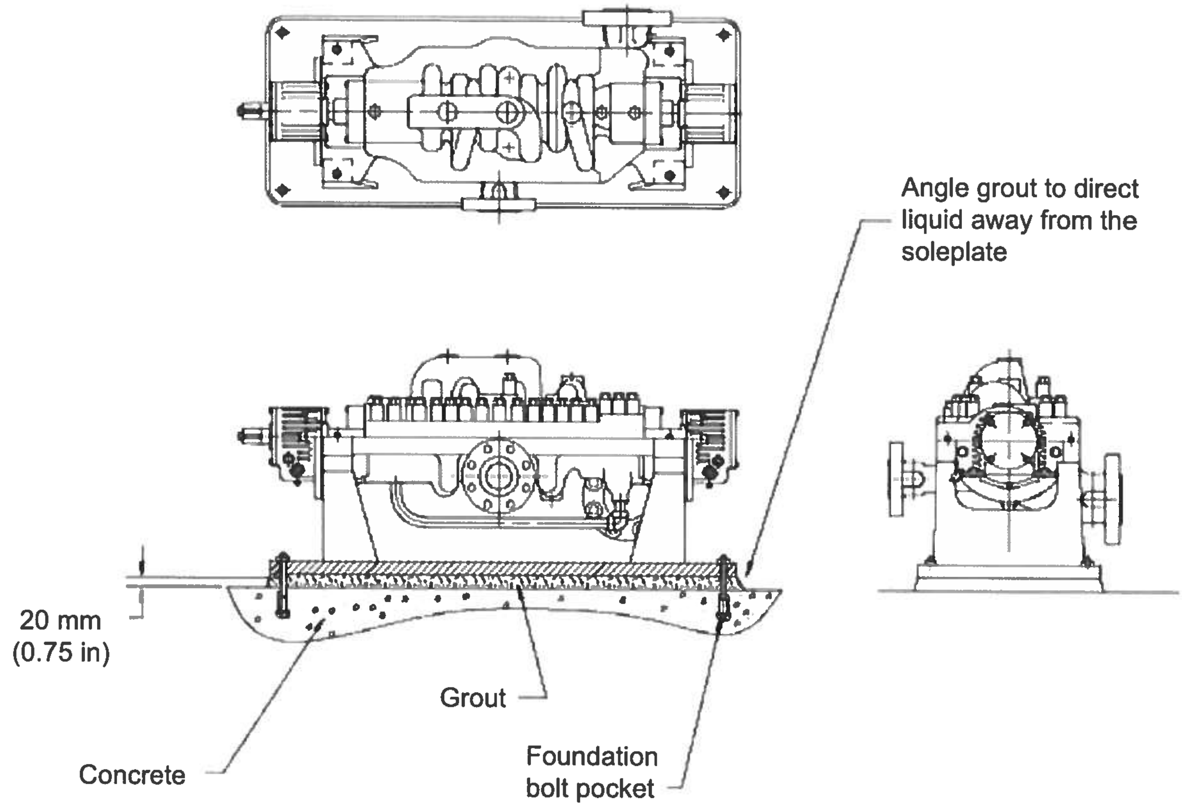

Figure 1.3.8.2.4 — Soleplate

- An allowance for height adjustment can easily be included
- Spring-mounted bases provide vertical displacement under applied loads associated with thermal expansion of the piping (reducing the associated piping stresses)
- Optional addition of slide bearing plates may also be provided below the stilts or springs to allow horizontal displacement associated with thermal expansion of the piping (reducing the associated piping stresses)

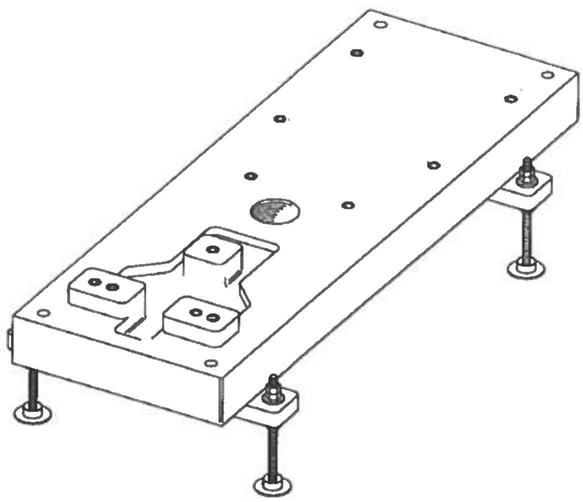

Figure 1.3.8.2.5a — Freestanding baseplate

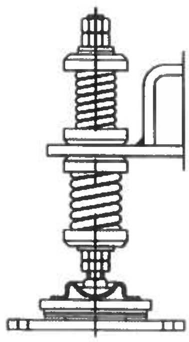

Figure 1.3.8.2.5b — Baseplate mounting springs with slide bearing

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1.3.8.2.6 ASME B73 and ISO 2858 standard baseplates

Horizontal end suction chemical process pumps and their baseplates are commonly designed and built in accordance with ASME B73.1 (also referred to as *ANSI pumps*) or ISO 2858, ISO 3661, and ISO 5199.

Baseplates to these standards are designed to support only the specific end suction, single-stage design, back pullout pump designs with foot-supported casings, stipulated in the standards themselves.

1.3.8.2.6.1 ASME B73 baseplates

Baseplates that conform to ASME B73.1 are identified by a three-digit number. The first digit is the number of the pump group for which each baseplate is designed, and the second and third digits are the overall length of the baseplate in inches. The specification requires the baseplate to be long enough to avoid any overhang of the driver beyond the end of the baseplate. ASME B73.1 standard requires that baseplates be designed per ANSI/HI 1.3 and dimensioned in accordance with Table 2 of ASME B73.1.

The specification recognizes the option of freestanding baseplate designs and requires the user to contact the manufacturer for additional space required for width dimension HAmax. It requires that freestanding baseplates be rigid enough to limit parallel misalignment of the pump and driver shafts to 0.05 mm (0.002 in) when driver torque corresponding to full nameplate power is applied.

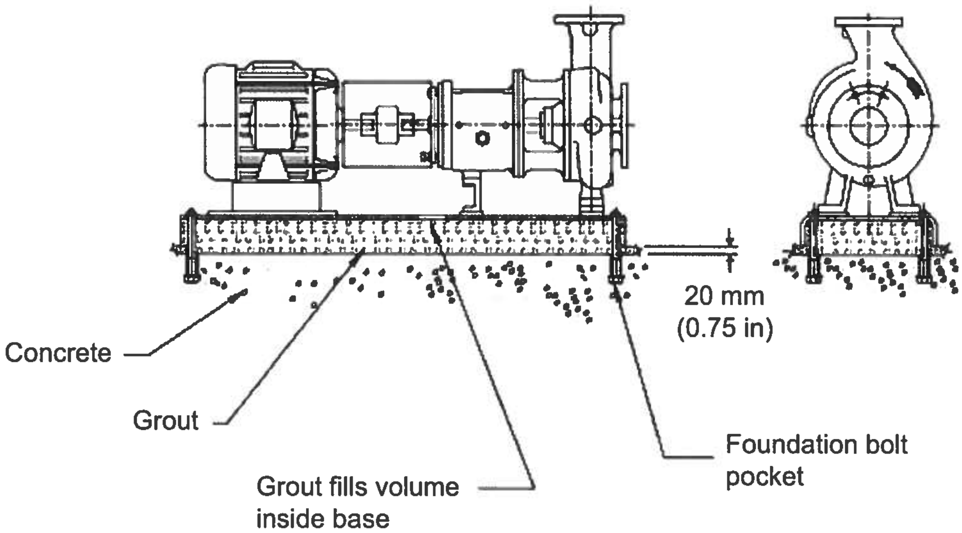

Figure 1.3.8.2.6.1 — ASME B73.1 pump and baseplate

1.3.8.2.6.2 ISO 2858 baseplates

ISO 3661 identifies baseplates for the ISO 2858 pumps and provides dimensions for those baseplates. Each baseplate is designed for a specific combination of pump size and IEC motor frame size and is identified by a number 2 through 9.

ISO 5199 includes additional information regarding the design of baseplates for ISO 2858 pumps. It requires baseplates to include provision for collecting and draining leakage of harmful liquids or of nonharmful liquids if specified by the purchaser. To facilitate drainage, it requires the top of the baseplate to be sloped at least 1/100, and a 25-mm (1-in) or larger drain connection to be located on the pump end of the baseplate. ISO 5199 requires non-grouted and freestanding baseplates to be designed to withstand specified external loads on the casing flanges without exceeding the shaft misalignment limits given in that standard.

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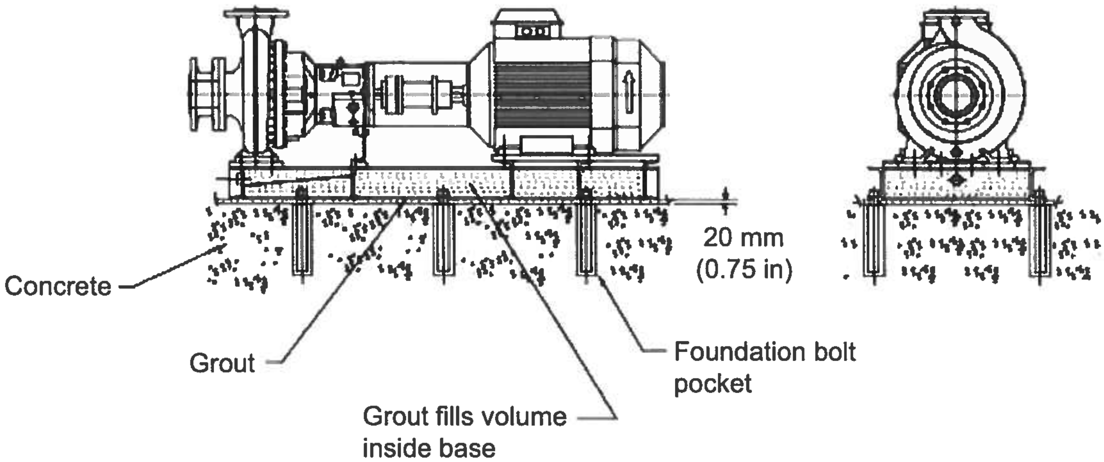

Figure 1.3.8.2.6.2 — ISO pump and baseplate

If baseplates are to be grouted, ISO 5199 requires them to be designed to facilitate grouting and have their undersides prepared for proper grout adhesion. Grout holes must be 100 mm (4 in) or larger in diameter and have raised edges if they are located in an area that collects leakage.

1.3.8.2.7 Nonmetallic baseplates

Polymer baseplates are made from nonmetallic materials, typically polymer concrete, epoxy, or fiberglass injected with vinyl ester. They are constructed in three basic styles, formed base, solid base, and pedestal base.

Some have pump and driver-mounting holes directly threaded into the polymer. Others have nonturning stainless steel inserts molded into the polymer.

The advantages of nonmetallic baseplates are

- Freedom from corrosion (with the exception of certain chlorinated solvents and concentrated sulfuric and nitric acids)
- Overall vibration of the equipment is less than a comparable ferrous baseplate, due to the improved damping of the material

Disadvantages are

- Operating temperature is generally limited to less than 300 °F (149 °C)
- Allowable pump nozzle forces and moments are less than for a metallic baseplate (except for pedestal bases), due to the lower rigidity of the material and the coefficient of friction between the pump feet and polymer bed being less than for two metallic surfaces

1.3.8.2.7.1 Formed baseplates

Formed bases are made of fiberglass injected with a polymer. The thickness of the form is from 13 to 25 mm (0.5 to 1 in). It may have a flange edge around the bottom perimeter to increase the rigidity. After the form is leveled over a reinforced concrete foundation it is filled with a nonshrinking grout.

These beds can be stilt mounted, but usually have a steel plate or polymer plate mounted to the bottom flange to give the bed extra rigidity.

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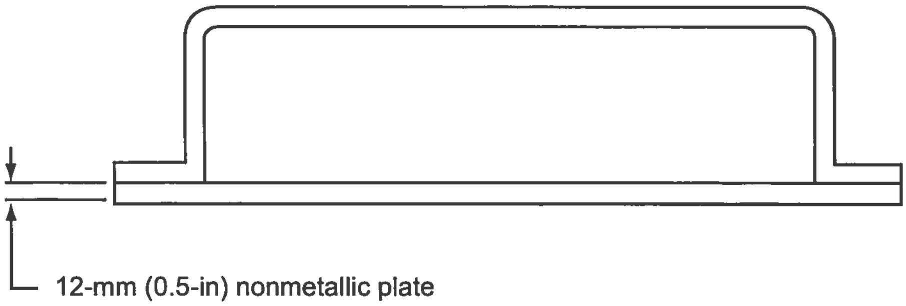

Figure 1.3.8.2.7.1 — Formed polymer baseplate

1.3.8.2.7.2 Solid baseplates

The solid base is always made from polymer concrete. The solid construction makes this design more rigid than the formed bed. Features such as an integral drainage slope can be cast into the baseplate.

These baseplates are usually grouted to the foundation. They can be mounted directly to a flat floor. These beds can be stilt mounted with additional bottom support.

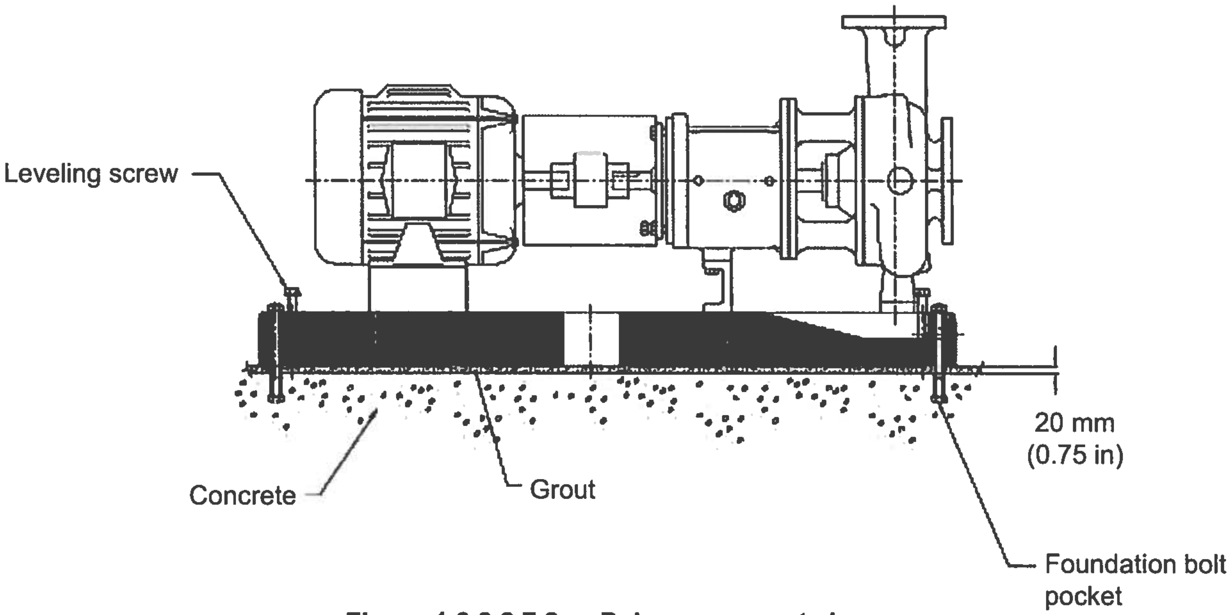

Figure 1.3.8.2.7.2 — Polymer concrete base

1.3.8.2.7.3 Pedestal baseplates

This design has rigidity similar to a ferrous baseplate and eliminates corrosion of anchor bolts. To accomplish this, the mounting surface is 0.5 to 1 m (1.5 to 3 ft) higher than the vertical height of the comparable metallic baseplate.

This design is basically a form that looks like a rectangular box with a slightly slopping side and no bottom. The thickness of the four sides and top is 50 to 75 mm (2 to 3 in). This pedestal box is made of polymer concrete and weighs 275 to 900 kg (600 to 2000 lb). The top of the pedestal has the mounting dimensions of ASME B73.1. The pump and driver can be premounted and aligned on the pedestal before the pedestal is put into installation position.

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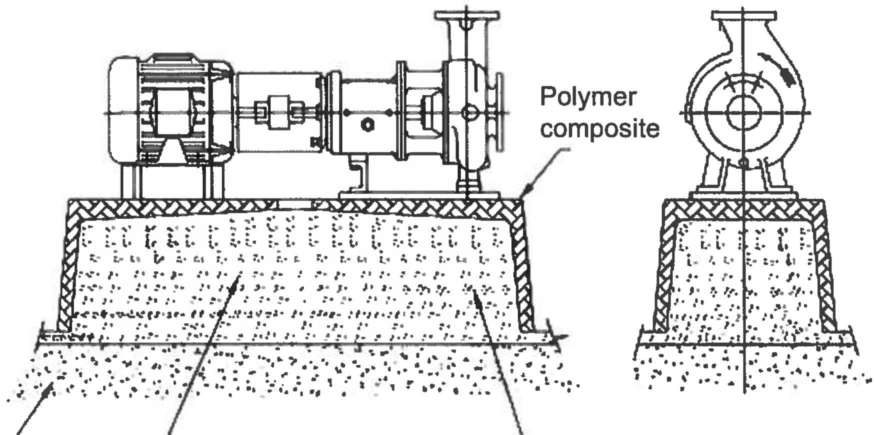

Figure 1.3.8.2.7.3 — Pedestal baseplate

1.3.8.2.8 Oil pipeline skid type baseplate

This is a baseplate design intended for installation in a remote location. Typically the base will contain the pump, driver, and auxiliary support equipment such as lubrication and seal supply systems. To allow the pumpset to be dragged into position, round hollow tubes are provided at each end of the baseplate. These serve the following two purposes:

- To provide an attachment point for the equipment used to drag the base into position
- To allow the base to slide over (as opposed to into), uneven ground

In some cases the baseplate may be provided with a tapered end (see inset) if required to cope with very uneven terrain.

It is the normal practice for such baseplates to be constructed in such a way as to be adequate for nongrout installation and operation. If the installation is considered permanent, then the end user may often grout the base in place.

1.3.8.2.9 Offshore skid type baseplate

A baseplate design intended for installation on an offshore platform or floating production storage and offloading platform (FPSO). Typically the base will contain the pump, driver, and auxiliary support equipment, such as lubrication and seal supply systems.

Typically these bases are large and require a high degree of stiffness to maintain the alignment of the pump and equipment installed on them. However, weight is often critical in the design of offshore platforms. It is common for weight limits for the total package to be specified by the customer. Penalty clauses may be applied to packages that exceed the weight limit.

The consequence of this is that these baseplate designs tend to use deep (but thin) sections to maximize stiffness for a given weight.

Bases may be attached to the platform by bolting or sometimes by welding of the skid to the platform.

This baseplate is always a nongrout type design.

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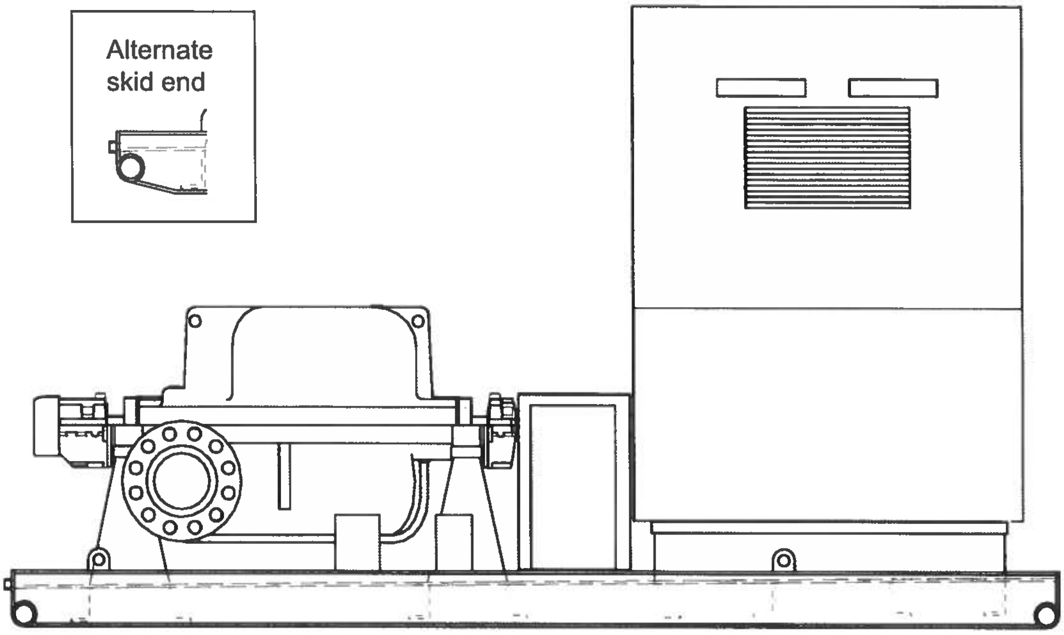

Figure 1.3.8.2.8 — Oil pipeline skid type baseplate

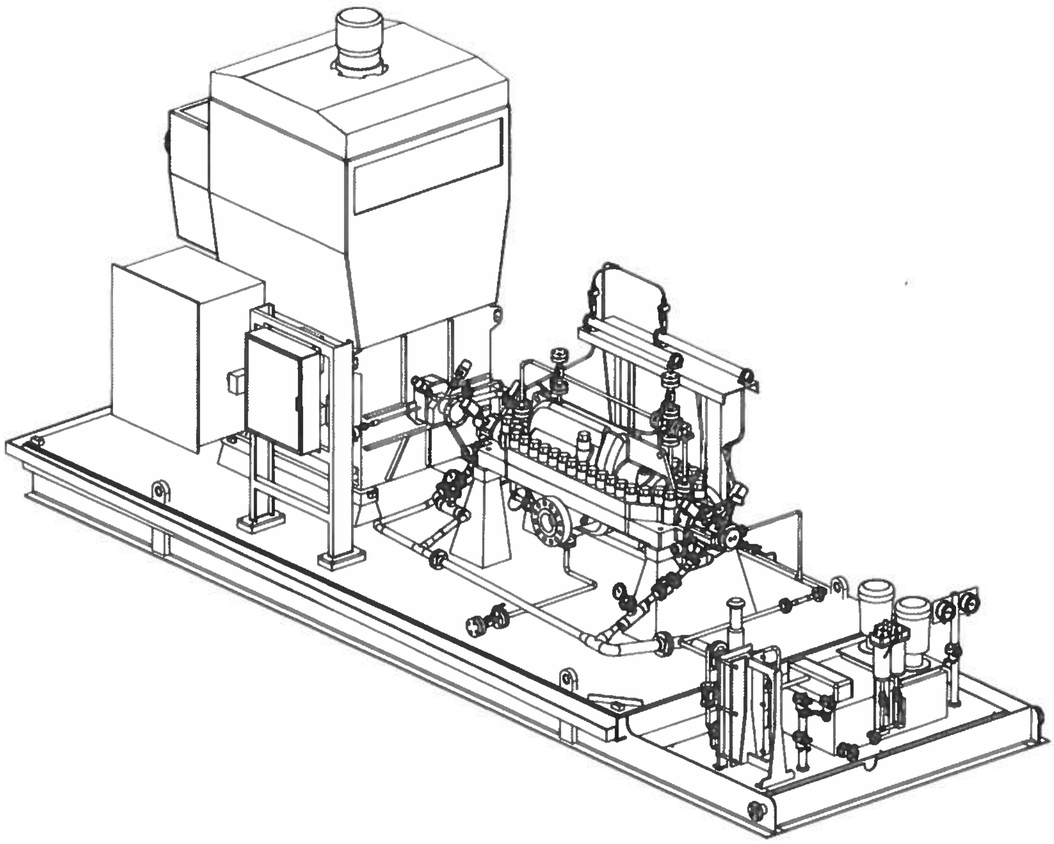

Figure 1.3.8.2.9 — Offshore skid type baseplate

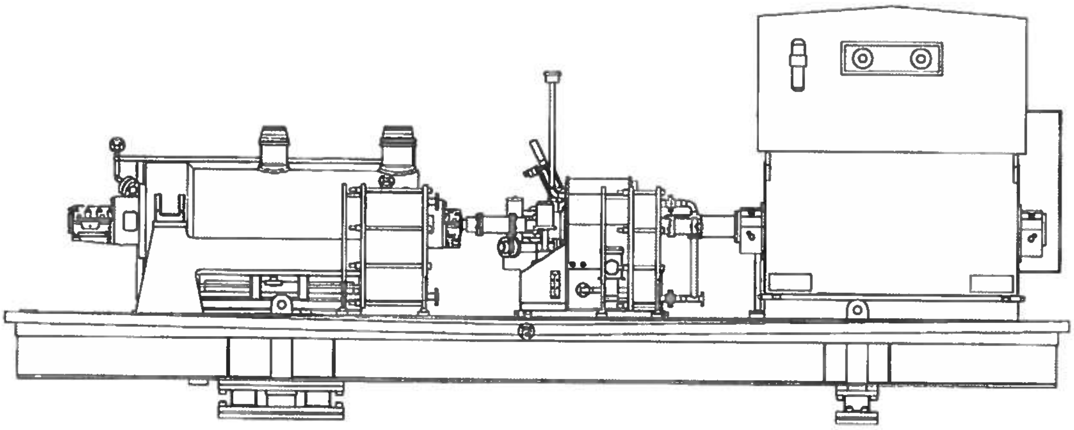

Figure 1.3.8.2.10a — Three-point mount type baseplate

1.3.8.2.10 Three-point mount baseplate

This baseplate design is intended to be elevated off the floor and supported at three points by stilts or gimbal-type mounts. It must be designed to provide its own rigidity, as there is no grout for additional support and only three points of support.

The design may be used when it is necessary to accommodate large deviations in altitude and position of the support structure or foundation.

Because this design is often used for pumps in critical, safety-related or unspared service, baseplate rigidity and strength are paramount. Customers may impose specific requirements with respect to allowable deflections or stresses and may require the submission of supporting stress analysis reports or other documentation.

See Figures 1.3.8.2.10a and 1.3.8.2.10b.

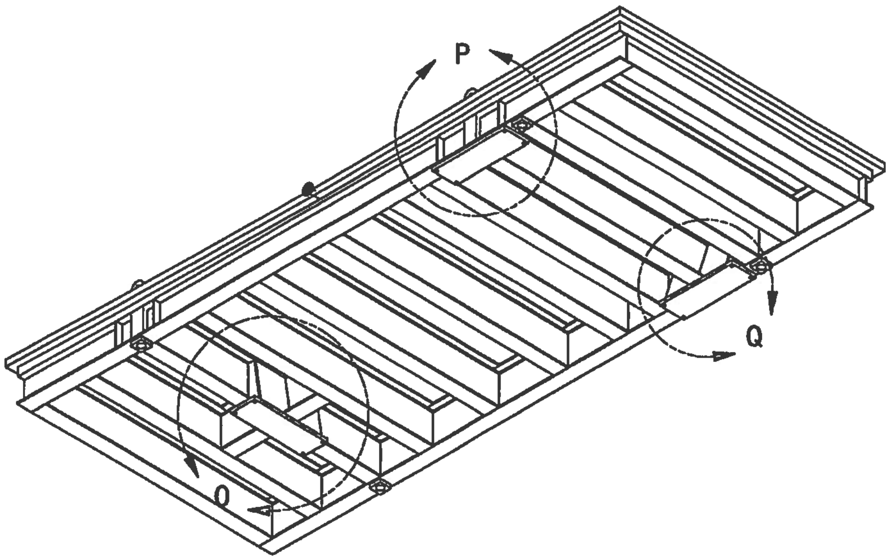

Figure 1.3.8.2.10b — Underside of the three-point mount baseplate (O, P, and Q show the mounting points)

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1.3.8.3 Tolerancing

Small pumps, less than 7.5 kW (10 hp), are often mounted on unmachined, light-duty baseplates. Heavier-duty small process pumps, less than 150 kW (200 hp), are often mounted on heavier-duty unmachined baseplates, which provide greater rigidity and more consistent mounting surfaces. Because of more precise coupling requirements for larger pumps, baseplates for pumps requiring 150 kW (200 hp) or greater are often mounted on heavy-duty machined baseplates. The critical dimensioning for baseplate design is as follows:

- Positional tolerance of equipment mounting and foundation bolt holes
- Equipment mounting height
- Equipment mounting pad flatness and parallelism

Driver size does not necessarily determine the class of baseplate. Installation procedures along with equipment size are factors taken into consideration when specifying the requirements. Excessively tight tolerances increase manufacturing cost, while excessively loose tolerances increase assembly or rework costs.

1.3.8.4 Stress and rigidity requirements

The analysis of the baseplate must cover two distinct areas of concern: transportation and operational loads.

During lifting the designer must limit the baseplate deflection and stress to prevent permanent deformation of the baseplate. It is normal practice to include a shock loading in the baseplate calculations of up to 3g[1] to accommodate for the possibility of excessive transportation loads.

If the lifting load criteria become impractical to meet, consideration should be given to shipping the driver separate from the base, and mounting it after the base is installed in the field.

Baseplates may be designed to be installed freestanding or to be grouted. A freestanding baseplate must be rigid enough to maintain coupling alignment when subjected to loads from piping or motor torque. The rigidity typically shall prevent no more than 0.25-mm (0.010-in) parallel coupling misalignment and 0.127-mm/mm (0.005-in/in) angular misalignment when subjected to maximum motor and piping loads simultaneously.

A grouted baseplate relies on the grout for a portion of its stiffness but must be sufficiently rigid to permit transportation. It is recommended that this type of baseplate have sufficient strength to prevent permanent deformation during transport. A check on the baseplate stress assuming support on three corners of the base and a 3g shock load is normally sufficient for this purpose.

1.3.8.5 Coupling alignment

Coupling alignment requires accurate location and flatness of equipment mounting surfaces and accurate drilling of driver, gearbox, and pump mounting holes.

Pump and driver mounting hole tolerancing must be carefully reviewed to ensure that coupling alignment can be attained. The required tolerancing on mounting holes is a function of the clearance between the mounting fasteners and the holes in the equipment feet, the distances between the equipment mounting holes, and the distances from the end of the driven equipment shaft to the driven equipment mounting holes. These relationships vary with different types of equipment, but tolerances greater than 0.75 mm (0.030 in) expressed in terms of true position, would be unusual. It is recommended that all tolerancing be done from one mounting hole.

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[1] g = gravitational acceleration.

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1.3.8.6 Shims and fasteners

A minimum of 3 mm (0.125 in) of shim pack thickness shall be provided under all drivers. Because of the pad height and parallelism tolerances, it is often necessary to design for a shim pack thickness of 9 mm (0.375 in) under the driver to ensure that the minimum of 3 mm (0.125 in) will be achieved.

Baseplates are not normally designed with shims under the pumps, although drivetrains involving three pieces of equipment may require shims under the pump for alignment. Good engineering practice limits the number of shims in a pack to five or less. Installation instructions shall emphasize the importance of minimizing shims.

Fasteners used in driver mounting holes should be 3 mm (0.125 in) in diameter smaller than the nominal mounting hole diameter where possible. The minimum fastener size for mounting equipment is 9 mm (0.375 in) except on 215 frame and smaller motors. When washers are used under fasteners, they shall be either hardened or extra thick.

1.3.8.7 Lifting requirements

When possible, provision should be made for lifting the base assembly with all equipment mounted. Lifting should be done from a minimum of four points. The four lifting points shall be as far apart from each other as possible to ensure stability, and in no event shall they be located axially inside the center of gravity of the pump or the driver. Preferred methods of lifting are

- Holes through the side of the base allowing the insertion and looping through of slings
- Holes in the side of the base for insertion of S-hooks
- Holes in or at the end plates of the base for insertion of S-hooks or shackles
- Lifting lugs attached to the baseplate siderails (longitudinal members)

1.3.8.8 Miscellaneous criteria

a) Baseplate vertical leveling screws are typically provided for larger pump sizes. Equipment mounting pads on the baseplate may be required to overlap the equipment feet to allow for leveling of the baseplate with the equipment installed. As pump feet are not typically provided with shims, leveling of the baseplate with the pump flanges should not be performed.

b) Motors and gearboxes may be required to be provided with jacking pads and screws to allow for movement of equipment during coupling alignment. Pads are located on the equipment mounting feet and are either of the welded-in-place or removable type.

c) Alignment blocks and/or guides may be required to be provided on the baseplate to allow for thermal expansion of the pump case. This requirement depends on pump fluid temperature and pump type selected.

d) Epoxy grouting of baseplates requires special paint preparation on all surfaces in contact with grout to ensure an adequate bond is achieved.

e) To collect and drain any spillage of fluid that may otherwise contaminate the area around the baseplate, two common design options are typically used. Depending on customer requirements, either a sloped deck plate underneath the collection area or a drip rim around the collection area, can be provided. In either case a tapped opening is provided at the low point for piping away the collected fluid to safe drainage. A nonskid deck plate may be used to prevent accidental injury due to such spillage.

f) Fasteners used to hold equipment feet to the baseplate shall either be threaded into material whose thickness is at least equal to the thread diameter, or use nuts on the underside of the base. If material thickness equal to

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one thread diameter cannot be achieved, an analysis of the joint must be made to determine if the necessary clamping force can be achieved with the thinner material, and to what torque the fastener shall be tightened.

g) Grout holes, at least 100 mm (4 in) in diameter or 90 mm (3.5 in) square must be provided and located to allow complete grouting of the base. The holes shall be spaced such that grout is not required to be forced more than 760 mm (30 in). Vent holes, a minimum of 12 mm (0.5 in) in diameter, shall be provided at the end of all grout runs. Provision shall be made for a minimum grout thickness of 25 mm (1 in).

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Appendix A

Introduction to pump classifications, industry segments, and general applications

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 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), or 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

A.2 Pump industry segments and general service applications

Identifying the most appropriate pump configuration for a specific application requires specialist knowledge relevant to the industry segment served.

The following is a list of industry-specific application categories established as a guideline.

Pump industry segments

- Chemical Industry
- Petroleum

  Oil production
  Oil and gas transportation (pipeline)
  Hydrocarbon processing

- Pulp and paper
- Slurry
- Municipal water and wastewater

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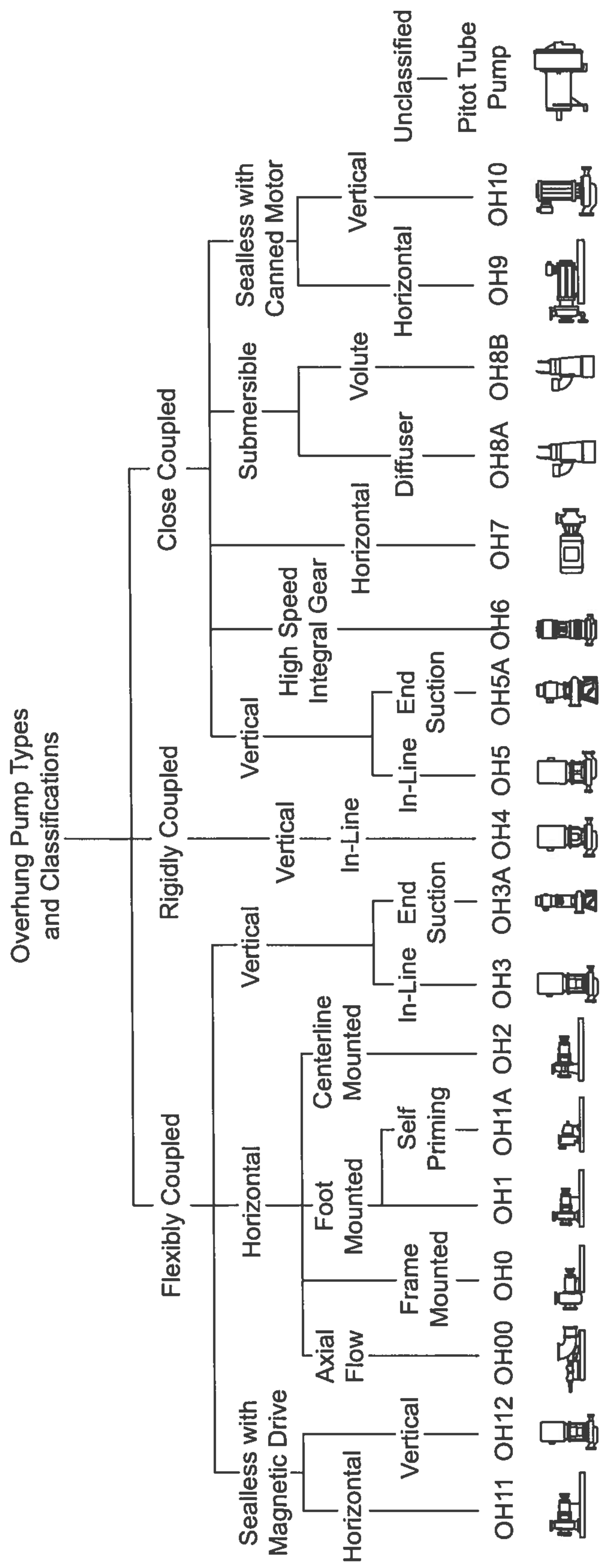

Figure A.1 — Overhung pump types and classifications

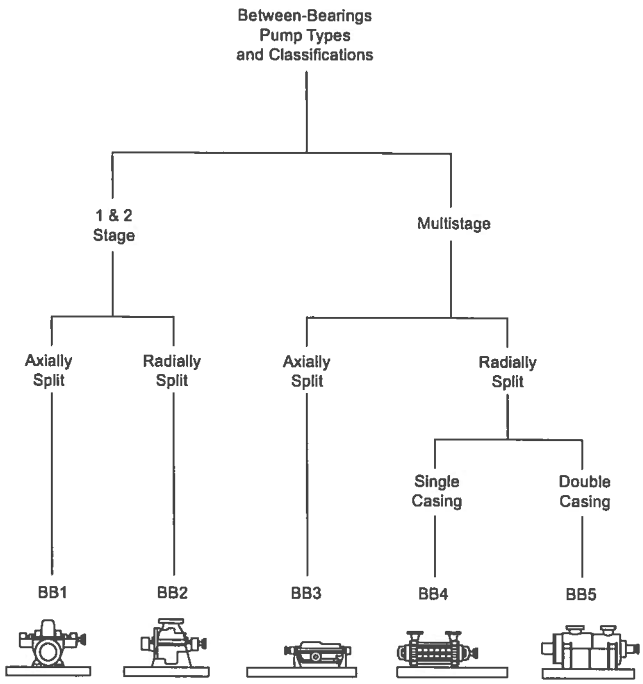

Figure A.2 — Between-bearing pump types and classifications

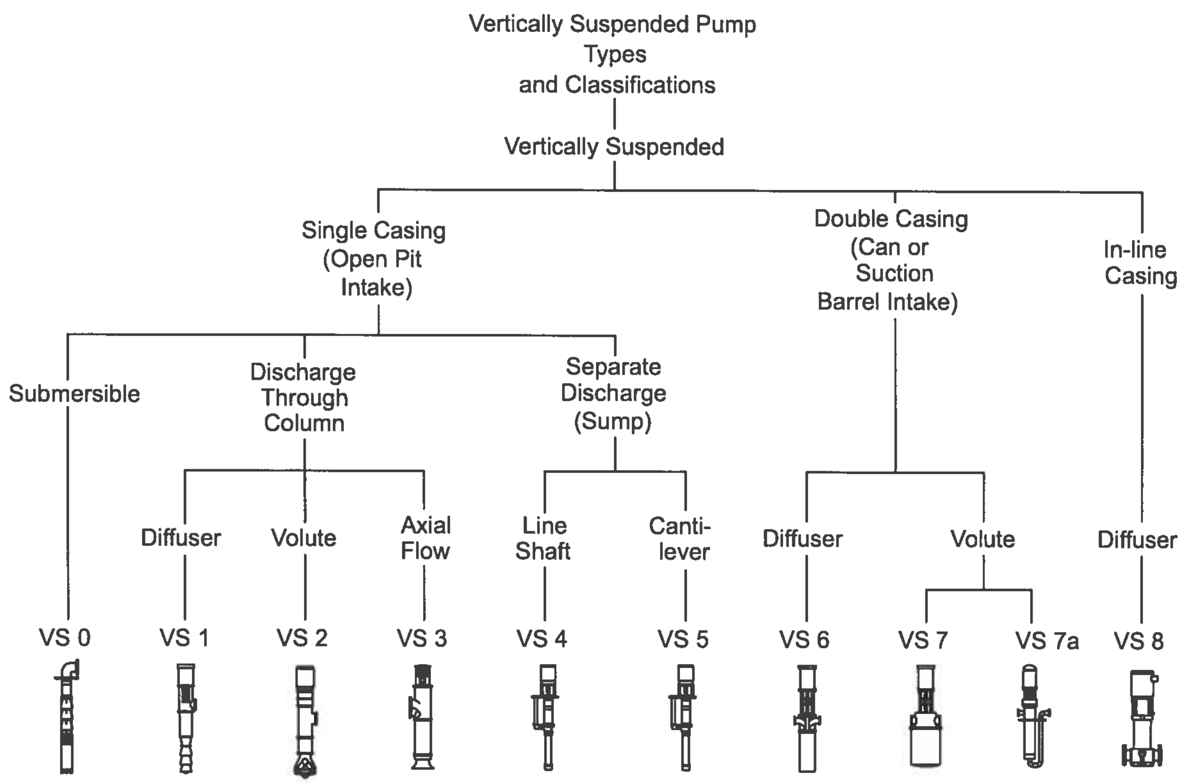

Figure A.3 — Vertically suspended pump types and classifications

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- Irrigation applications
- Residential applications
- Electric power industry
- Cooling tower
- Fire pumps

A.2.1 General service applications

Within the above application categories, many services are described that are not unique to one segment of industry. For this purpose, Appendix A is included to cover numerous aspects of general service applications and operational aspects that are commonly of interest across a variety of industry segments. Principal features of pumps and the necessary precautions for proper use are pointed out. Some elements of pump service are also covered by this category. These services cover a diverse range of commonly encountered pump application issues.

A.2.2 Transfer pumping

Overview

This service is normally a part of a process where a fluid is transferred from one location or process to another in a given plant.

Services

There are many services that require transfer of fluid.

Typically the requirement is for a low developed pressure but high flow rate.

Fluids handled

Almost all fluids may be encountered in this application. The limitations on fluid are determined mainly by the design of pump used.

Pump types used

Almost all rotodynamic pumps are suitable and capable of operating in transfer service.

The most common pump types used in this type of application are single-stage overhung (OH) and single-stage between-bearings pumps (BB1, BB2).

Special features

The type of pumps used will vary with the duty involved as well as construction details and materials of construction.

Accurate specifications of the liquid characteristics and the range of suction pressures expected must be provided with rate of flow and total head for the pump manufacturer to make a proper selection.

Pumps used in the transfer of fluids from tanks need to be designed for the suction conditions that occur when the fluid in the tank is at its lowest level. To prevent the pump from losing suction, adequate control systems need to be in place to prevent drawdown of liquid beyond the safe design point of the pump.

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A.2.3 Booster service

Overview

Rotodynamic pumps in this service handle liquids piped to them at various levels of pressure, normally above atmospheric, and discharge at a higher pressure into the system.

Services

There are many services that require a boosting of system pressure.

A common application is to use the output of a booster pump to feed into the suction of a second more powerful pump. A booster pump is typically selected for a lower NPSHR than the higher-pressure pump it feeds. The second pump provides sufficient suction pressure so that it can be optimized for efficiency, operating range, or other parameters.

Booster pumps are also used in overland pipelines so that the operating pressures can be controlled within pump casing and piping limitations.

Fluids handled

Almost all fluids may be encountered in this application. The limitations on fluid are determined mainly by the design of the pump used.

Pump types used

Almost all rotodynamic pumps are suitable and capable of operating in booster service.

The most common pump types used in this type of application are single-stage overhung (OH) and single- or multi-stage between-bearings pumps (BB).

Special features

Accurate specifications of the liquid characteristics and the range of suction pressures expected must be provided with rate of flow and total head for the pump manufacturer to make a proper selection.

Booster pumps used to provide the suction pressure to a second pump (pumps in series) need to be designed so that their performance matches the requirements of that second pump.

A.2.4 Pumps used as hydraulic turbines

Overview

Rotodynamic pumps may be operated 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.4. The applied head is usually constant, so the other parameters are shown as they vary with speed.

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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.

Services

A common application is a hydraulic power recovery turbine (HPRT). The potential for power recovery from high-pressure liquid streams exists any time a liquid flows from a higher pressure to a lower pressure in such a manner that throttling occurs.

Reverse running pumps are used instead of throttling valves to recover the power in the high-pressure liquid. The installation cost for a HPRT is about the same as for an equivalent pump, and reliability and maintainability are also comparable. Because the efficiency of a pump operating as a turbine is comparable to the pump efficiency, the use of reverse running pumps as primary or secondary drivers becomes quite practical.

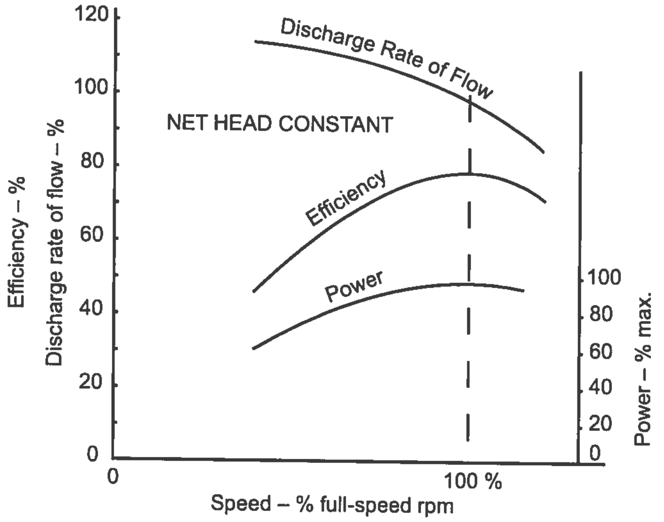

Figure A.4 — Turbine characteristics

Pumps operating as turbines are classified by their turbine specific speed (NST), which is a quantity that governs the selection of the type of runner best suited for a given operating condition.

$$NST = \frac{n \times (Pt)^{0.5}}{(Ht)^{1.25}}$$

Where:

NST = turbine specific speed

n = revolutions per minute

Pt = developed power, in kW (hp) at best efficiency point (BEP)

Ht = net head, in m (ft) per stage across the turbine

The values of NST will be slightly different between a pump operating as a pump and the same pump operating as a turbine. The rate of flow and total head at BEP will be greater for the turbine operation than for a pump operation. The amount of shift from pump performance depends on the specific speed and other design factors.

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

The power output is the rotational energy developed 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

$$Pt = 2.725 \times Q \times H \times \eta_t \times s$$

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US customary units

$$Pt = \frac{Q \times H \times \eta_t \times s}{3960}$$

Where:

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

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

H = total head, in m (ft)

s = specific gravity

η_t = efficiency of the turbine

Fluids handled

In theory almost all fluids may be used in a HPRT application. The limitations on fluid are determined mainly by the design of pump used. However, for good results, the fluid should preferably be limited as follows:

- Low viscosity (approximately the same as water)
- No abrasive particles
- Vapor pressure similar to or less than water

Pump types used

Almost all rotodynamic pumps are suitable and capable of operating as turbines.

The most common pump types used in this type of application are single-stage overhung (OH) and between-bearings pumps (BB1 and BB2). Less frequently, back-to-back multistage pumps (BB3 and BB4) may be used.

Special features

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 PAT 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 subject to a careful calculation of combined stresses in shafts.

Because of the reverse rotation, be sure that the bearing lubrication system will operate in reverse, and threaded shaft components, such as impeller locking devices, cannot loosen.

Pumps operated in reverse as turbines tend to have relatively narrow operating bands compared to variable nozzle turbines. At constant speed, the power developed and efficiency drop to zero at approximately 40% of the hydraulic turbine best efficiency rate of flow. See Figure A.5. Energy must be added to the hydraulic turbine for it to rotate at the constant speed below this rate of flow.

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Changing the 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, results in some uncertainty when applying a pump to a power recovery turbine application unless actual test data are available on the specific pump running in reverse as a turbine.

Precautions should also be taken to ensure that the PAT will operate without cavitation. The turbine industry typically uses the terminology *total required exhaust head* (TREH) and *total available exhaust head* (TAEH) in place of NPSH. Total exhaust head is defined as the total fluid energy at the impeller eye less the vapor pressure of the fluid. Some of the other factors that affect the use of pumps as turbines are:

- Runaway speed
- Rate of flow at runaway speed
- Required solids passage
- Fluidborne abrasives
- Torque reversals during startup or shutdown
- Overspeed trip and control

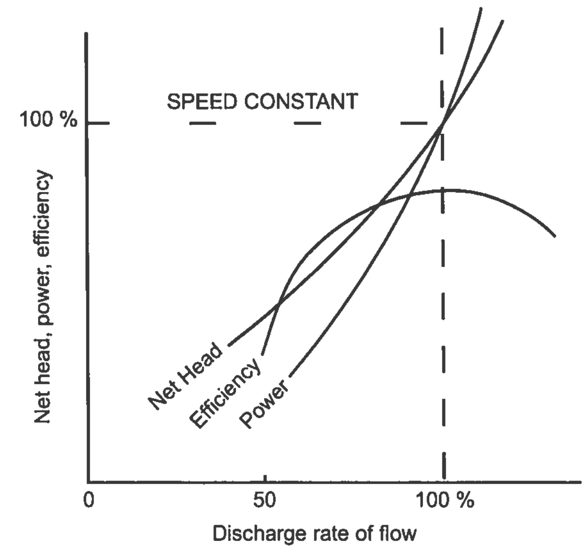

Figure A.5 — Turbine performance

A.2.5 Dry pit (nonclog) pumps

Overview

Rotodynamic centrifugal pumps are used extensively in water and wastewater applications. Many applications are particularly challenging due to the wide variety of liquids, gases, solids, and living microorganisms that are present in the fluid being pumped. There are a wide variety of services and thus many pump types, ranging from submersible to large dry pit pumps. Wastewater services generally involve the handling of large solids, which in turn, demands unique design solutions including special impellers with large passages to prevent clogging. Dry pit pumps are configured for these applications in several different ways depending on installation preferences.

Services

Water service applications can be considered as those which provide for the transport of pure or slightly contaminated liquids. Such applications include drinking water supply systems, irrigation and drainage plants, and cooling water circuits. Wastewater services can be divided into waste collection and waste treatment. Waste collection requires pumps that are designed to ensure maximum freedom from clogging when handling liquids containing solids or stringy materials. These are commonly called *nonclog pumps*. They are also designated as *wastewater*, *sewage*, or *trash pumps*. Waste treatment pumps may be required to deal with a combination of treatment chemicals along with all the waste.

Fluids handled

Dry pit (nonclog) pumps are recommended for handling raw or unsettled sewage, activated sludge, industrial wastewaters containing solids, and similar liquids where excessive clogging would otherwise be encountered. The largest solid sizes that the pump will be required to handle in normal operation must be specified. The term *sphere size* denotes the largest diameter ball that can be passed through the pump.

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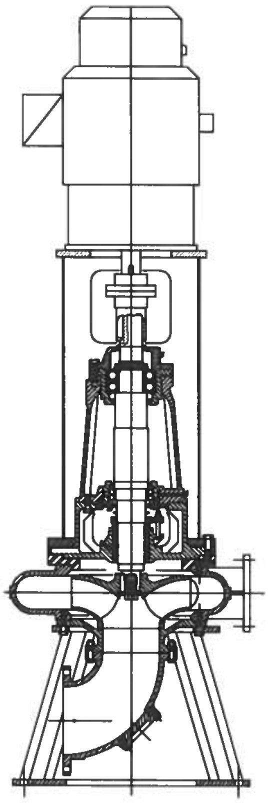

Figure A.6 — Dry pit pump — overhung flexibly coupled vertical end suction

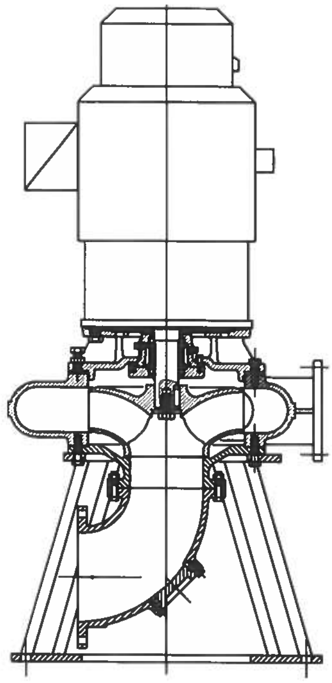

Figure A.7 — Dry pit pump — overhung close coupled vertical end suction

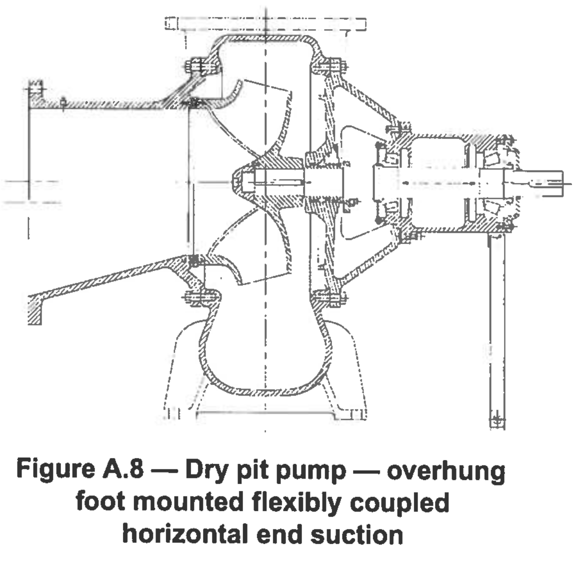

Figure A.8 — Dry pit pump — overhung foot mounted flexibly coupled horizontal end suction

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Pump types used

Dry pit pumps come in a wide range of sizes and may utilize standard motors. They are available in horizontal or vertical configurations for space-saving considerations. Pump types OH1, OH3A, and OH5A are generally applicable. These are single stage, overhung, horizontal foot or frame mounted; vertical frame mounted, including close coupled, flexibly coupled, or independently mounted motor with extended universal joint shaft drive. Nonclogging impellers are commonly used in all configurations.

Operating parameters

These pumps can be quite large in capacity and dimension.

- Flow rates are possible in excess of 45,000 m^3/h (200,000 US gpm)
- Heads to 90 m (300 ft)
- Discharge nozzle sizes 75 mm (3 in) to 1800 mm (72 in)

Special features, characteristics, or considerations

Dry pit pumps are popular due to the ease of maintenance, and easy diagnosis of pumping problems. For sewage applications, the pump can be serviced without pulling the pump from a pit and thus less decontamination is required. Comminution[1] and/or adequate bar screens must be provided to prevent large solids from entering the pump. When used, bar screen openings should be sized to prevent clogging from irregularly-shaped solids. For sewage service, pumps built to the manufacturer's material specifications are ordinarily used.

Corrosion-resistant and wear-resistant shaft sleeves and wearing rings are desirable for maximum life. Inspection openings in the casing or adjacent piping, for access to the impeller, are recommended. Stuffing boxes may be furnished with mechanical seals or packing, either water or grease lubricated. When water is used for the stuffing box or wearing ring lubricant or flush, the supply line must be isolated from any potable water system.

A.2.6 Self-priming pump applications

Overview

Self-priming pumps are designed to have the following abilities: to prime themselves automatically after being initially filled, when operating under a suction lift; to free themselves of entrained gas without losing their prime; and to continue normal pumping without attention.

Services

Self-priming rotodynamic pumps are frequently used for unattended service in industrial, construction dewatering, wastewater, and agricultural applications where manual priming is not practical during operation.

Most pumps are designed to accept the major modern drive sources through direct drive, flexible coupling, or belt drives.

Self-priming centrifugal pumps are commonly placed up to about 8 m (25 ft) above the liquid level of the source. During initial startup, the impeller rotation causes the liquid in the impeller and suction side of the pump reservoir to be forced to the discharge cavity. Differential pressures cause the priming recirculation to start. The priming action reduces pressure in the impeller eye and allows atmospheric pressure on the liquid source to fill the suction line.

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[1] Comminution is one of the four main groups of mechanical processing and describes the movement of the particle size distribution (grains, drops, and bubbles) into a range of finer particle sizes. (The other groups are agglomeration, separation, and mixing.)

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Fluids handled

These pumps most commonly operate with water at ambient temperature.

In the case of chemical industry pumps, other liquids may be involved and are dependent on the process. Near-boiling liquids or liquids with high vapor pressures are not good choices for self-priming applications as these liquids limit the amount of suction lift possible.

Pump types used

Pumps in this class are usually of the OH1A design, having single inlet overhung impellers. Self-priming rotodynamic pumps have additional design features when compared to conventional rotodynamic pumps. These features are required to perform the self-priming function and are described in the special features section.

The materials of construction are usually based on the application. Many manufacturers of self-priming pumps build units to conform to the specifications established by the Contractors Pump Bureau (CPB), an arm of the Construction Industry Manufacturers Association (CIMA). See Figure A.9. In these cases, the pumps may carry a CPB rating decal and are built to conform to CPB specifications. Other manufacturers build pumps that are closer in concept to the chemical industry requirements and appear as shown in Figure A.1.

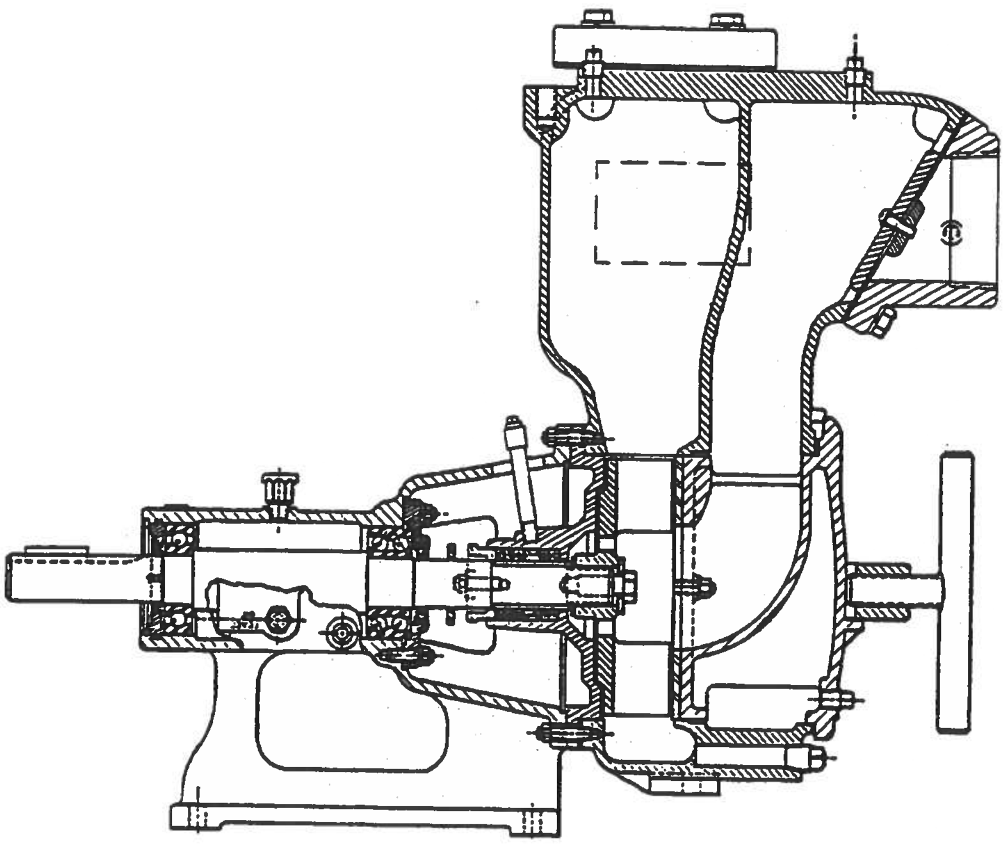

Figure A.9 — Self-priming pump — construction industry

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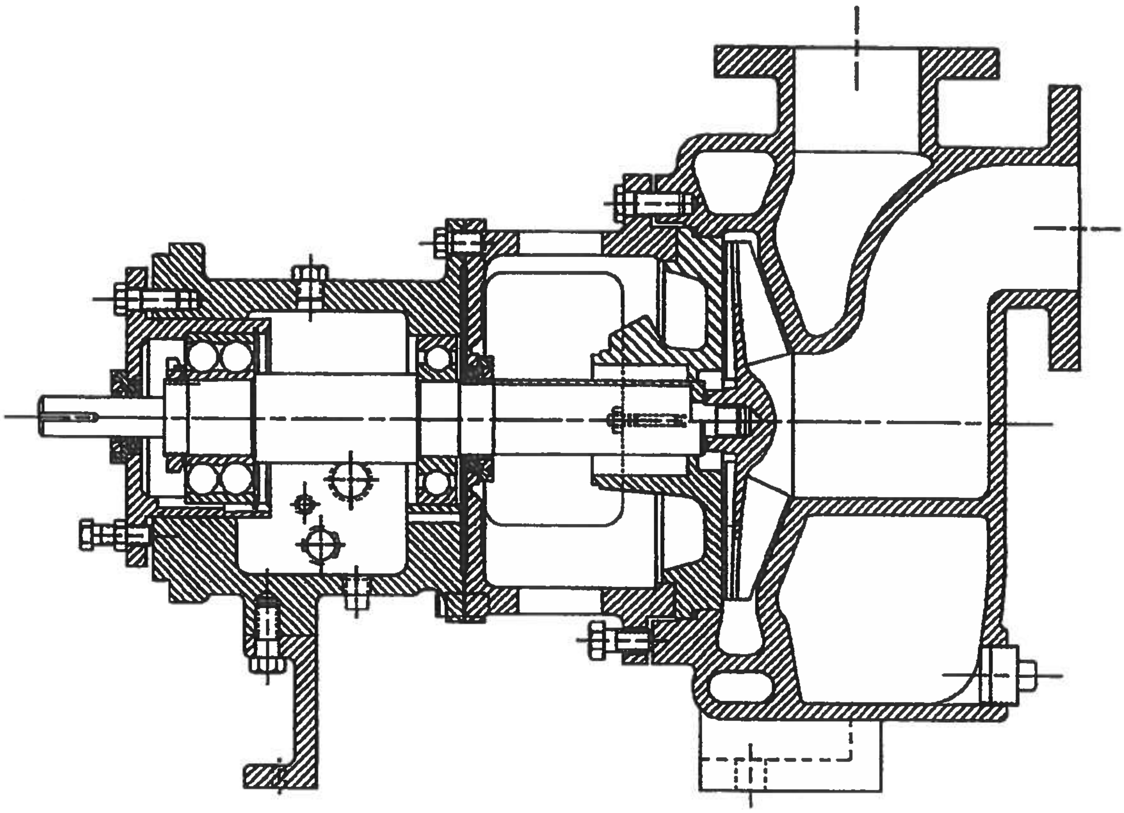

Figure A.10 — Self-priming pump — chemical industry

Special features

All self-priming pumps should have these three features:

1) A reservoir, either integral with or external to the volute and impeller, to retain priming liquid. This reservoir is filled during the initial prime of the pump. The suction line itself is not filled. When the pump completes a pumping cycle and shuts down, the reservoir retains liquid for the next priming cycle.

2) A means of recirculating liquid. The majority of modern self-priming pumps have integral reservoirs with internal recirculation, the most common being the recirculation from the pressure side of the volute back to the periphery of the rotating impeller. A less common method of recirculation directs the priming liquid back into the suction side of the pump.

3) An integral suction check valve to prevent the loss of liquid in the suction leg is common in many designs. Some designs allow for check valve failure due to debris-laden water and will reprime with residual amounts of priming liquid. Self-priming can also be accomplished in diffuser design centrifugal pumps used primarily for clear liquids. Closed impellers with suction side wearing rings can also be used.

During priming, air in the discharge chamber separates up and out from the mixture while the heavier water continues to recirculate. It is important that the air in the discharge chamber have a means to escape either through the discharge pipe or an air release valve. This process continues to draw air from the submerged suction line until it is full of liquid and the pump goes to normal pumping.

The different designs of self-priming pumps have limited and varied capabilities of priming against discharge heads. When a discharge check valve is used, or the discharge design can form a pressure trap, air release lines or valves may be necessary to get rid of air from the suction side.

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A.2.7 Two-phase pumping applications

Overview

Two-phase flow pumping applications include situations where undissolved vapors or gases are being carried by the pumpage. The ratio of gas to liquid and flow velocity determines the composition of the two-phase flow. Depending on these parameters, flow may be in the form of a mist, a slug, or bubbles of gas contained within the liquid.

Services

Two-phase flow is often found in 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. Depending on the wellhead conditions, suction pressure at the pump can vary between 20 to 13,000 kPa (3 to 1885 psi), although pressures in the range of 1000 to 2000 kPa (150 to 300 psi) are more typical.

Other applications are those where 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.

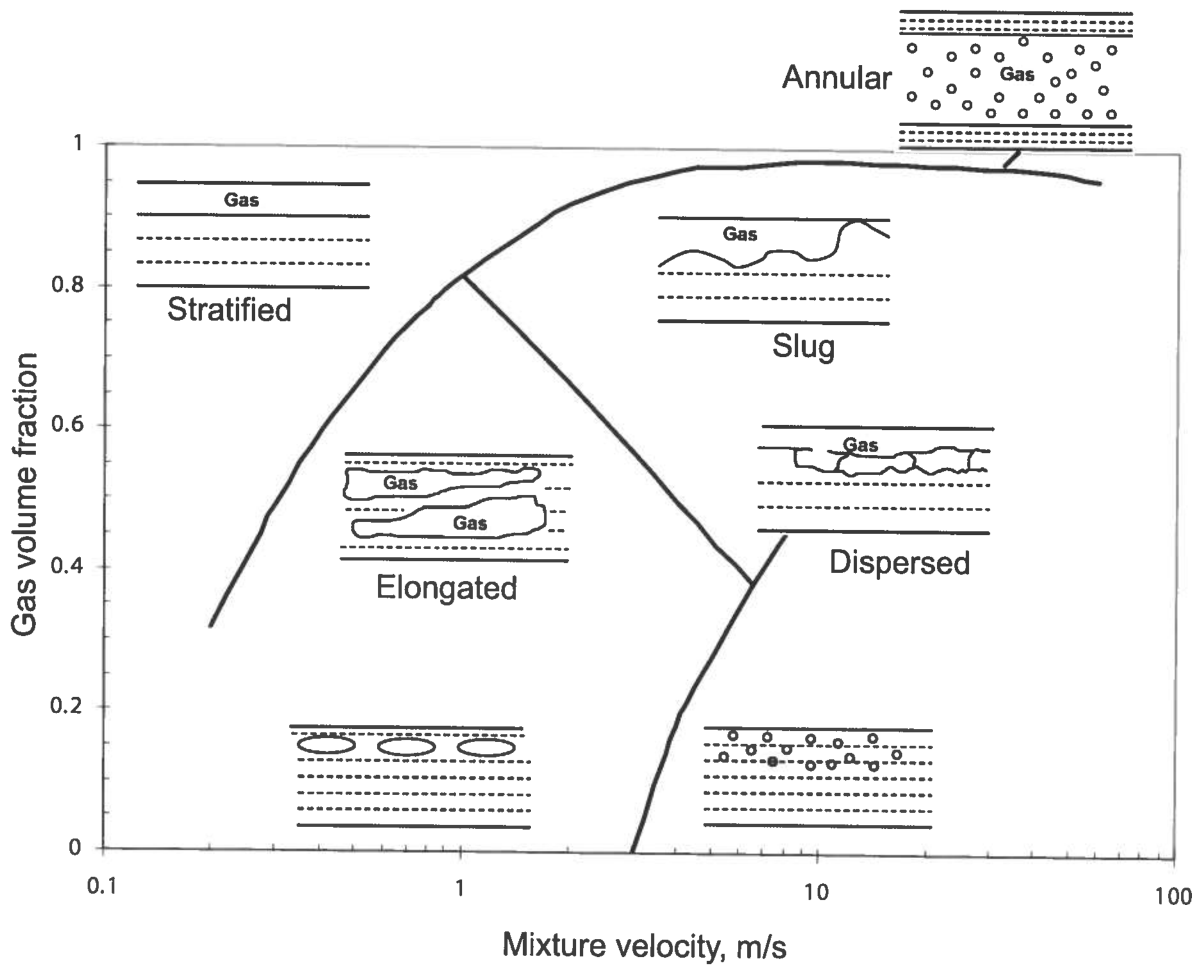

Figure A.11 — Two-phase pumping applications

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Fluids handled

Almost any liquid/gas combination can exhibit two-phase flow. Fluid/gas combinations commonly encountered are

- Water/air
- Oil/liquid natural gas (LNG)

Pump types used

Self-priming pumps (Figures A.9 and A.10) can handle higher amounts of gas without becoming completely clogged (airbound), especially at low rates of flow. These units will not become airbound because they will revert to the priming mode if necessary and evacuate the air on the suction side. However, they will suffer reductions in rate of flow and head similar to conventional rotodynamic pumps when operating at higher rates of flow with entrained gas.

Helico-axial pumps (Figure A.12), while not true centrifugal pumps (since the flow through them is by definition axial in nature), are able to handle large amounts of gas due to the optimization of the rotor and stator elements for this purpose. This type of pump can handle up to 80% gas or even higher.

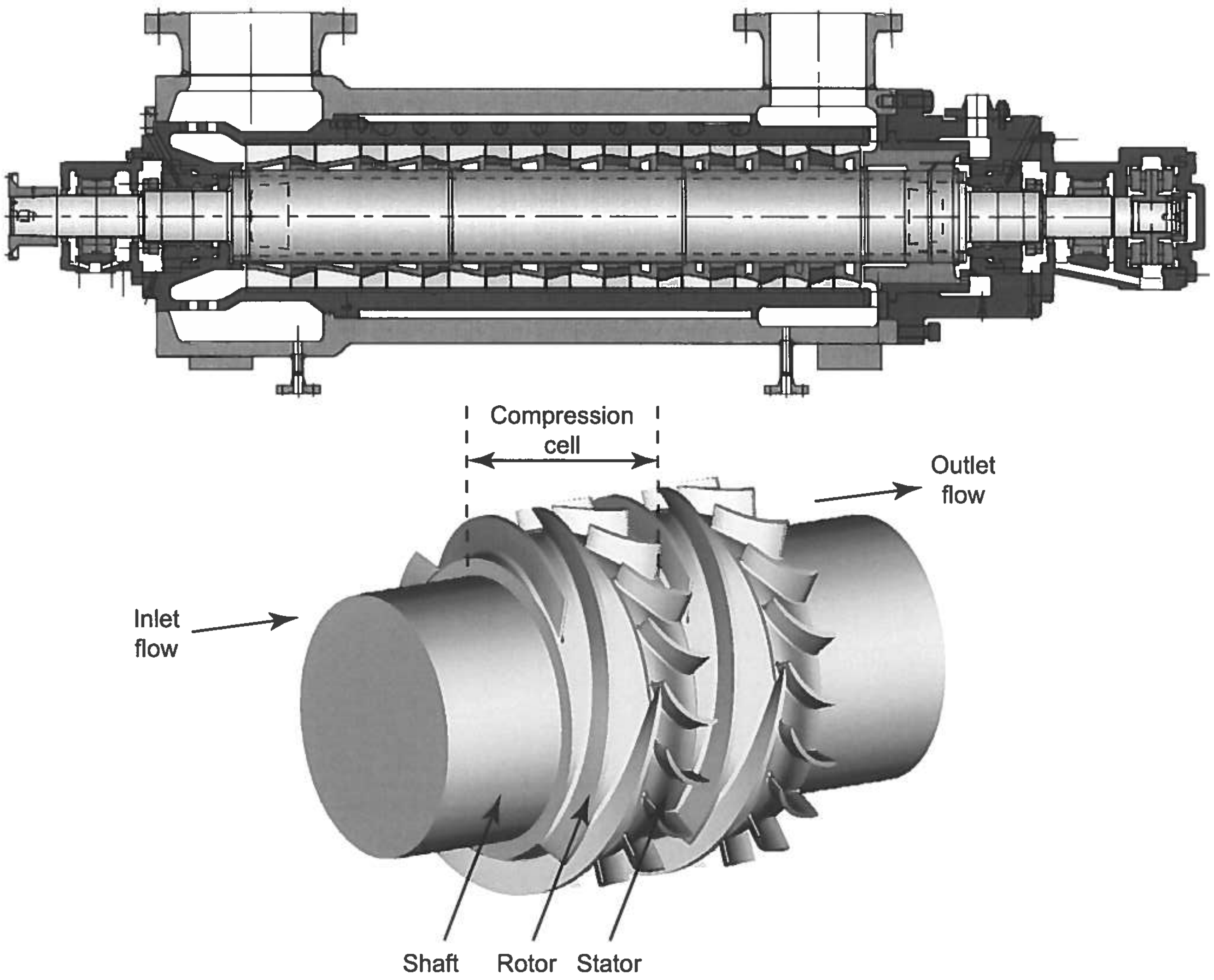

Figure A.12 — Helico-axial multiphase pump

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Screw-type pumps (Figure A.13) and progressing cavity pumps can also handle large amounts of gas (up to around 98% gas) because they use positive displacement and are not subject to the problems caused by the complex fluid dynamics associated with the centrifugal pumping action. The single screw pump is particularly suitable in that it can handle low-viscosity liquids. Many other rotary units, such as gear and vane pumps, depend on the viscosity of the liquid being pumped to lubricate the moving parts.

Special features

The most dramatic effect of gas or vapor on rotodynamic pump performance is the complete blocking of the impeller inlet as the pump becomes "airbound." When this happens, the impeller acts as a centrifuge, and tends 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 the air 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 discharge pressure are reduced (Figure A.14). This reduction is the result of the blockage of the flow by the gas, and a reduction in developed pressure due to the reduced specific gravity of the pumped mixture. When the specific gravity of liquid alone is used to convert pressure to head, a lower head measurement is indicated.

It can be seen from Figure A.14 that even with small percentages of gas, the unit stops pumping liquid due to accumulated air in the impeller when operating near the shut-off condition of the pump. High velocities at higher rates of flow can carry with it 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.

Inducers or inlet boosters are devices designed to benefit the functioning of the impeller in that they increase the fluid pressure before the mixture enters the pump. This increase in pressure reduces the volume of the gas, thereby reducing its negative effect on the impeller performance. Since inducers generate low levels of pressure, they will have little benefit on high suction pressure applications.

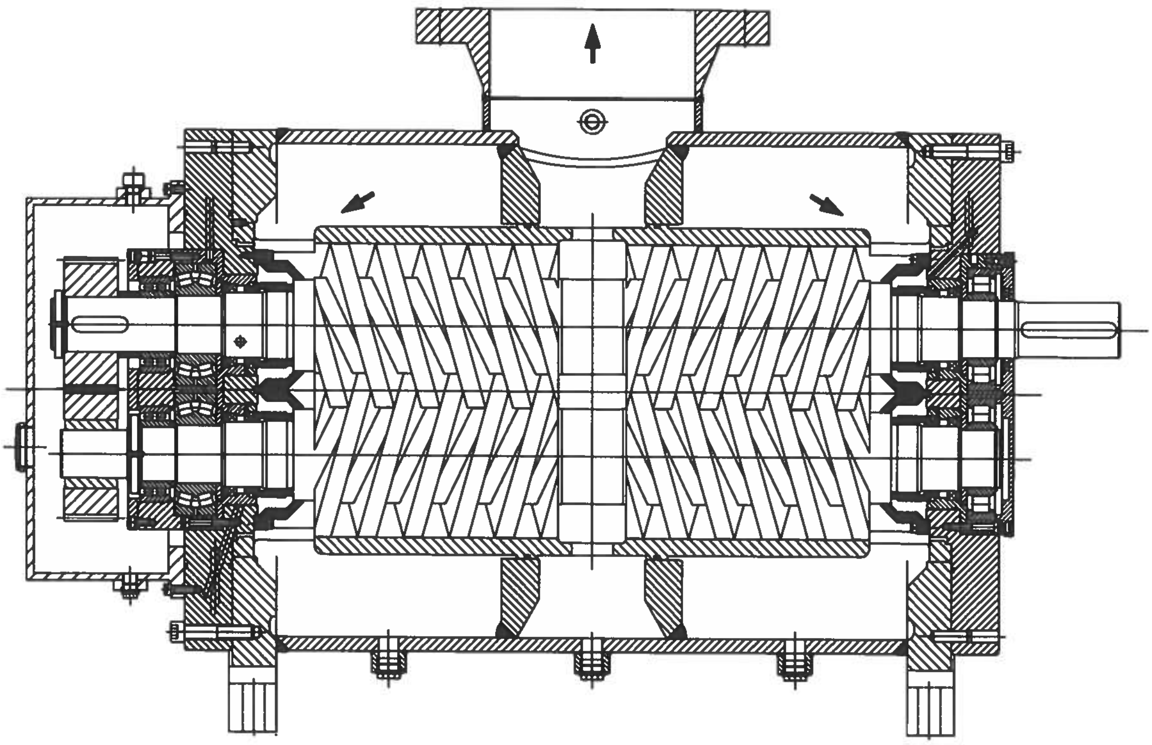

Figure A.13 — Screw-type multiphase pump

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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]). In some cases, it may be helpful to use a high specific-speed booster pump in series with a low specific-speed pumping unit in order to minimize the effect of the gas.

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

Another helpful action is to provide a gas vent at the pump inlet (see Figure A.15). The suction pipe should be sized about twice as large as the flange at the pump inlet 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.

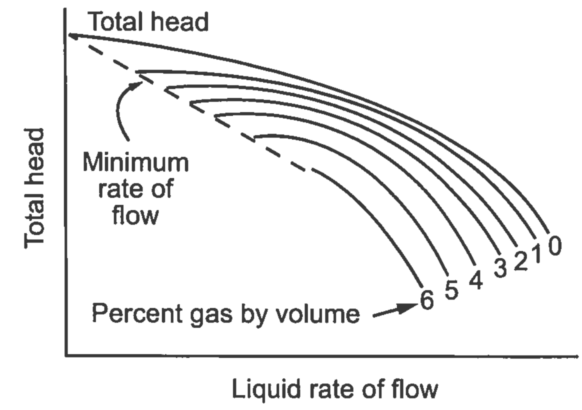

Figure A.14 — Effect of gas on pump performance

Figure A.16 shows a top suction impeller. This design has the advantage of being self-venting, which prevents large quantities of gas building up in the impeller eye. Excess gas moves towards the top of the pump casing where it can be vented in a controlled way.

If the pump takes suction from a closed tank, it may be possible to pressurize the inlet, thereby reducing the volume of entrained gas, or turn some vapors back to liquid. Where vapor is the primary problem, subcooling of the inlet pipe may be helpful. This will also tend to turn vapor back to liquid, and thus reduce the volume of free vapor that must be handled by the pump.

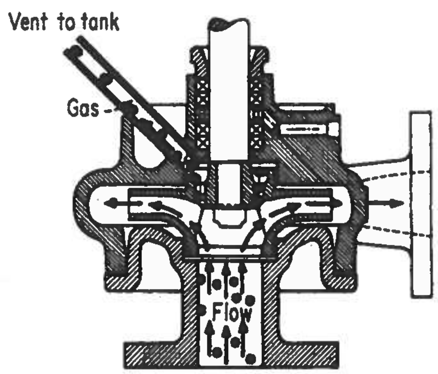

Figure A.15 — Venting the eye of the impeller

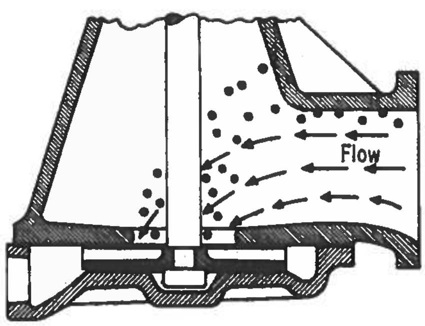

Figure A.16 — Top suction impeller

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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 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.

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

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

B.1.1 Regenerative turbine pumps

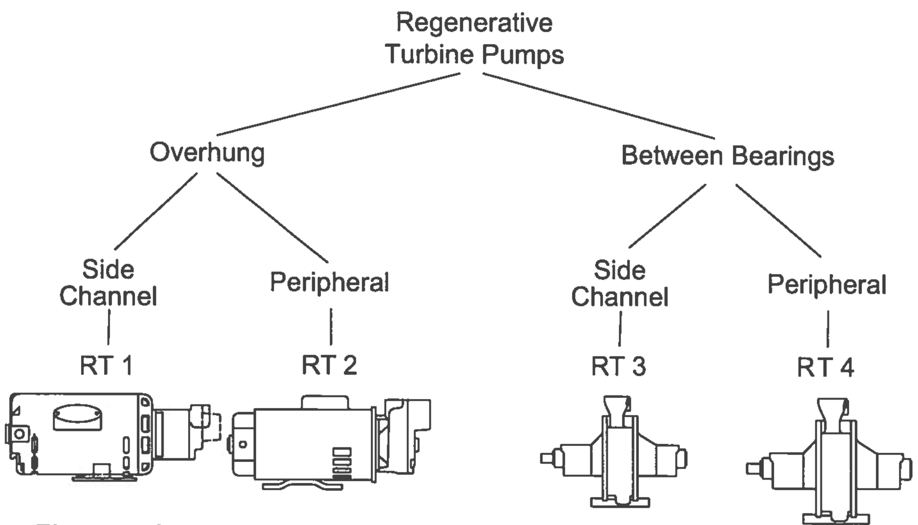

Figure B.1 — Regenerative turbine pumps types and classifications

B.1.1.1 Principle of operation

The regenerative turbine pump develops three to four times the pressure of a rotodynamic pump for the same impeller diameter and speed. Where the rotodynamic pump develops its pressure as a result of the difference of discharge and suction velocities, the impeller of the regenerative pump depends on the specific drag of the impeller and its peripheral velocity. The mechanism consists of an impeller with 20 to 40 machined buckets or teeth at its periphery. Usually, they are on both sides of the impeller. The impeller's teeth turn within a flow-through area, which has side channels. The teeth are staggered from one side of the impeller to the other to obtain a more uniform flow.

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Between the suction and discharge connections is a dam that blocks the channel. The impeller passes through the dam with close-fitting clearance on its sides and periphery. As the liquid enters the pumps, an orderly circulatory flow is imposed by the bucket of the impeller. This transmits momentum to the liquid as it travels around the periphery. The circulatory flow out of the impeller, into the side channels and back into the impeller, may occur four to five times during one revolution of the liquid around the periphery. When the liquid hits the dam, its inertia has built up a high level of energy that provides for a high-pressure coefficient and the resulting high pressures that are characteristic of regenerative turbine pumps.

The high specific drag of the impeller depends on the number of teeth (or buckets), the ratio of channel area to bucket area, the flow-through area's height to width ratio, and clearances of the impeller where it passes through the dam. The higher the specific drag, the higher the pressure coefficient and higher the efficiency.

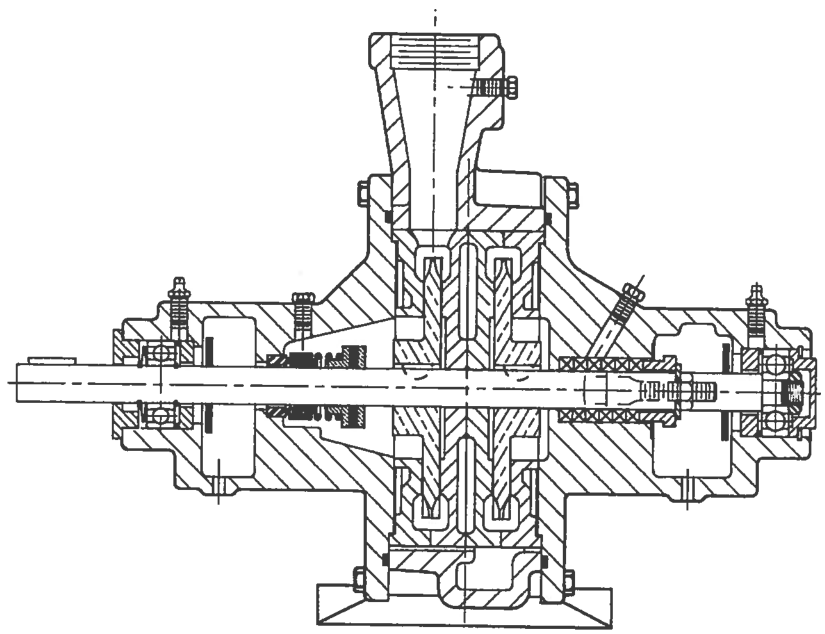

Figure B.2 — Regenerative turbine – impeller between bearings – two stage

B.1.1.2 Definitions and terminology (refer to Figure B.3 for terminology)

B.1.1.2.1 Bucket (impeller channels)

The bucket is where the liquid goes into the impeller for regeneration. The buckets are machined into the metal blank of the impeller. Usually an index gear machine is used for this operation. There can be 20 to 40 teeth per side of the blank. They are staggered in position from one side to the other to prevent pulsation and to give extra strength to the land between buckets.

B.1.1.2.2 Circulatory flow

This is flow of the liquid as it goes around the periphery of the casing-impeller. As the liquid goes into the side channel a circulatory flow is imposed by the bucket and transmits a momentum that throws the liquid back into the side channel at a greater pressure. The liquid then goes back into the bucket where the process is repeated. The pressure is increased each time it is thrown back into the side channel. This process may be repeated four or five times as the liquid goes around the periphery.

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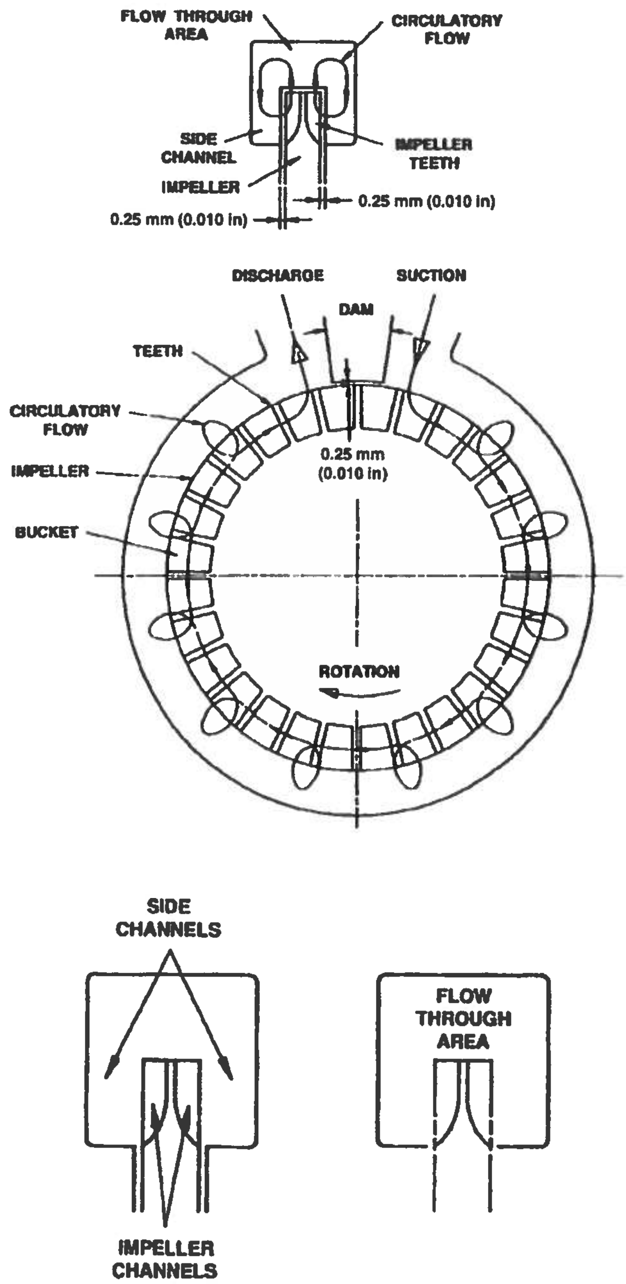

Figure B.3 — Regenerative turbine flow path

B.1.1.2.3 Dam

This is the block between the suction and discharge. When the liquid hits the dam its inertia builds up the pressure. There have to be close clearances between the sides and periphery of the impeller to the dam. If the clearances are not maintained, there will be an increase in slippage from discharge to suction resulting in a loss of pressure.

B.1.1.2.4 Discharge

To obtain the greatest effectiveness of the regenerative action around the periphery, the discharge and suction ports are adjacent to each other. The discharge and suction are usually the same size.

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B.1.1.2.5 Impeller

The impeller is a solid round disk that is completely machined on all surfaces. The buckets are machined in the periphery. The metal between the buckets is called the *tooth* or *web*. To prevent axial thrust from developing, most designs allow the impeller to float axially on the shaft. This also allows the close clearance to be maintained between the impeller and casing walls without mechanical restraint.

B.1.1.2.6 Side channels (flow-through area)

These are the passages of the casing that maintain the circulatory flow. The amount of flow of the pump depends on the size of the flow-through area of the side channel.

Some designs only have one-sided buckets and a channel. These types of design will develop axial thrust.

B.1.1.2.7 Side clearance

Clearance between the impeller and casing walls.

B.1.1.2.8 Suction

The suction port is adjacent to the discharge port and usually is the same size as the discharge.

B.1.1.2.9 Teeth

This is the metal web between the buckets. The web has to be thick enough to account for the fatigue loading on the tooth as it goes from full discharge pressure to suction pressure as it passes through the dam.

B.1.1.3 Application

These types of pumps, which are low flow; high head; one, two, or more stages, are used for small boiler feed pumps and desalination plants.

B.1.1.4 Design

B.1.1.4.1 Performance changes

B.1.1.4.2 Change in head at constant speed

Unlike a rotodynamic pump, the impeller diameter is not cut to change head. For a given pump, there is a given head. To change the head, a different casing-impeller combination is needed.

B.1.1.4.3 Change in flow at constant speed

Similar to rotodynamic pumps, i.e., to change the flow, change the impeller width and volute area. In this case, change the impeller width and channel area. This requires a new casing.

B.1.1.4.4 Change in speed

A change in speed will affect the rate of flow directly; head as the square and power as the cube – the same affinity rules as used on a rotodynamic pump.

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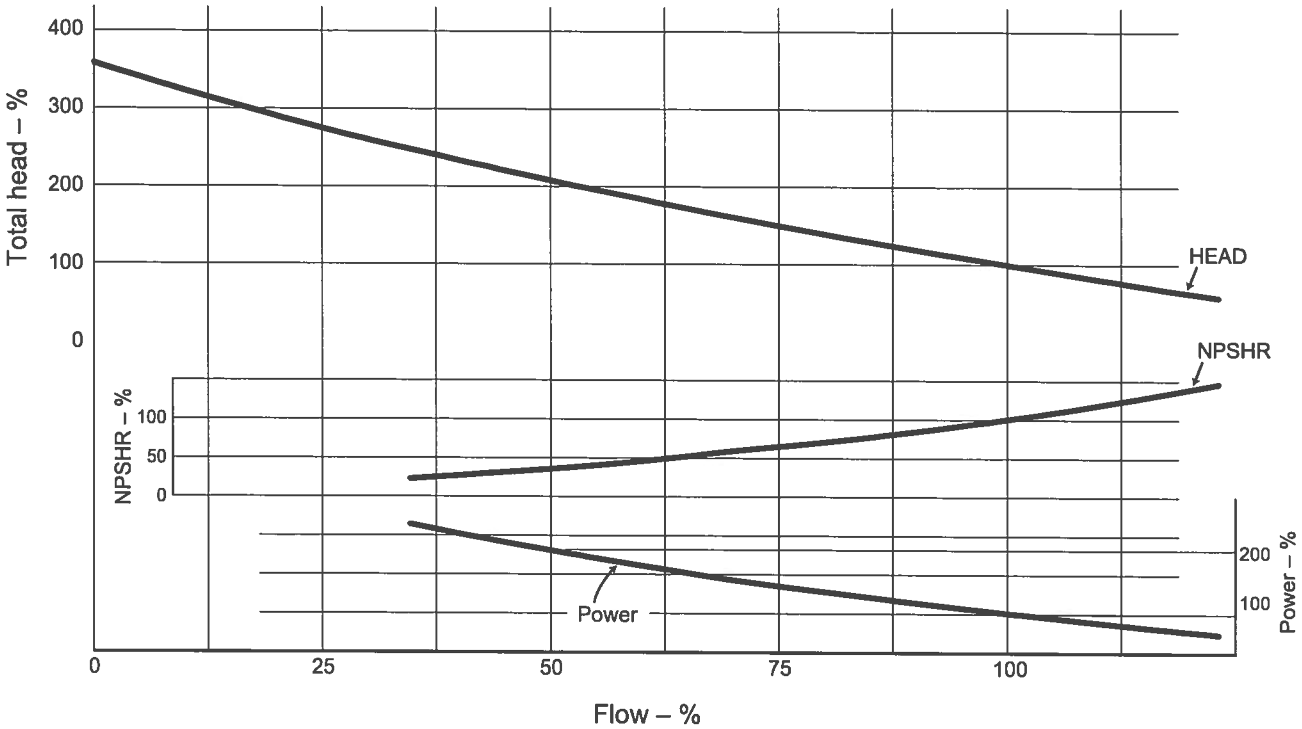

Figure B.4 — Regenerative turbine pump performance

B.1.1.4.5 Efficiency

For a specific speed n_s below 8 (N_s below 400), the regenerative turbine pumps are higher in efficiency than rotodynamic centrifugal pumps. This is because the fluid friction losses are not as great in a regenerative turbine as in a rotodynamic pump since the rotodynamic pump not only has disk losses, but also the internal passage losses of the vanes and volutes.

The maximum theoretical efficiency for a regenerative turbine is about 50%. Actual designs have been 47 to 48%. The regenerative turbine can be designed for n_s as low as 2 (N_s as low as 100) with efficiency of 35%; with an n_s of 4 (N_s of 200) with efficiency of 44%. If a centrifugal pump could be made with an n_s of 4 (N_s of 200), its efficiency would be around 30%.

B.1.1.4.6 Special characteristics

B.1.1.4.6.1 Radial thrust

A rotodynamic centrifugal pump casing is usually designed as a constant mean velocity volute. This means at best efficiency point (BEP), the velocities are the same around the periphery of the volute. The resultant radial thrust is almost zero. In a regenerative turbine the pressure increases by 25% for each quadrant. As the liquid reaches discharge, there is a strong resultant radial thrust. This occurs at BEP and gets stronger as the performance gets close to shutoff. This is why the shafts and bearings of regenerative turbines are heavier than those of rotodynamic pumps. With multistage pumps, the discharges are staggered so the radial thrusts of the impellers oppose each other.

B.1.1.4.6.2 Axial thrust

With the high pressures and close side clearances (0.25 mm [0.010 in] per side), it would be very difficult to maintain even pressures if the impellers were locked axially on the shaft. To distribute the pressure, buckets are put on

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both sides of the impeller. To compensate for residual uneven forces, the impeller is allowed to float axially on the shaft. This means that the bearing only has to absorb the radial thrust of the impeller(s).

B.1.1.5 Special maintenance

B.1.1.5.1 Clean liquids

Due to the close clearances of the dam and sidewalls, it is necessary to have clean liquid. The particle size should be no greater than 0.025 mm (0.001 in). Particles exceeding this parameter will result in reduced performance and the subsequent need to replace the close-fitting casing and impeller.

B.1.1.5.2 Strainer

A strainer with minimum of 20 mesh should to be installed in the suction line.

B.1.1.6 Summary

a) The pumps work on the principle of a specific high drag impeller imposing a circulatory flow within the channels of the casing and impeller. The momentum of the circulatory flow against the dam at the end of the channel creates high pressure.

b) These pumps are used for filtration, cooling, injection, lubrication, or smaller boiler feed.

c) For low flow, high head, they are more efficient than rotodynamic pumps. Efficiency range is 30 to 47%.

d) The impellers are not trimmed to change head or flow. Different sized casing–impellers combinations are required to obtain a change in performance.

e) The performance changes according to the affinity rules with change in speed.

f) The radial thrust is much higher than a rotodynamic pump. Opposing the outlets counteracts radial thrust on multistage pumps.

g) Axial thrust is eliminated by letting the impeller float axially on the shaft.

h) To prevent damage from particles, use a suction strainer.

i) Check seal chamber pressures for balanced seal requirements.

j) Check running materials for galling.

B.1.2 Pitot tube pumps

B.1.2.1 Nomenclature

B.1.2.1.1 Description of a Pitot tube pump

The Pitot pump is a variation of the rotodynamic pump design. It uses a closed casing that is attached to and rotates with the impeller (cover) while a stationary Pitot tube captures the discharge flow (Figure B.5). The inlet of the Pitot tube is positioned near the maximum inner diameter of the casing. The fluid enters the impeller (cover) along the axis of rotation and picks up momentum as it passes through the radial vanes of the impeller and into the casing. The liquid in the casing maintains an angular velocity slightly less than that of the casing. It then impacts the inlet orifice of the Pitot tube near the periphery of the rotating casing. The fluid is then discharged through the internal passageway of the Pitot tube and out of the pump. The total head developed by this type pump is the sum of both the static pressure head and the velocity head. This sum is equal to approximately 1.6 times the head

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produced by a conventional rotodynamic pump of the same impeller size and speed. In general, Pitot tube pumps are used for low flow and relatively high head (low *Ns*) applications.

In general, nomenclature of the Pitot tube pump is very similar to that of an overhung impeller, separately coupled, single-stage, frame-mounted rotodynamic pump. Those items that are unique to a Pitot tube pump are defined in Section B.1.2.1.2 below.

B.1.2.1.2 Definitions and terminology

(See Figures B.5 and B.6.)

B.1.2.1.2.1 Rotor assembly (1)

The rotor assembly consists of the rotor and the rotor cover, along with miscellaneous fasteners and O-rings. The rotor assembly contains the static head developed by the pump.

B.1.2.1.2.2 Rotor cover (2)

The rotor cover bolts to the rotor and acts as an impeller to conduct the liquid into the rotor assembly. Together with the rotor housing, the rotor cover forms the rotor assembly.

B.1.2.1.2.3 Rotor (3)

The rotor is connected directly to the driveshaft of the pump. Together with the rotor cover, the rotor forms the rotor assembly.

B.1.2.1.2.4 Bearing housing (4)

The bearing housing contains the bearings that support the rotating assembly.

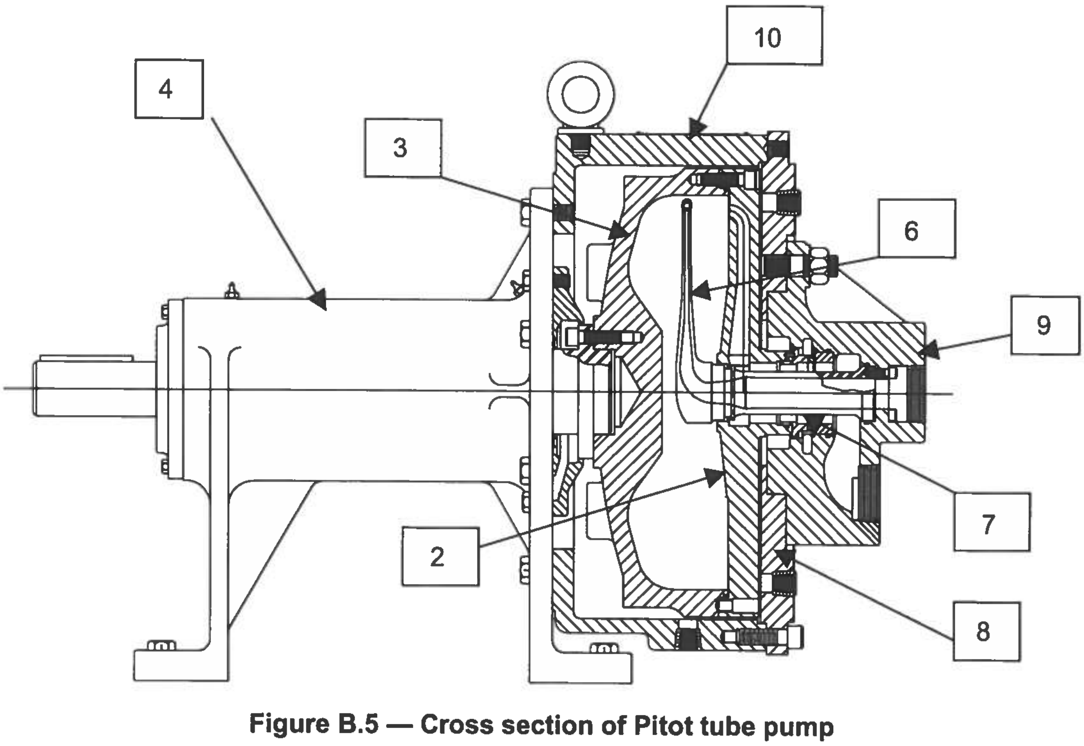

Figure B.5 — Cross section of Pitot tube pump

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B.1.2.1.2.5 Rotating assembly (5)

The rotating assembly consists of the rotor assembly, the shaft, bearings, and associated fasteners.

B.1.2.1.2.6 Pitot tube (6)

The Pitot tube extends from the manifold into the rotor assembly. The entrance to the Pitot tube is located near the periphery of the rotor assembly. In operation the Pitot tube converts the velocity head within the rotor assembly to pressure.

B.1.2.1.2.7 Mechanical seal (7)

The mechanical seal in a Pitot tube pump seals the suction liquid between the stationary manifold and the rotating assembly. In a Pitot tube pump, the mechanical seal operates in a suction (lower) pressure environment. The geometry of the Pitot tube pump is such that the use of a pack stuffing box is precluded.

B.1.2.1.2.8 End bell (8)

The end bell covers the suction side of the rotating assembly and supports the manifold.

B.1.2.1.2.9 Manifold (9)

The manifold consists of the suction and discharge nozzles of the pump, supports the Pitot tube, and contains the mechanical seal.

B.1.2.1.2.10 Rotor case (10)

The rotor case surrounds the rotor assembly and acts as a guard for the rotor assembly. It also supports the end bell and manifold.

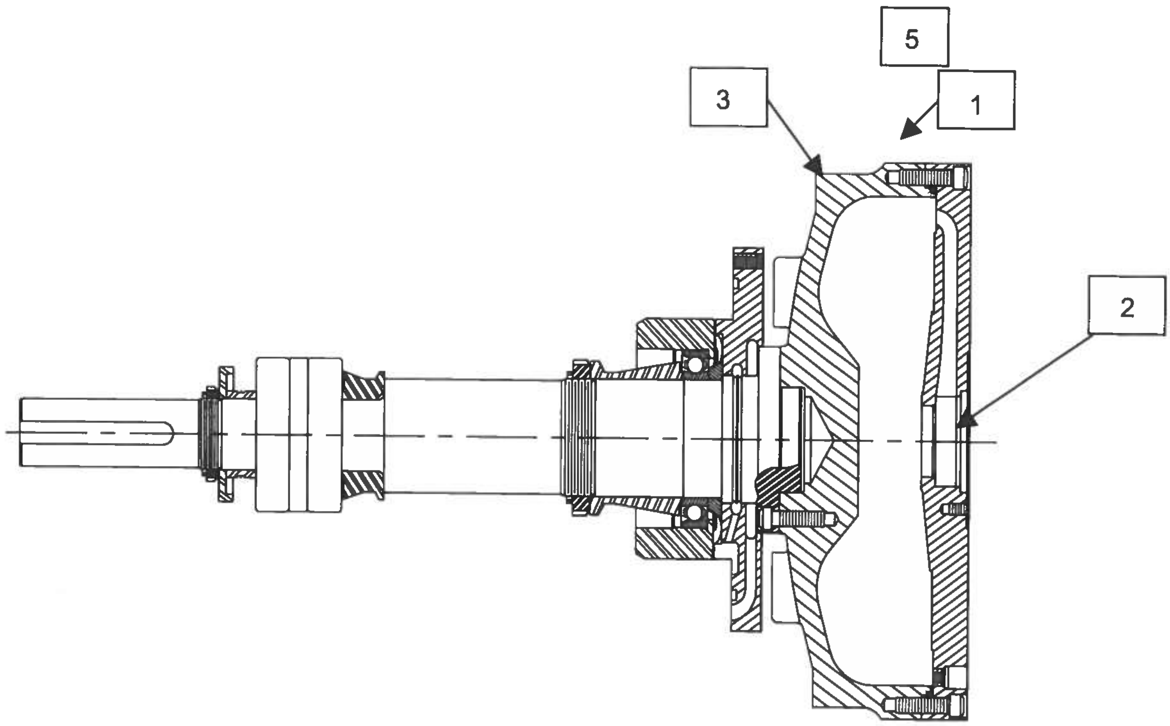

Figure B.6 — Rotating assembly

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B.1.2.2 Design and application

General design and application of the Pitot tube pump are very similar to that of an overhung impeller, single-stage, rotodynamic pump. Those items that are unique to a Pitot tube pump are discussed in this section.

B.1.2.2.1 Theory

B.1.2.2.1.1 Hydraulic theory

The Pitot pump develops its head in two mechanisms, by means of both a centrifugal (static) and a velocity head. As with a conventional rotodynamic pump, the combination of these two results in total head developed by this pump. However, the conversion of velocity to pressure head by the Pitot tube is much more efficient than in a conventional rotodynamic pump. A rotodynamic pump of similar specific speed may in practice produce an actual total head in the range of 70 to 80% of the theoretical head. The Pitot pump can produce an actual maximum head very close to the theoretical value.

There is a significant difference between a rotating casing Pitot pump and a conventional rotodynamic pump with regards to internal friction. A rotodynamic pump that has an impeller size in the same range as a Pitot pump rotor will be limited in its practical rotation speed due to the effects of disk friction. The amount of energy lost to overcome the disk friction between the rotating impeller shroud and the walls of the stationary casing grows at approximately the fifth power of the impeller diameter. This is why rotodynamic pumps designed for the same flow and head as Pitot pumps typically have very small impellers and operate at very high speeds.

In contrast, in the Pitot pump, the fluid is rotating at nearly the same speed as the rotating casing, thus the disk friction losses are low. The only hydraulic frictional effects that take place at the full fluid velocity are those between the fluid and the relatively small surface of the Pitot tube. It is the combination of a more effective conversion of velocity head into pressure and minimal friction loss that allows the Pitot pump to develop high heads at moderate speeds with good efficiencies.

B.1.2.2.2 Performance changes

There are two basic methods for changing the performance characteristics of the pump. Both the speed at which the pump operates and/or the size of the Pitot tube can be changed. This pump obeys the rotodynamic pump affinity rules with regard to the effects of speed change on head, flow, and efficiency. Changing the size of the Pitot tube opening results in a corresponding change in the head versus rate of flow performance of the pump.

B.1.2.2.3 Test standards

In general the Hydraulic Institute test standards for rotodynamic pumps (ANSI/HI 1.6 *Centrifugal Pump Tests*) will apply to Pitot tube pumps with the exception of hydrostatic testing of the rotor assembly and manifold.

B.1.2.2.3.1 Rotor assembly hydrostatic testing

Due to the nature of how the Pitot tube pump develops its pressure, hydrostatic testing of the rotor assembly will be different than that performed on a conventional rotodynamic pump. The rotor assembly in operation experiences an increasing pressure gradient that varies from suction pressure at the center to suction pressure plus 50% of the developed head at the outside edge. Full discharge pressure is not achieved until the fluid has passed into the Pitot tube and the velocity head is converted into pressure; the rotor never undergoes full discharge pressure. The structure of the rotor/cover assembly is not designed to handle a uniform load of 1.5 times the maximum discharge pressure.

Because of the above, the hydrostatic pressure testing of the rotor assembly is performed at a pressure of 1.5 times half the value of the maximum developed head plus 1.5 times the maximum suction pressure. (The rotor/cover assembly is supported in the center during this testing to avoid excess stress in the part, since the above testing places uniform pressure in the assembly that results in higher stresses than during operation.)

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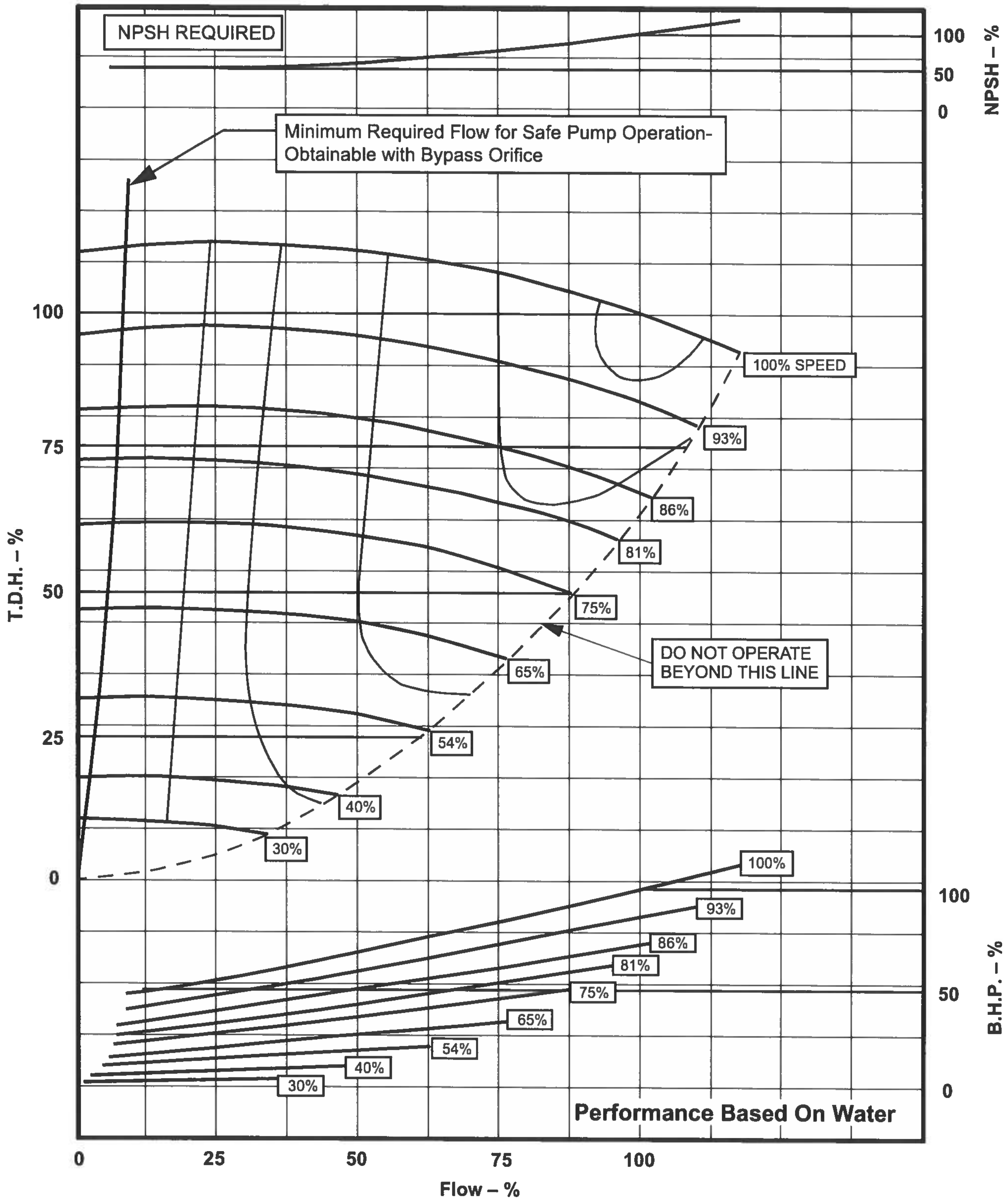

Figure B.7 — Pitot tube pump performance

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B.1.2.2.3.2 Manifold hydrostatic testing

In operation the outlet side of the manifold is subject to full discharge pressure and the inlet side is subject to suction pressure only. Standard procedure is to test the discharge side to 1.5 times maximum operating pressure and the suction side to the maximum rating of the suction nozzle. A fixture is required to separate the suction and discharge sections during testing of the discharge side.

B.1.2.2.4 Special characteristics

Because of its unique geometry, consideration must be given to several performance characteristics when applying the Pitot tube pump.

B.1.2.2.4.1 High vapor pressure fluid applications

The geometry of this type of pump requires the liquid to pass through the inside diameter of the mechanical seal and along the Pitot tube extension before it gets to the eye of the impeller. This allows the liquid to absorb heat prior to entering the impeller. High vapor pressure fluids, such as light hydrocarbons, may experience significant increases in vapor pressure with small temperature changes, giving a corresponding decrease in NPSHA. This may lead to cavitation problems when the NPSHA is close to the NPSHR. This type of application needs to be carefully evaluated and certain remedial measures, such as special seal design or higher minimum flows, may be required.

B.1.2.2.4.2 Solids handling

A Pitot tube pump is not designed to handle solids, because solids will cause rapid wear and balance problems, resulting in a short life.

Pitot tube pumps operate at very high fluid velocities (in some cases in excess of 100 m/s [300 ft/s]), so wear can be a serious problem if any abrasives are present. Abrasion will quickly wear out the Pitot tube. The rotating casing acts as a centrifuge and heavy particles will accumulate at the periphery of the rotating casing and remain in place. If the particles are abrasive, then the top of the Pitot tube may be worn off by abrasive action due to the very high relative velocity that exists between the periphery of the rotating casing and the Pitot tube. Also, the solids will settle to one side when the pump is stopped and there will be a very high unbalance at startup, which will cause high levels of vibration. If the solids do not redistribute themselves, this can lead to rapid bearing and seal failure.

Pitot tube pumps are often successfully applied in solids applications provided that the solids are not abrasive, are of a specific gravity relatively close to the liquid, and do not solidify or adhere to the walls of the rotor. In cases where low amounts of objectionable solids may be present, the use of a strainer in the suction line may allow application of the Pitot tube pump. Pitot tube pumps have been successfully applied in some moderately abrasive services through the use of abrasion-resistant coatings on the internal parts.

B.1.2.2.4.3 Pump starting

The rotating inertia of a Pitot tube pump is very high in relation to the motor size as compared to a conventional rotodynamic pump. Also they are often driven through a speed increaser that adds additional inertia to the load of the driver. Low-horsepower applications need to be evaluated for acceptable starting times. A motor larger than that required for operation may be required for starting. Usually a slow-trip overload relay is needed to prevent tripping during startup. Because minimizing the current draw by the motor on startup is of paramount importance, the pump is normally started against a closed discharge valve

B.1.2.2.4.4 Configuration

Pipe connections, seal maintenance, and shaft removal all take place at the front of the pump. Sufficient space must be available at the front of the pump for all of these activities.

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B.1.2.2.4.5 Mechanical seal

Both the incoming and the outgoing fluid must pass through the inside diameter of the mechanical seal. Thus for a given capacity this type of pump uses a larger mechanical seal than a conventional rotodynamic pump. In addition, the mechanical seal is designed to handle only the suction pressure of the pump. To protect the relatively large low-pressure seal typically used in this type of pump, a pressure-relief valve should be installed in the inlet piping between the pump and the intake shut-off valve. This will prevent damage to the mechanical seal (and the rotor) in case the pump is connected to a high-pressure manifold when shut down and the isolation valve is leaking slightly.

B.1.2.2.4.6 Flow considerations

A Pitot pump can be safely operated at any point along its head/capacity curve, from full flow to minimum bypass. The pump can be operated at minimum flow indefinitely, with no wear or damage to the pump. Only a small amount of bypass flow is required by this pump to keep the seal faces cool and to keep the fluid in the pump from overheating and vaporizing. The amount needed to do this will vary from 10 to 15% of the BEP flow rate for the pump. Because the Pitot pump is often operated in variable flow systems, correct design of the bypass piping is imperative.

B.1.2.2.5 Special maintenance requirements

In general, maintenance is not significantly different than for a conventional rotodynamic pump, except for the following two points.

B.1.2.2.5.1 Rotor and rotor cover

Material removal due to erosion, corrosion, mechanical damage, or cavitation may affect the balance of the rotating assembly and cause vibration problems. The rotor and rotor cover should be regularly inspected for signs of the above damage. If damage is evident, then the cause should be identified and eliminated.

B.1.2.2.5.2 Pitot tube

The condition of the Pitot tube will affect the successful operation of the pump. For optimum operation, the Pitot tube must be free of blockage and the inlet must be smooth and free of damage due to erosion, corrosion, or cavitation. Regular inspection of the Pitot tube is recommended.

B.1.3 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. 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 utilizes 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.

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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.1.3.1 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. 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 on 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 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 vary 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.1.3.1.1 Definitions and terminology

B.1.3.1.1.1 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.1.3.1.1.2 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.1.3.1.1.3 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.1.3.1.1.4 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).

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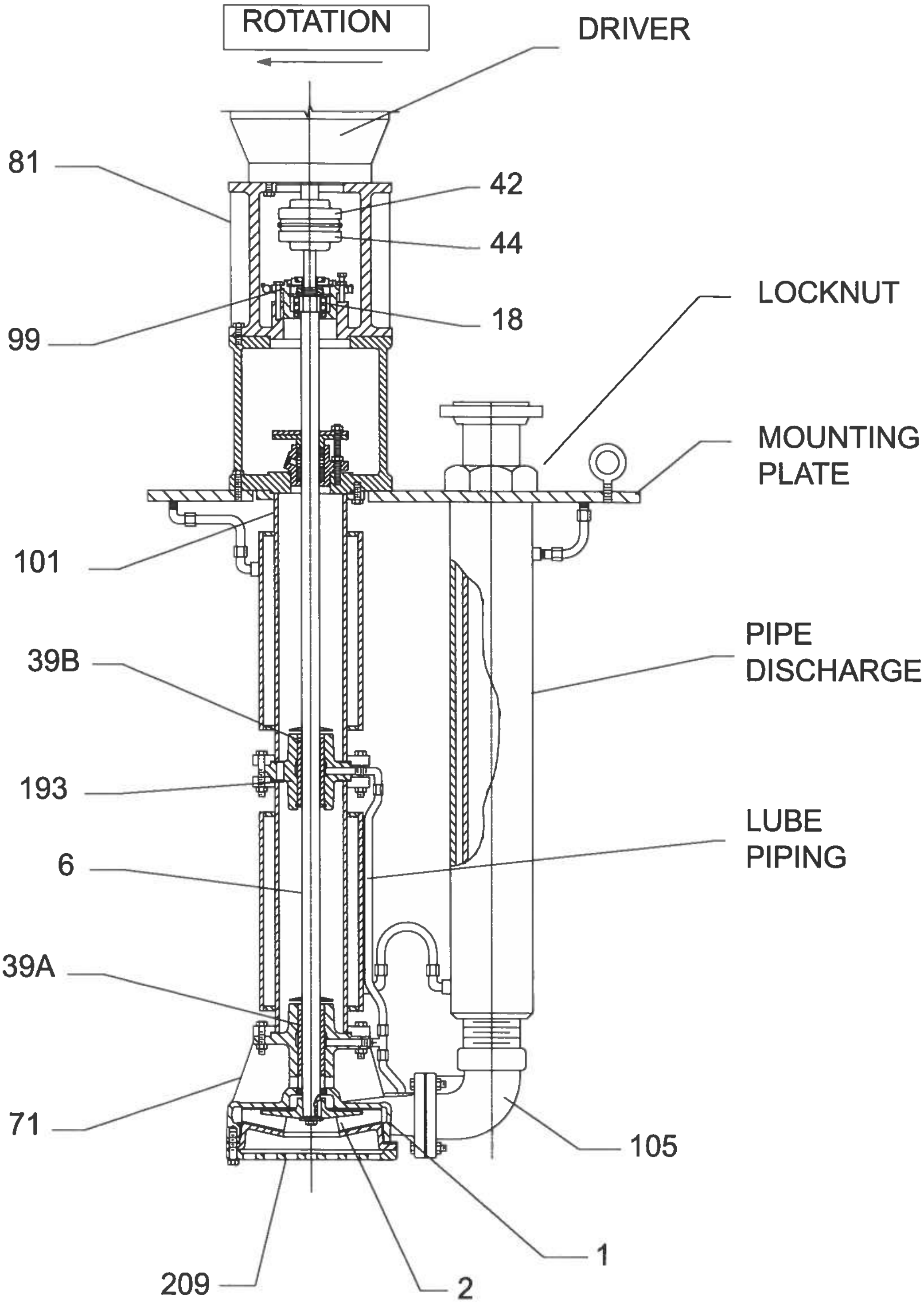

Figure B.8 — Type VS4 line-shaft design sump pump

B.1.3.1.1.5 Bearing, retainer (193)

The retainer holds the intermediate bearings. They are drilled and tapped 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.1.3.1.1.6 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.

B.1.3.1.1.7 Casing (1)

Single-stage overhung water pump casings or ANSI B73.1 casings.

B.1.3.1.1.8 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.1.3.1.1.9 Coupling (42 and 44)

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

B.1.3.1.1.10 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.1.3.1.1.11 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.1.3.1.1.12 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.1.3.1.1.13 Lube piping

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

B.1.3.1.1.14 Mounting plate

This is the plate that the motor support head, lubrication manifold, and discharge pipe are mounted or affixed to. 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.

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B.1.3.1.1.15 Shaft (6)

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

B.1.3.1.1.16 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.1.3.1.1.17 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 shaft, coupling, and driver shaft. The length of the support head is high enough to accommodate maintenance of the bearing housing and coupling.

B.1.3.1.1.18 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.1.3.1.2 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.1.3.1.3 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.1.3.1.3.1 Static head

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

B.1.3.1.3.2 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.

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B.1.3.1.3.3 Minimum submergence

The submergence to prevent vortex and vibration. Measured from the bottom of the strainer to liquid level.

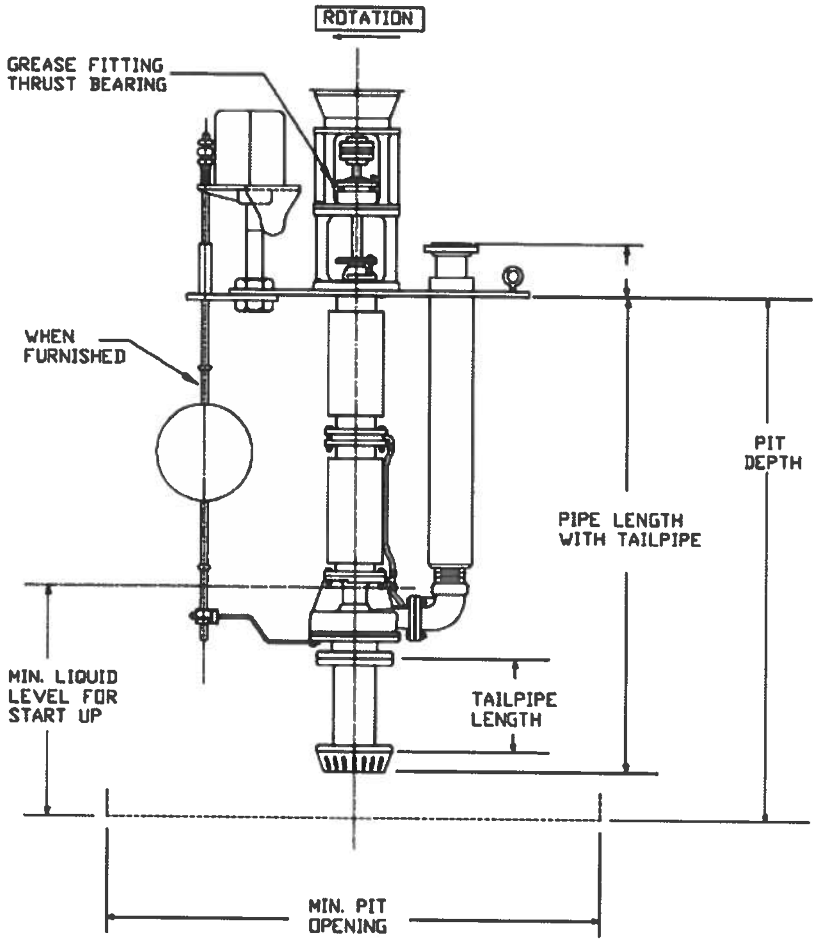

Figure B.9 — Tailpipe and float control

B.1.3.1.3.4 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.

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

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.

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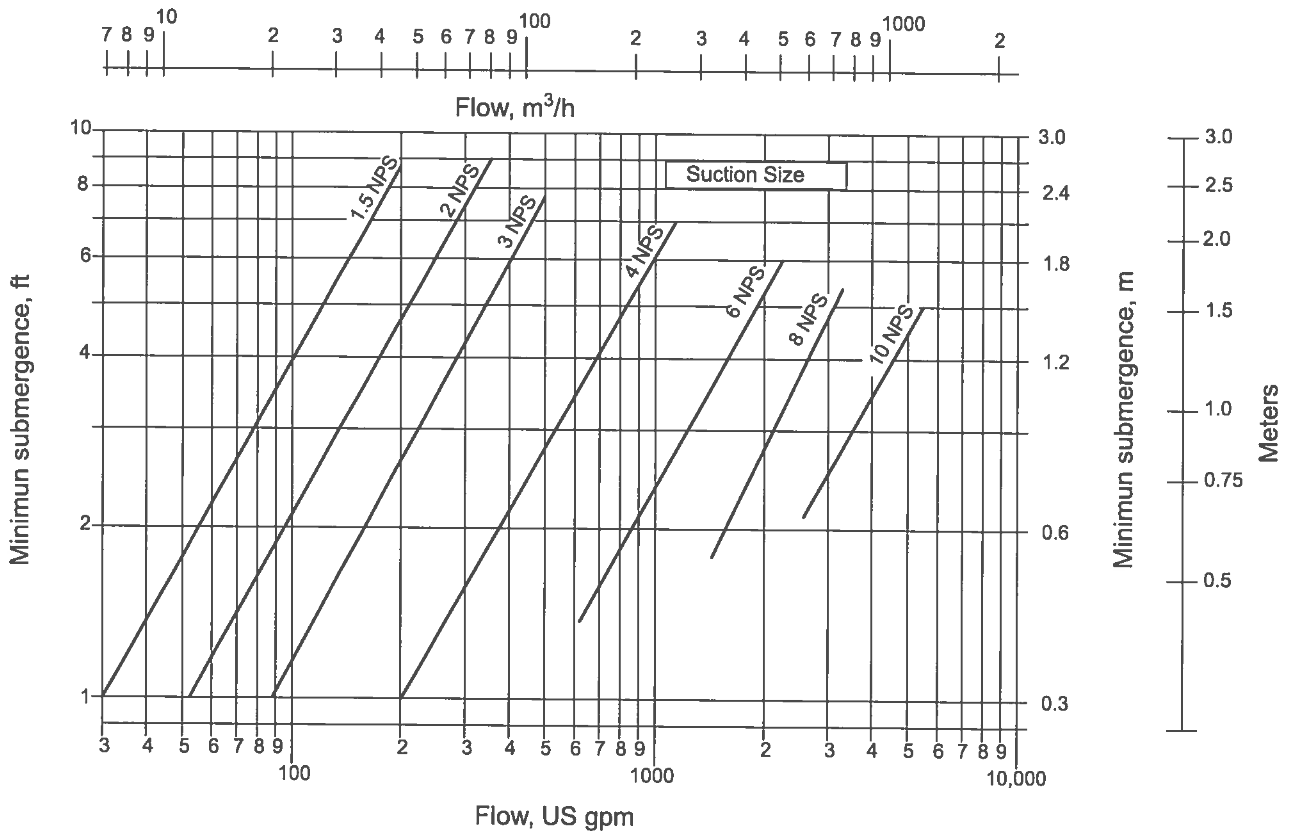

NOTE: Absolute minimum submergence for the above suction sizes at the lower flow rates are:

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.10 — Rate of flow versus minimum submergence

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B.1.3.1.4 Special characteristics

B.1.3.1.4.1 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, austenitic stainless steel
- Mounting plate: 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 μm). The following is a list of bearing materials and applications:
  - Carbon: Maximum 175 °C (350 °F), acids, general chemical, hydrocarbons
  - 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.1.3.1.4.2 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.1.3.1.4.3 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 column and discharge pipe, column and shaft.

B.1.3.1.4.4 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.

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

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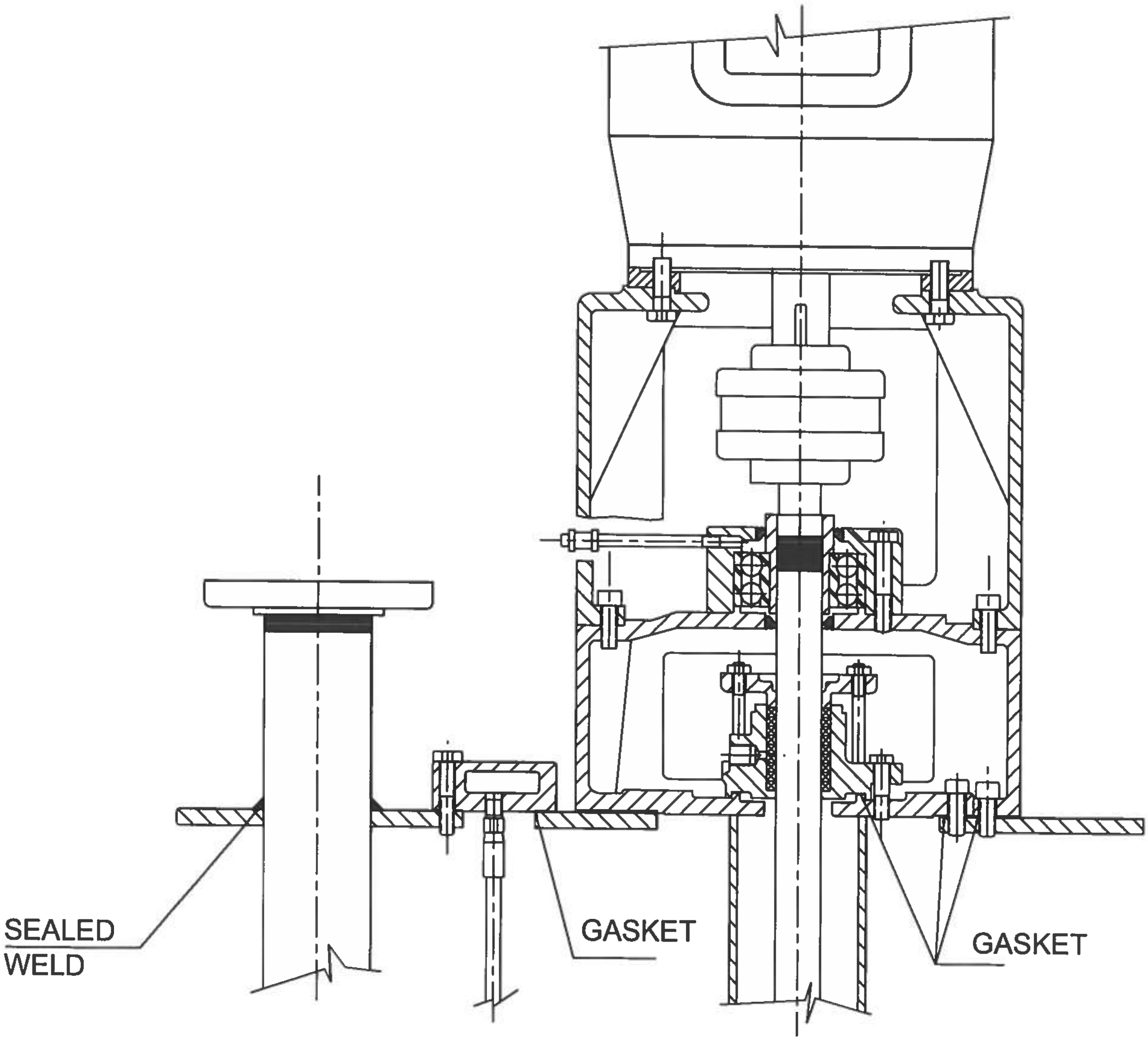

Figure B.11 — Schematic showing vaporproof/pressurized design

B.1.3.1.4.5 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.1.3.1.4.6 Alarms and controls

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

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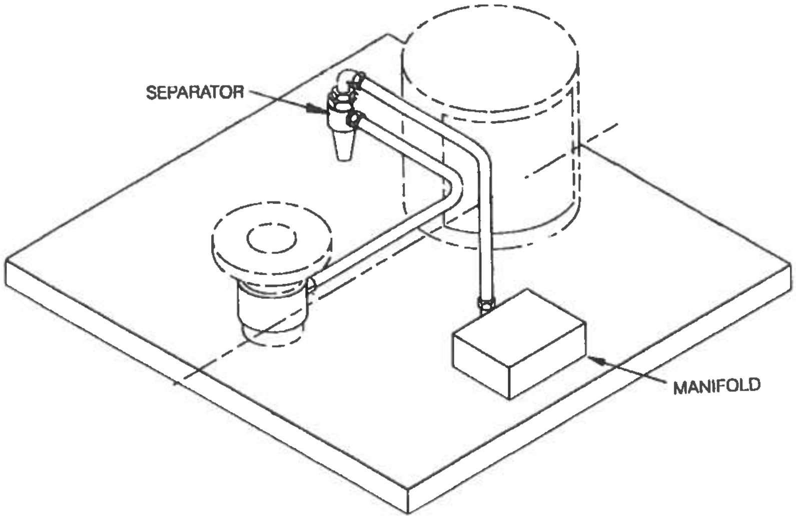

Figure B.12 — Alarms and controls

B.1.3.1.4.7 Nozzle loads

For pump types 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.

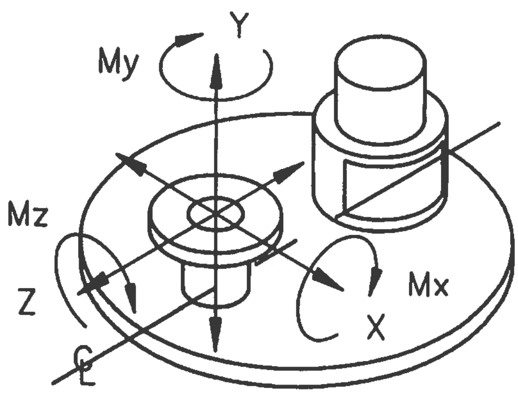

Figure B.13 — Applied forces and moments

B.1.3.2 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 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 overhang 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).

The reasons to use a cantilever include the following:

- Eliminate packing or mechanical seal
- Requires no submerged bearings

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- 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 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.1.3.2.1 Definitions and terminology

B.1.3.2.1.1 Bearing, outboard (antifriction) (18)

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

B.1.3.2.1.2 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.1.3.2.1.3 Casing (1)

Single-stage overhung water pump casings or ASME B73.1 casings.

B.1.3.2.1.4 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.1.3.2.1.5 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.

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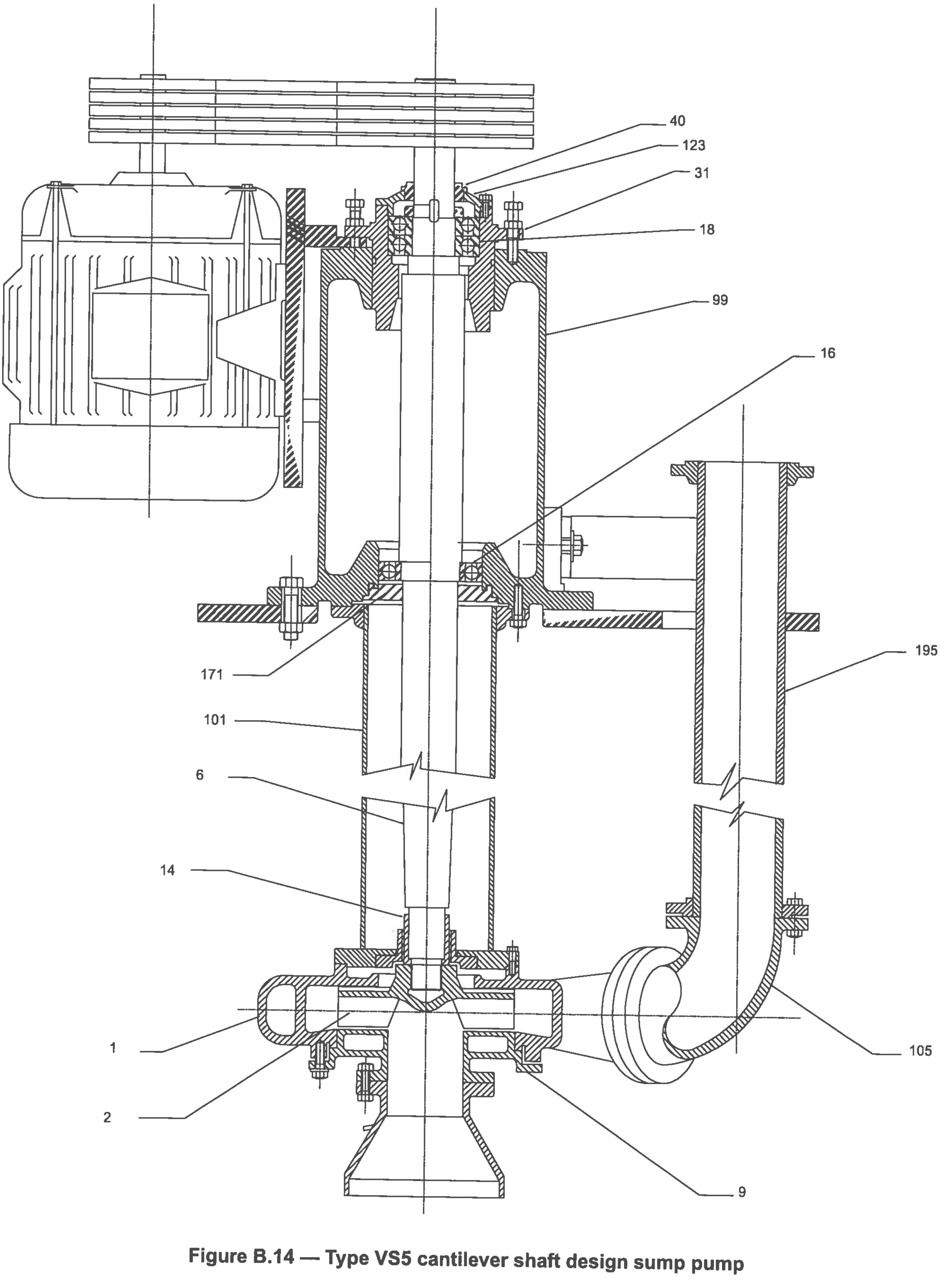

Figure B.14 — Type VS5 cantilever shaft design sump pump

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B.1.3.2.1.6 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.1.3.2.1.7 Impeller (2)

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

B.1.3.2.1.8 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 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.1.3.2.1.9 Shaft (6)

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

B.1.3.2.1.10 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.1.3.2.1.11 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 plate. 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.1.3.2.2 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.1.3.2.3 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.1.3.2.3.1 Static head

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

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B.1.3.2.3.2 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.1.3.2.3.3 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.

B.1.3.2.3.4 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.1.3.2.4 Special characteristics

B.1.3.2.4.1 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.1.3.2.4.2 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 of the sump, the vaporizing of the sump liquid 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.1.3.2.4.3 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.

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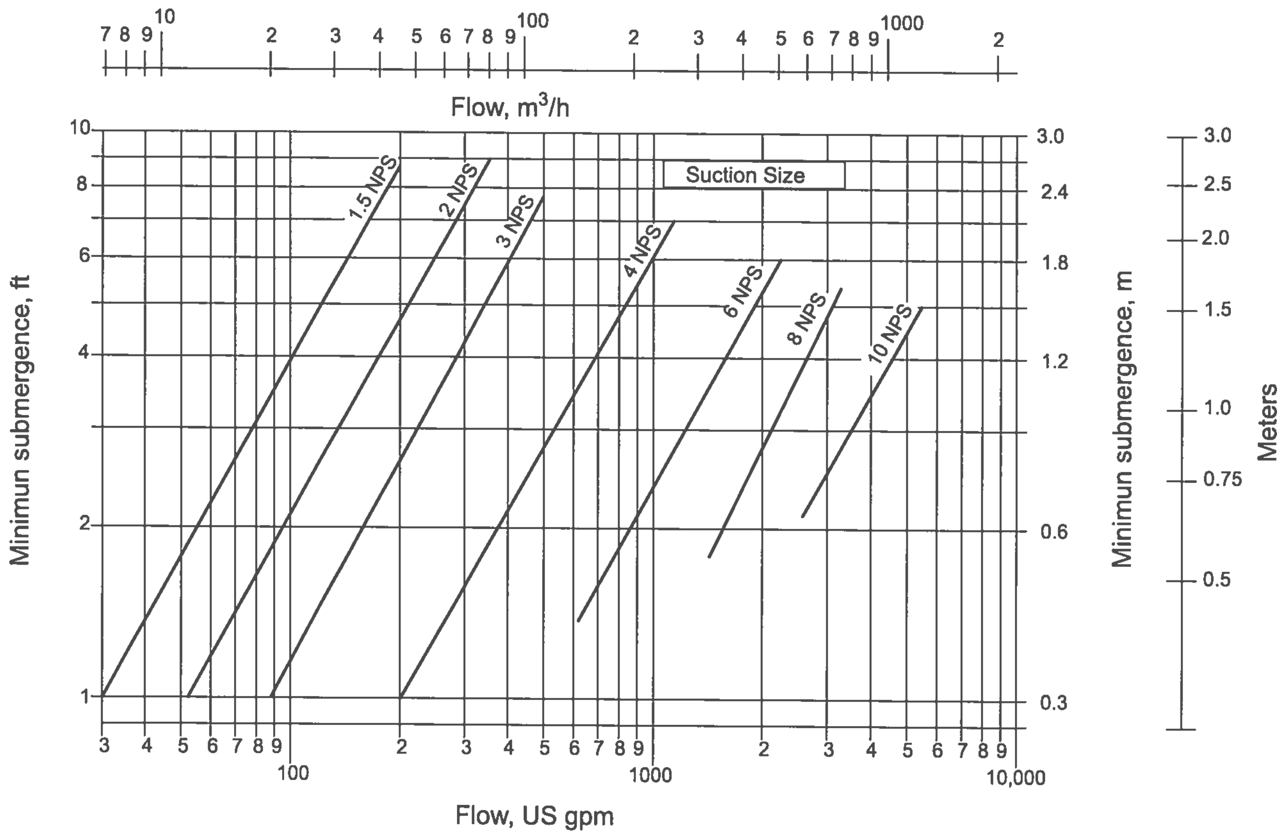

NOTE: Absolute minimum submergence for the above suction sizes at the lower flow rates are:

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.15 — Rate of flow versus minimum submergence

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B.1.3.2.4.4 Alarms and controls

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

B.1.3.2.4.5 Nozzle loads

For pump types 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.

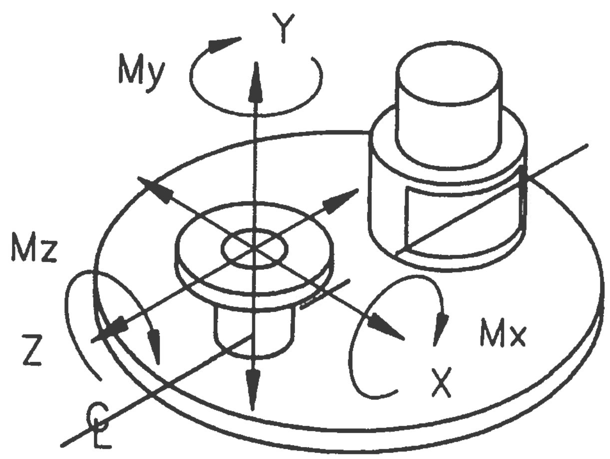

Figure B.16 — Applied forces and moments

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Appendix C

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.

C.1 Introduction to Appendix C

Every 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 transform 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.

Prime movers used for the purpose of driving rotodynamic pumps include the following types:

Electric motors:	Induction, synchronous: AC, DC, canned motor, submersible motor
Engines:	Diesel oil, gasoline, natural gas, crude oil
Turbines:	Gas, steam, hydraulic

Speed changes are accommodated by the use of additional devices:

Speed increasing:	Gears, fluid coupling, frequency converters
Speed reducing:	Gears, fluid coupling, frequency converters

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 rotodynamic pump turns out to be either electric motor or diesel engine. However, both methods may have limitations that dictate the need for other solutions.

C.1.1 Electric motors

Electric motor drivers

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

An electric motor is typically rated in horsepower at a given speed. For sizing, electric motors are specified by rated power and service factor. The 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.

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Table C.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 or speeds.	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
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 rotodynamic 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

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The motor must also be sized to develop sufficient torque to accelerate the pump to full speed. For smaller pumps with more than 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).

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.

C.1.1.1 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).

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.

C.1.1.1.1 Alternating-current (AC) single-phase motors

Split phase: A split-phase motor is a single-phase induction motor equipped with a main winding and an auxiliary starting winding. This type of motor has a switch that deactivates the starting winding as the motor comes up to speed. Split-phase motors are used in spa, jetted tub, and aboveground pool pump applications. The motors are usually rated from 100 W (1/6 hp) through 1.1 kW (1.5 hp).

Capacitor-start: A capacitor-start motor is a single-phase induction motor equipped with a main winding and an auxiliary starting winding with a series capacitor. This type of motor has a switch that deactivates the starting winding and capacitor as the motor comes up to speed. Capacitor-start motors are the most common type of single-phase motors found on in-ground pool, irrigation, and dewatering pump applications. They are usually rated from 100 W (1/6 hp) to 5.5 kW (7.5 hp).

Permanent-split capacitor: A permanent-split capacitor motor is a single-phase induction motor equipped with a main and an auxiliary starting winding with a series capacitor. The motor does not have a switch and both the main and starting windings are always energized. They are found on irrigation and dewatering pump applications. The motors are typically rated from 500 W (1/2 hp) through 11 kW (15 hp).

Capacitor-start, capacitor-run: A capacitor-start, capacitor-run motor is a single-phase induction motor equipped with a main and an auxiliary winding with a series run capacitor. Both windings and the run capacitor are permanently energized. In addition, this motor has a second capacitor called a *start capacitor* that is deactivated by a switch as the motor comes up to speed. This type of motor is also called a *two-value capacitor motor.* They are found on irrigation and dewatering pump applications. The motors are usually rated between 1.1 kW (1.5 hp) and 11 kW (15 hp).

Other types of single-phase motors are in use throughout the commercial and industrial world. Some of these are listed below for reference, but they are not usually found in industrial applications. Refer to NEMA for additional information on these and other types of motors.

Shaded pole: A shaded pole induction motor uses a short-circuited auxiliary winding for starting. These motors are used on small fans in fractional or subfractional horsepower ratings.

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C.1.1.1.2 AC polyphase motors

Types of polyphase motors:

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

Wound rotor induction: A wound rotor induction motor has an AC primary winding 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. 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.

C.1.1.1.3 Direct-current (DC) motors

DC motors have a field circuit energized by either magnets or direct current flowing through a coil and an armature circuit energized by a source of DC power. The speed of a DC motor is varied by changing the DC voltage and current applied to the armature. There are three basic types of DC motors with energized fields:

- A shunt wound motor has its field in parallel with its armature. The armature receives its power from either the same source as the field or from a separate source.
- A series wound motor has its field in series with its armature.
- A compound wound motor has two separate field windings. The main field winding is usually connected in parallel with the armature and the second field winding is connected in series.

In a permanent magnet DC motor, field excitation is provided by permanent magnets.

C.1.1.2 Electric motor construction

C.1.1.2.1 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.

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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 C.2 provides a brief description of several of the most common enclosure types.

Table C.2 — Common electric motor enclosure types

	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 Nonvent	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 externally to all enclosed parts.
	Totally Enclosed 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.

Source: NEMA MG 1-1998

C.1.1.2.2 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 represents the enclosure's level of protection against incidental contact with internal components. The second defines the amount of water ingress that the enclosure must protect against. Tables C.3 and C.4 define the IP designation system. For instance, 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 C.3 — 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 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 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 C.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

C.1.1.2.3 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 shutdown of a motor if the temperature exceeds a predetermined maximum.

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.

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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 since 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 medium 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.

C.1.1.2.4 Motor bearings

Electric motors are constructed using either antifriction or sleeve bearings, with the former being used in the vast majority of motors from fractional to 375 kW (500 hp).

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 hydrodynamic plain bearings. However, there are some disadvantages to the use of antifriction bearings. The limiting speed and geometry can affect motor design. Replacement requires the complete disassembly of the motor enclosure. The most obvious disadvantage is that every antifriction bearing, no matter how well maintained, has finite service life. The load zone of an antifriction bearing is subjected to cyclical loading. When the rolling elements pass over a given area on the outer race, that area is loaded and unloaded every time another ball passes over it. This eventually fatigues the metal in that area and causes the bearing to fail.

Sleeve bearings are often used in higher horsepower, more specialized motors.

C.1.1.2.5 Mounting methods and orientation

Horizontal mounting

Foot-mounted motors can be mounted horizontally using the integral feet attached to the enclosure to support the unit. When mounting a motor in this fashion, it is important that the base or foundation be designed to adequately support the unit as well as limit excessive vibration. The natural frequencies of the motor on its permanent foundation should be at least 20% removed from both its rotating frequency and twice its rotating frequency. Foundations should be flat and motor feet should be shimmed to ensure good contact between the motor and base.

Another design available for mounting a motor horizontally is a C-face or C-flange. This design is used on close-coupled machinery where the mounting holes in the flange are threaded to receive bolts. The motor is then connected directly to the driven equipment by passing bolts through the driven equipment flange and threading them into the motor flange. Some C-flanged motors also have mounting feet to provide mounting flexibility.

Vertical mounting

Motors can be mounted vertically in one of two ways. The motor feet can be used to mount a unit to a vertical base or wall. Special flanges such as a C-flange or P-base can also be used to connect the motor directly to the driven equipment and position the motor in a vertical position.

Motors operating in a vertical position face a variety of unique design and operating issues. Refer to ANSI/HI 2.3 *Rotodynamic (Vertical) Pumps for Design and Application* for expanded details on drivers for vertical pumps.

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C.1.1.3 Performance characteristics

C.1.1.3.1 Relationship between 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.

C.1.1.3.2 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.

Torque can be calculated by transposing the formula for power.

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

τ = torque, in N·m
P = power, in kW
n = speed, in rpm

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

τ = torque, in lbf-ft
P = horsepower, in hp
n = speed, in rpm

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

Starting torque is also referred to as *locked rotor* 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 B motor develops approximately 150% of its full-load torque.

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% of full-load torque. Breakdown torque is the maximum torque a motor can produce. If the motor were overloaded beyond its torque capability, it would stall or abruptly slow down at this point.

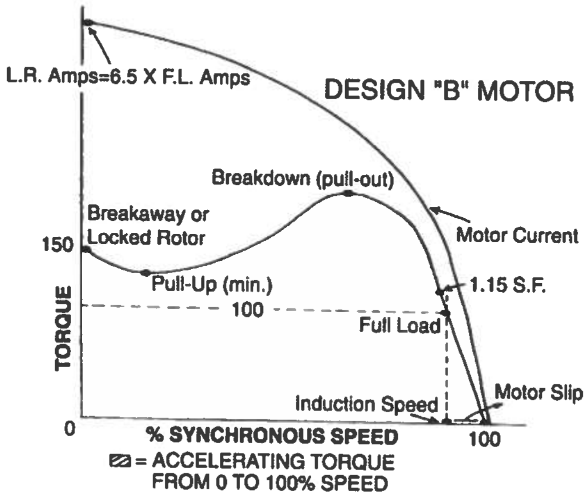

Figure C.1 — Torque–speed curves for NEMA Design AC motors

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. Full-load torque is often determined by acceptable temperature rise.

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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 C.2 shows the performance curve for each type. Note that this figure has torque on the vertical axis and speed on the horizontal.

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 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.

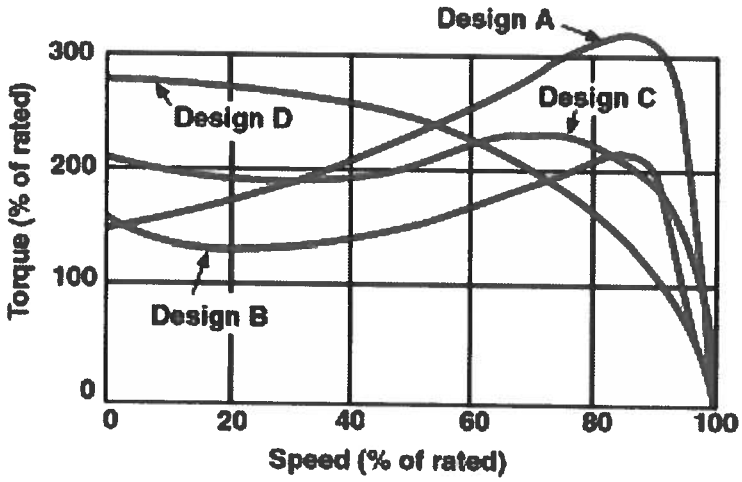

Figure C.2 — Design A, B, C, and D for AC motors

NEMA design C

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

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.

C.1.1.3.3 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 a 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.

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C.1.1.3.4 Power factor

Power factor is the ratio of actual power used by the motor, expressed in watts or kilowatts, to the power that is apparently being drawn from the electrical supply system, expressed in volt-amperes or kilovoltamperes. (Power Factor [PF] = watts / VA or kilowatts/kVA.)

Since the current drawn by an AC induction motor is out of phase with the supply voltage, this electrical term is used to rate the degree of synchronization of power supply current with the power supply voltage (where the power supply voltage is the voltage applied to the motor from the electrical supply system).

Discussion

Power factor is an electrical term that applies to the supply and delivery of electric power. It is not easy to explain, but it can be important to understand as it directly relates to the cost of electricity used to power electric motors. Power factor can essentially describe the degree of synchronization between the current (amperes) supplied to a load versus the applied voltage of AC power systems. If the synchronization ratio is 1.0, it is defined as *unity power factor*. If the ratio is less than 1.0, it is expressed in decimal form similar to efficiency, e.g., 0.90. A lagging power factor means that the power supply must supply more current than the load requires to deliver the power being demanded. There can be added costs to the user for a large electrical power load to have a lagging power factor, such as in a large factory, refinery, or pumping station.

To understand power factor, it is necessary to understand that power has two components: working power and apparent power. Working power is the result of the applied current and voltage consumed as power by a load, such as a motor, light bulb, heaters, etc. Working power is calculated by multiplying volts by amps and is expressed in watts. One horsepower equals 746 watts, or 0.746 kW.

Apparent power is defined as kVA (kilovolt-amperes) and it is the total of working power and the power component called *reactive power*. Reactive power does not perform useful work but it is needed to sustain the electromagnetic field associated with inductive loads, such as electric motors. It is expressed by the term *kVAR*. It occurs because an electromagnetic device such as a motor or transformer tends to resist the flow of current, which results in current "lagging" the voltage waveform. To make up the total sum of current required for apparent power, magnetizing current must be supplied in addition to working current.

Motors, or factories with many motors, that have lagging power factor causes the user additional fees for the utility to supply this extra current, which is not directly measured by the power meter. This additional measurement is power factor and is defined as kW/kVAR. These values typically range from 0.80 to 0.98. However, in a lightly loaded or cyclically applied motor, these values can be much smaller, which makes avoiding these situations appropriate to reduce power costs.

Higher power factor equipment will reduce power costs. Motors can have their power factor increased by the addition of power factor correction capacitors (PFCCs), where appropriate. Induction type motors normally have lagging power factor values, but synchronous motors usually have unity or leading power factors. They can also be used instead of induction motors to reduce power costs, but they typically are only considered cost-effective when the ratio of motor power (hp) to rpm is 1.0 or less. Low-speed motor applications (514 rpm or less), may frequently benefit from the use of a synchronous motor.

In summary, it is important that the meaning of power factor and its effect on the electrical supply system is understood. Power factor is a criterion that a user may evaluate when choosing a motor 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) A customer may request assistance in selecting equipment to correct a low power factor motor (PFCCs).

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

C.1.1.3.5 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.

NOTE: Typically a 1.15 service factor motor runs cooler at nameplate horsepower than a 1.0 service factor motor does at the same nameplate horsepower. A 1.15 service factor motor at service factor horsepower runs hotter than a 1.0 service factor motor at nameplate horsepower.

C.1.1.3.6 Altitude

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.

Altitude (meters)	Altitude (feet)	Derating Factor
> 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

C.1.1.3.7 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 (cycles per second) is common. The frequency of the AC current will affect the speed of a motor, i.e., synchronous speed = [120 × f (frequency)]/No. 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.

C.1.1.3.8 Starting

Motors shall start and accelerate to running speed a load with 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.

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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. (Refer to NEMA MG1-12.44.1.)

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. Since 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.

Number of starts

Induction motors with horsepower ratings and performance characteristics in accordance with NEMA 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 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.

C.1.1.4 Classified (or regulated) areas (hazardous atmospheres)

C.1.1.4.1 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 a location where a hazardous gas or vapor is present. Dust-ignition-proof defines enclosure requirements for motors used in locations where ignitable dust is present. Here are the definitions from the NEMA MG1:

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.

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C.1.1.4.2 National Electrical Code – Hazardous locations and materials: class, division, group

Articles 500 through 504 of the 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.

Class III – Locations that are hazardous because of the presence of easily ignitable fibers or flying, 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. In general, 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," i.e., 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.

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.

Explosion-proof motor requirements

- The enclosure must be able to contain an explosion – enclosure parts meet minimum tensile strength and thickness requirements. Fasteners must be sized to contain the explosion and meet 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

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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.

Dust-ignition-proof motor requirements

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

C.1.1.5 Variable-speed power sources for electric motors

Operating a pump at a speed that coincides with its most efficient point on its performance curve can provide optimum efficiency and/or process performance. To control the speed of a pump connected to an electric motor driver, the motor must be controlled differently than a fixed-speed motor. Instead of connecting the motor to a fixed voltage and/or frequency power source, devices that can vary voltage and/or frequency must be used to control these parameters and thus vary the speed of the motor. The subject of variable-speed pumps is presented in detail in the Europump/HI Guide *Variable Speed Pumping - A Guide to Successful Applications*. Refer to this document for extended review of the benefits and techniques used to vary the speed of pumps via electric motors.

A brief review is presented here for reference.

In the case of AC motors, the method of speed control depends on the type of AC motor. For the typical induction motor, the most common method is use of an adjustable or variable-frequency control. These controls are also called *drives*. This leads to the use of following terms in reference to these controls:

- ASC – Adjustable speed control
- AFD or VFD – Adjustable/variable-frequency drive

These devices control voltage and/or frequency of the power connected to the motor. The variation of these parameters will vary the speed of the motor according to the following principles.

The operating speed of an induction motor depends on the number of magnetic poles it is designed and built with. The following formula governs: $N = 120 \times (F/P)$, where N = speed in rpm, F = frequency, and P = number of poles. In the common world of fixed-frequency utility power, fixed-speed induction motors are restricted to the following options (rotational speeds [rpm] shown are at synchronous or unloaded condition):

- 2 pole = 3600 rpm at 60 Hz and 3000 rpm at 50 Hz

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- 4 pole = 1800 rpm at 60 Hz and 1500 rpm at 50 Hz
- 6 pole = 1200 rpm at 60 Hz and 1000 rpm at 50 Hz
- 8 pole = 900 rpm at 60 Hz and 750 rpm at 50 Hz; etc., for 10 poles and slower motors

To operate an AC motor at different speeds, the only practical solution is to vary the frequency of the applied power/voltage. A direct relationship exists so that a two-pole motor operating at 60 Hz will rotate at 3600 rpm, and the same motor will operate at 1800 rpm at 30 Hz; and a four-pole/1800-rpm motor will operate at 900 rpm at 30 Hz, etc. By utilizing a variable-frequency control/drive, the rpm of a motor can be varied throughout all possible frequencies available from the control, limited to the capability of the control and/or motor. The limitations are typically thermal limits of the drive and the motor, and/or rotational mechanical limits of the motor. The relative size of both the drive and the motor also has an impact on practical limits of speed variation.

Another important parameter to consider is that of the motor load during the range of speed variation. In the case of a typical rotodynamic pump, the load (torque requirement) of the pump varies by the square of the change in speed. This type of load profile can be plotted as a speed versus torque curve, commonly called a *variable torque load*. Therefore, as the amount of current draw from the motor varies in direct proportion to torque required by the load, a variable torque load (typical rotodynamic load) will have its peak current demand at full-load conditions. As the speed is reduced, current draw is also reduced and typical thermal limits of the drive and/or motor will not be challenged. Thus, the motor and drive of a given power (kW or hp) rating would not typically require extra capacity (oversizing) for rotodynamic pumps.

In the case of other types of loads, such as a positive displacement pump, load torque remains constant as speed is reduced, and a load profile known as a *constant torque load* applies. In a constant torque application, constant current is also required, which in turn requires the motor to deliver full-load thermal capability as speed is reduced. Thermal capability is often directly proportional to speed; therefore as speed is reduced, motor cooling capability may be reduced.

The thermal capability of the variable-speed drive will also need consideration in a constant torque application, as the cooling capacity and current density of the solid state devices in the drive may be subject to limits that could require an oversized control.

Other parameters that could affect motor sizing are related to the VFD technology. A VFD converts a fixed power source into a variable source by disassembling the power source and then reassembling it so that frequency can be varied. A side effect of this process is the introduction of undesirable electrical harmonics in the output waveform that negatively contribute to motor heating. In early-generation drive technologies this effect could be pronounced. In technology used since the late 1990s, the effect has been minimized. Nonetheless, a typical motor will operate at a higher temperature when powered by a VFD compared to "pure" power from a utility grid. Fortunately, in most low-voltage (≤ 480 V) induction motors through 185 kW (250 hp), there exists sufficient thermal capability to withstand extra heating resulting from a VFD waveform on a variable torque load. Motors of higher power ratings may need to be oversized and consultation with the motor manufacturer is advised.

A large volume of information is available from both motor and VFD manufacturers. An initial investigation into the capabilities and effects of and/or to an induction motor driver should be considered before assuming satisfactory performance when applying a VFD to a motor.

In the case of DC motors, speed control is usually managed by varying voltage applied to DC armature windings and/or DC motor field windings. This basic technology has been applied since before1950, and it is well known. Some type of voltage control is always needed for a DC motor, even for fixed-speed operation. Since the advent of AC induction motors, DC motor pump drivers are rarely found in industrial pumping applications due to the inherent simplicity, ruggedness, and reduced maintenance of AC induction motor design.

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Other types of motors can be controlled via variable frequency and/or variable voltage controls. Motor types such as wound rotor and synchronous AC, brushless DC, and others are still in use for various applications and can be operated for variable-speed processes as well as fixed-speed applications.

C.1.1.6 Reference industry standards organizations

NEMA - National Electrical Manufacturers Association

IEC - International Electrotechnical Commission

IEEE - Institute of Electrical and Electronics Engineers, Inc.

ANSI - American National Standards Institute

API - American Petroleum Institute

C.1.2 Engines

Introduction - general information: Uses, benefits, and limitations

Engine ratings are qualified by the severity of service, the extent of accessories, combustion air conditions, and performance variations due to manufacturing tolerances. Engine sizing depends on many factors, including brake horsepower required at various pump operating conditions, the duration of such operation, and ambient conditions (altitude and temperature).

Engines are typically used as drivers for rotodynamic pumps where there is an unacceptable risk of interrupted, unreliable, unavailable, or unsuitable electrical power source. They are preferred for these reasons on numerous common pump applications, including emergency and standby service, such as fire pump applications. Also they are used extensively in the many applications where portably powered systems are required, including pumps for marine applications or trailer-mounted pumps for dewatering or boiler tube descaling. As back-up for electric-motor-driven pumps, engine-driven pumps are used in the event of a power failure for industrial and water distribution applications.

Engines do not find nearly as many applications for driving pumps (as compared to electric motors) due to drawbacks such as

- Typically higher cost than a comparable electric motor.
- Typically higher installation costs. (In some situations, however, such as a fire pump application that requires a back-up power source and a transfer switch, the installation cost for an electric motor could go higher, making a diesel engine package more attractive.)
- Inefficiency: Engines have efficiencies of 30 – 40% compared to electric motors that can achieve 75 – 94% efficiency.
- Excessive noise – mechanical and raw exhaust.
- Excessive heat – radiated and raw exhaust.
- Environmental impact of emissions.

C.1.2.1 Engine types

Engines can be classified as spark ignition (gasoline or natural gas fueled engines) or compression ignition (diesel fueled [or equal] engines).

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For some applications, such as fire pumps, spark ignition engines are not allowed due to possible unreliability for emergency, standby service.

In following sections, diesel engines will be discussed in detail. Many of the items noted below for a diesel-fueled engine would apply to a gasoline-fueled engine as well.

C.1.2.2 Gasoline engines

Although gasoline engines may also be considered as drivers for rotodynamic pumps, their use is not as prevalent as diesel engine drives, which are covered in detail in the following paragraphs.

C.1.2.3 Diesel engines

Diesel engines find greater acceptance in pump applications than spark ignition engines due to the much lower volatility and flammability of diesel fuel and the ability to store diesel fuel more readily than gasoline.

Engines used to power pumps most often come from major engine manufacturers whose primary production is truck (on-road and off-road engines) or tractor (off-road – agricultural) engines. Such engines are produced in very high volumes. Since truck engines are not typically hard mounted (base mounted) or have mounting feet, engines that drive pumps have to be modified to suit most pump applications. Typically, the engine distributors perform these modifications.

C.1.2.4 Construction

C.1.2.4.1 Engine mounting

Not all engines perform well when they are hard mounted (base mounted). Significant vibration problems can be encountered. Engines equipped with balancers will produce less vibration than engines without balancers. Typically engines of four cylinders or less will often require vibration isolators to be placed between the bottom of the engine feet and the pump base engine mounts. Vibration isolators are typically made out of a specific durometer (hardness) and thickness rubber. The engine supplier will advise if vibration isolators are necessary. Typically six-cylinder and higher engines with either a rubber damper or dynamic (fluid-filled) damper will not require vibration isolators.

Note that when an engine must be mounted on rubber isolators, it will vibrate more and therefore a suitable coupling should be used to prevent this vibration from being transmitted to the pump. In most instances a soft rubber element coupling or a short-coupled driveshaft will suffice.

For hard mounted engines with no vibration isolation, most traditional pump couplings can be used, however, since engine vibrations are still typically higher than electric motors for the same horsepower and speed, the preferred coupling is a driveshaft.

As a rule of thumb, for an engine with a balancer, the typical maximum permissible vibration level is about twice that of an electric motor.

C.1.2.4.2 Types of drive connection

Engines typically come with standard SAE flywheel housings (per SAE J617) and flywheels (per SAE J620). Bolting a drive disk to the locator fit on the flywheel and then bolting a stub shaft, half coupling, or driveshaft to this drive disk is typically how the connection to the pump is made. An alternative to this is driving the pump shaft directly through either a splined or keyed connection with the drive disk. In this type connection, the pump and bearing housing are aligned and bolted directly to the engine.

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A power take-off device (PTO) can also be employed, which normally comes with a manual-disengaging clutch. A PTO is sized for the engine rated horsepower and is made to bolt up to the flywheel housing and also attach to the flywheel. The PTO normally has a keyed output shaft for mounting a typical pump coupling or driveshaft.

A centrifugal clutch is used on engine applications where the coupling "drive" member is mounted onto the engine and the coupling "driven" member is mounted onto the pump shaft. A centrifugal clutch coupling is used when another driver drives the pump (via a double extended shaft on the pump) and therefore the pump needs to rotate while the engine remains stationary.

C.1.2.4.3 Engine cooling

In many pump applications, the fluid being pumped can cool the engine. Such an engine is called a *heat exchanger cooled engine*. The two heat exchanger mediums are raw pump water to engine jacket water (engine coolant, i.e., 50% glycol, 50% water). Heat exchanger cooled engines are connected to the engine through a cooling loop that runs from a tap off the pump main discharge line to the inlet to the heat exchanger. The discharge from the heat exchanger is typically piped to a drain, but since this fluid has been heated as it passes through the heat exchanger, the system designer must make sure he can discharge water at this elevated temperature directly to a drain.

This cooling loop typically has a main line with a pressure regulator and DC-actuated solenoid valve and a bypass line with a pressure regulator and a manual bypass valve. The pressure regulator is to prevent the heat exchanger on the engine from becoming overpressurized, and the DC solenoid-actuated valve is to allow only water to flow through the engine heat exchanger when the engine is running. A rule of thumb for minimum raw water through the cooling loop is 1.5 L/min per pump shut-off kW (1/3 gpm per pump shut-off hp). This is typically enough to keep the pump from overheating at maximum power and for cooling the engine. However, the engine supplier should be consulted for the actual amount of cooling water required.

If the fluid being pumped cannot be used to cool the engine, or if water is very scarce and cannot be wasted by discharging to an open drain, then a radiator-cooled engine or air-cooled engine can be used.

Radiator-cooled engines have an internal cooling circuit and require a fan to cool the engine jacket water coolant circulating through the radiator. A radiator-type heat exchanger uses the two mediums of air and jacket water. Air is either pushed (pusher fan) or pulled (suction fan) through the radiator to provide adequate heat transfer and engine cooling. Since often times this fan is driven off the engine, the output power of the engine must be reduced by the fan power draw. Like all centrifugal devices, the fan's power varies with the cube of the speed so the fan curve must be known to determine the amount of power derate.

Air-cooled engines require air to be forced across the engine block for proper engine cooling. Typically the engine has fins or ducts around it to direct the air and provide adequate cooling. An air-cooled engine uses only one medium of airflow to convey away engine heat. The engine does not have jacket water internal cooling. Air is pulled (suction fan) across the engine and then must be discharged away from the engine. Since often times this fan is driven off the engine, the output power of the engine must be reduced by the fan power draw. Like all centrifugal devices, the fan's power varies with the cube of the speed so the fan curve must be known to determine the amount of power derate.

C.1.2.4.4 Enclosures for engine protection

When an enclosure or building is required, the following has to be accounted for by the designer of the enclosure/building:

- Properly sized AC- or DC-operated louver and powered fan, with thermostat automatic control, to maintain the building temperature below 50 °C (120 °F). Typically the thermostat is set at 30 °C (85 °F).

- Total airflow must take into account other heat loads within the enclosure.

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The following formula is normally used to calculate the air flow (in cubic feet per minute [cfm]) to carry away the radiated heat of the engine and discharge outside the enclosure/building:

cfm (air) = Btu/min (radiated heat) divided by (0.018 × Delta T [°F])

Where:

Btu/min comes from the engine supplier and is the engine-radiated heat, and

Delta T is the temperature rise the designer wants to allow for inside the enclosure/building.

- Properly sized space heater with thermostat automatic control to maintain the building above 4 °C (40 °F). Typically the thermostat is set at 18 °C (65 °F).

- Suitable DC-operated louver (i.e., DC solenoid operated to only open when the engine is running) to provide recommended combustion airflow for the engine. This louver should also be as closely situated to the engine air filter as possible.

- For heat exchanger cooled engines, a suitable drain shall be provided to handle the heated water being discharged from the heat exchanger.

- For radiator-cooled engines, a DC-operated louver sized to meet the cooling and combustion airflows of the engine should be installed on the pump side of the enclosure/building. The engine is then equipped with a pusher-type fan. The outlet side of the radiator then needs to be connected to a plenum to allow cooling air to be discharged outside the enclosure/building.

- For air-cooled engines, a DC-operated louver sized to meet the cooling and combustion airflows of the engine should be installed on the installed suction fan side of the engine. Then, on the pump end side of the enclosure/building, another louver must be provided to allow the cooling air to be discharged outside the enclosure/building.

- A suitable silencer (muffler) should also be provided to limit the noise emitted to the environment outside the enclosure/building.

Silencers are manufactured differently to achieve greater levels of attenuation:

a) An industrial silencer typically attenuates overall sound levels by 10–15 dBa (approximate).

b) A residential silencer typically attenuates overall sound levels by 15–25 dBa (approximate).

c) A hospital or critical grade rated silencer typically attenuates overall sound levels by 20–30 dBa (approximate).

d) A flexible exhaust connection member should always be used between the engine and the exhaust system. This flexible member allows for thermal expansion of the exhaust system and keeps engine vibration from being transmitted to the exhaust system that is normally supported on hangers within the enclosure/building.

e) The silencer is typically mounted outdoors either on the side or roof of the enclosure/building. Note: Safety design considerations must be followed due to the extreme temperatures associated with the exhaust system.

The exhaust system within the building is normally insulated to keep the amount of radiated heat within the enclosure/building to a minimum as well as for personnel safety. Exhaust systems can be exposed to temperatures that range from 315 to 650 °C (600 to 1200 °F).

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C.1.2.5 Engine operation

Operating speeds

Engines are variable-speed drivers, but typically for pump applications, they are set to run at one speed. Most diesel engines are rated between 1500 and 2500 rpm. A power rating for an engine is typically displayed as a curve with speed as the abscissa and power as the ordinate. Although there are a few diesel engines that run as high as 3600 rpm (two-pole motor synchronous speed), engine applications at 3600 rpm and higher may be achieved by using a spark ignition (gasoline-fueled) engine.

When an engine is set to run at a preset speed for its rated power condition, and this speed is "locked," and then, when the applied pump load is below the engine rated power condition, the engine speed will increase. The more the pump load is reduced, the higher the engine speed will go. This condition is known as *engine droop*. This is very similar to slip on an electric motor. The engine manufacturer can supply an engine with different amounts of droop typically ranging from 0–10%. This is another factor that must be taken into account during the quote phase.

A specific engine may have different power/speed ratings based on its use as follows:

- Engines used on emergency, standby service typically have the highest power ratings. These engines typically run less than 100 hours per year.
- Engines on standby service should be set up for weekly exercise of no more than half an hour to ensure it will be functional when called on during an emergency.
- Engines used for intermittent duty service typically run between 100 and 1000 hours per year and typically have power/speed ratings less than those of an emergency, standby service engine and more than those of a continuous duty service rated engine.
- Engines rated for continuous duty service, greater than 1000 hours per year, have the lowest ratings.
- In all cases, engine manufacturers typically keep a 10% power reserve above ratings quoted.

Diesel engines can be either naturally aspirated or turbo-charged.

Turbo-charging is a way of increasing the horsepower developed by the engine as a function of speed.

Other modifications can be made to the engine to increase engine horsepower as follows:

- Turbo-charged/jacket water intercooled
- Turbo-charged/supercharged/jacket water intercooled
- Turbo-charged/air to air-cooled (charge – air – cooled) – for radiator-cooled engines
- Turbo-charged/raw water to air-cooled (charge – air – cooled) – for heat-exchanger-cooled engines

NOTE: As the above modifications are made to the engine to increase horsepower, the materials of some of the component pieces within the engine may also have to be upgraded to stronger materials.

C.1.2.5.1 Speed governing

Diesel engines have been traditionally mechanically governed, however, due to the US Environmental Protection Agency (EPA) requiring the engine manufacturers to design for lower and lower levels of emissions (these levels are referred to as *tiers*, i.e., Tier 1, 2, and 3, and they change by engine power rating and by the year they go into effect), most diesel engines have been converted over to be electronically governed. It is important to know when

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selecting a diesel engine driver if it *must* meet current EPA limits for nonroad engines (Reference 40 CFR Part 89 Sub-parts A-K for land-based nonroad applications). There are currently applications that are exempt from the non-road EPA emissions limits, such as fire pump engine drivers. In addition, if the engine is going offshore, EPA emissions limits may not apply. Since there are some engines that can meet EPA emissions limits with either mechanically governed or electronically governed engines, it is very important that the engine supplier know if the engine is exempt from the EPA emissions levels at the time of quoting the engine. Being exempt from having to meet emissions levels can dramatically affect the engine supplied and its cost.

C.1.2.5.2 Engine starting

The most common means of starting an engine is with an electric motor starter. For some types of service, such as fire pumps, two electric starters, each connected to an independent battery supply, is required (or one starter connected to two separated start solenoids that are then connected to their own battery supplies). The most common engine voltage employed is 12 volts (V) DC (for engines rated less than 300 kW [400 bhp] approximately). Typically 24 V DC is employed on engines rated above 300 kW (400 bhp).

When electric starters are used, AC-powered battery chargers are normally part of the installation to ensure batteries stay charged. This is especially true on engines rated for emergency, standby service or intermittent duty service. However, for engines that run continuously, the engine can be equipped with an alternator to charge the battery or batteries while it is running.

Other types of starting devices used follow:

- Air starting – requires complete system of compressed-air sized for a specific number of starts
- Hydraulic starting – requires a complete system of pressurized hydraulic oil sized for a specific number of starts
- Inertia start

A diesel engine starts better if the engine block is maintained at a minimum of 50 °C (120 °F). Block heaters or engine jacket water heaters should be employed to maintain the block temperature. Block heaters and engine jacket water heaters are typically connected to an AC source.

It is recommended that if the temperature the engine will be subjected to can be less than 4 °C (40 °F), then the engine (and its fuel supply) should be installed in a suitable enclosure or building.

C.1.2.5.3 Condition monitoring

It is advisable to fit the engine with an instrument panel that has, as a minimum, the following gauges (note that sensors and senders need to be mounted to the engine for these gauges to function):

- A toggle or key start for starting the engine
- A voltmeter or ammeter
- A tachometer
- An hour meter
- A speed switch for automatic crank termination and for overspeed shutdown (optional)
- An alarm for warning of high jacket water temperature or low oil pressure

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- As an option, a shut-down device can be employed to shut down the engine on high jacket water temperature or low oil pressure

C.1.2.6 Engine testing and performance considerations

Engines are tested and rated in accordance with SAE J1349, Engine Power Test Code. All test results are corrected to 750 mm (29.61 in) Hg barometer (approximately 90 m [300 ft] above sea level) and 25 °C (77 °F). Engines must be derated for altitude and temperatures above those stated above for specific applications. Common derate factors often used are a 3% reduction for each 305 m (1000 ft) of altitude above 90 m (300 ft), and 1% reduction for every 8 °C (10 °F) above 25 °C (77 °F) ambient temperature.

Engines are also rated on a specific grade of diesel fuel. The most common diesel fuel is diesel fuel #2, however, diesel fuel #1, blended diesel #1 and #2, and jet fuels (JP-4 and JP-8) are other common fuels used to fuel diesel engines. The power output rating of the engines is based on a specific grade of diesel fuel, with its known specific gravity. Using a different fuel, with a lower specific gravity, will also lower the power output of engine. As a rule of thumb, the engine power is directly proportional to the ratio of the specific gravity of the diesel fuel used in the engine to the fuel that the engine was rated on.

Diesel fuel #2, the most common diesel fuel that engines are rated for, has a specific gravity of:

Diesel fuel #2-D (at 15 °C)

= 849.6–860.2 kg/m^3

= Gravity of 35–33 API

= 0.85 specific gravity (average)

= 7.13 lb/gal

Diesel fuel #2 has a heating value of 43,000 - 45,000 kJ/kg (18,500 - 19,300 Btu/lb).

C.1.2.7 Package design considerations

The following identifies the information that a designer of a complete engine-driven pump package needs to know (this information comes from the engine supplier):

- Engine type of service (standby, intermittent, or continuous) and its power/speed rating.
- Engine/pump rpm to determine specific engine power at pump rated speed and operating limitations at reduced speed conditions.
- Any possible engine power derates for ambient air or for altitude.
- Whether the engine is heat exchanger cooled, radiator cooled, or air cooled, in order to employ the proper design of the enclosure/building.
- For a heat exchanger engine, the amount of raw water needed for cooling the engine.
- The volume flow rate of air for combustion.
- The volume flow rate of air for cooling the engine if radiator or air-cooled.
- The minimum cold cranking amps of the battery (if electric start).

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- The volume flow rate of exhaust gas.
- The maximum allowable engine exhaust backpressure. This value must not be exceeded when designing the exhaust system, which consists of a certain number of elbows, straight pipe and silencer of a given pipe size. The engine supplier can typically help the designer calculate the backpressure for a given exhaust system and advise if the system is too small.
- Engine radiated heat.
- Engine fuel consumption at rated power.

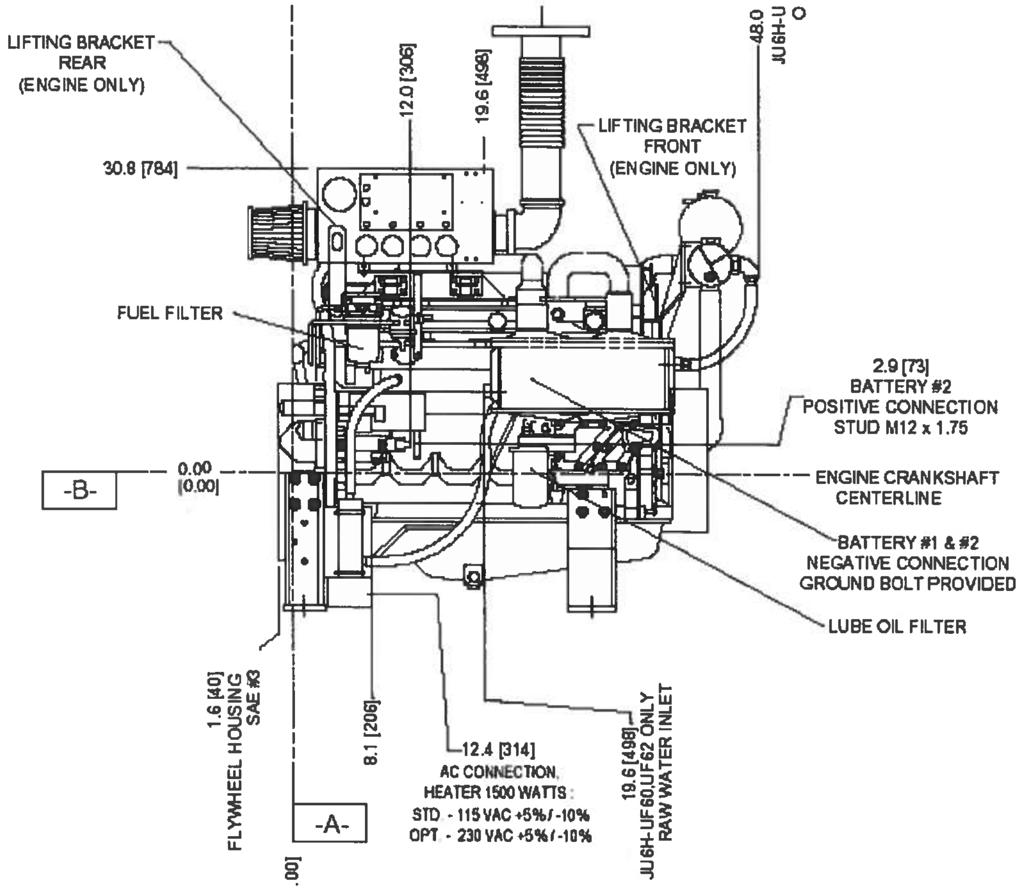

Figure C.3 — Outline of an internal combustion engine

Special considerations

For all engine/pump applications, a mass elastic analysis (torsional natural frequency analysis) of the drivetrain should be performed. The engine supplier or the coupling supplier can perform this calculation. This analysis will show if there are any torsional natural frequencies being excited at or near the pump operating speed, and if there are, they will be able to recommend solutions.

C.1.3 Steam turbine

Steam turbine ratings are specified for particular inlet and exhaust steam conditions. Sizing is usually based on maximum continuous power required. Overloads, if applicable, are accommodated by allowing the turbine to slow down or by opening additional steam admission valves. The expected load range and the desired load response should be given to the turbine manufacturer.

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C.1.4 Eddy current drives

In rotodynamic pump drivetrains, magnetic drives are used to vary pump speed. Losses to be taken into account in determining drive rating are slip, windage, etc. Eddy current drive selection is based on the torque to be transmitted and slip loss dissipation capacity.

C.1.5 Deceleration devices

In some applications, it is desirable to provide additional rotating inertia in 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 pressure for a longer than normal interval. This allows more time for check valves and other flow-control devices to work. The result is less likelihood of damaging backflow through the pump or water hammer effects on the system.

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

Typically, the moment of inertia of the flywheel might be equal to 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 may exceed the starting capabilities of the driver.

Flywheel applications should be carefully analyzed to match need and performance before they are installed.

C.1.6 Variable-speed drives and gears

Rotodynamic pumps can be accurately matched to system requirements that vary with time either by throttling the pump discharge or changing drive speed. Control of the direct connected pump and driver speed can be manual or automatic. Variable-speed drivers may be AC or DC electric motors, steam or gas turbines, internal combustion engines, variable-frequency drives, or other devices. Intermediate variable-speed hydraulic and mechanical drives are also used between the pump and a constant-speed driver. For the pump manufacturer to make a proper selection, the operating conditions over the full range of rates of flow and heads must be provided. Suction conditions over the full range are most important. Possible vibration problems must be checked over the full speed range by the manufacturer for all components in the set. The user must consider mounting arrangements more carefully than for a constant-speed unit. The driver and its automatic control when used must also be matched carefully to each other and the system requirements.

Rotodynamic pumps may be operated at speeds other than the driver speed by using intermediate gears. The maximum permissible pump speed must not be exceeded. The lubrication system for gears requires careful maintenance. Cooling is often required using air or water. Noise levels may be a consideration when gears are employed. A user choosing a gear must check pump operating speeds with the pump's manufacturer to avoid operating at or near critical speeds.

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Appendix D

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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 E

Index

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

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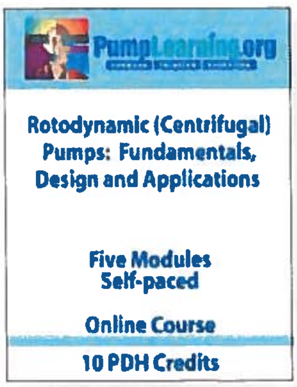

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