Centrifugal Pumps for Petroleum, Petrochemical, and Natural Gas Industries

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Introduction

It is necessary that users of this standard be aware that further or differing requirements can be needed for individual applications. This standard is not intended to inhibit a vendor from offering, or the purchaser from accepting, alternative equipment or engineering solutions for the individual application. This can be particularly appropriate where there is innovative or developing technology. Where an alternative is offered, it is necessary that the vendor identify any variations from this standard and provide details.

A bullet [•] at the beginning of a subsection or paragraph indicates that either a decision is required or further information is to be provided by the purchaser.

This standard shows U.S. customary (USC) units with other units in parentheses for information.

Centrifugal Pumps for Petroleum, Petrochemical, and Natural Gas Industries

1 Scope

This standard specifies requirements for new centrifugal pumps, including pumps running in reverse as hydraulic power recovery turbines (HPRTs), for use in petroleum, petrochemical, and gas industry process services.

This standard is applicable to overhung pumps, between-bearings pumps, and vertically suspended pumps (see Table 1). Section 9 provides requirements applicable to specific types of pumps. All other sections of this standard are applicable to all pump types. Illustrations are provided of the various specific pump types and the designations assigned to each specific type.

Relevant industry operating experience suggests pumps produced to this standard are suitable for pumping liquids at conditions exceeding any one of the following:

— discharge pressure (gauge): 275 psi; 19.0 bar (1900 kPa);

— suction pressure (gauge): 75 psi; 5.0 bar (500 kPa);

— pumping temperature: 300 °F (150 °C);

— rotational speed: 3600 r/min;

— rated total head: 400 ft (120 m);

— impeller diameter, overhung pumps 13 in. (330 mm).

NOTE 1 For sealless pumps, reference can be made to API 685.

NOTE 2 For pump repairs, reference can be made to API 697.

2 Normative References

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any addenda) applies.

API Standard 20S, *Additively Manufactured Metallic Components for Use in the Petroleum and Natural Gas Industries*

API Standard 5L, *Line Pipe*

API Standard 541, *Form-wound Squirrel Cage Induction Motors—375 kW (500 Horsepower) and Larger*

API Standard 547, *General Purpose Form-wound Squirrel Cage Induction Motors—185 kW (250 hp) through 2240 kW (3000 hp)*

API Standard 611, *General-purpose Steam Turbines for Petroleum, Chemical, and Gas Industry Services*

API Standard 612, *Special Purpose Steam Turbines*

API Standard 613, *Special Purpose Gears*

API Standard 614, *Lubrication, Shaft-sealing and Oil-control Systems and Auxiliaries, 6th Edition*

API Standard 670, *Machinery Protection Systems*

API Standard 671, *Special-purpose Couplings for Petroleum, Chemical, and Gas Industry Services*

API Standard 677, *General-purpose Gear Units for Petroleum, Chemical and Gas Industry Services*

API Standard 682, *Pumps—Shaft Sealing Systems for Centrifugal and Rotary Pumps*

API Recommended Practice 686,*Machinery Installation and Installation Design*

ANSI B11.19-2010, [1] *Performance Criteria for Safeguarding*

ABMA 7, [2] *Shaft and Housing Fits for Metric Radial Ball and Roller Bearings (Except Tapered Roller Bearings) Conforming to Basic Boundary Plan*

AGMA 9000,[3] *Flexible Couplings—Potential Unbalance Classification*

AGMA 9002, *Bores and Keyways for Flexible Couplings (Inch Series)*

ASME B1.1,[4] *Unified Inch Screw Threads (UN, UNR, and UNJ Thread Forms)*

ASTM A53, *Standard Specification for Pipe, Steel, Black and Hot-Dipped, Zinc-Coated, Welded and Seamless*

ASTM A106, *Standard Specification for Seamless Carbon Steel Pipe for High-Temperature Service*

AWS D1.1, *Structural Welding Code—Steel*

ASME B1.13M, *Metric Screw Threads: M Profile*

ASME B16.5, *Pipe Flanges and Flanged Fittings: NPS 1/2 through NPS 24 Metric/Inch Standard*

ASME B16.11, *Forged Fittings, Socket-Welding and Threaded*

ASME B16.47, *Larger Diameter Steel Flanges NPS 26 Through NPS 60 Metric/Inch Standard*

ASME B18.18.2M, *Inspection and Quality Assurance for High-Volume Machine Assembly Fasteners*

ASME B31.3, *Process Piping*

ASME *Boiler and Pressure Vessel Code (BPVC) Section V, Nondestructive Examination*

ASME *Boiler and Pressure Vessel Code (BPVC) Section VIII, Rules for Construction of Pressure Vessels*

ASME *Boiler and Pressure Vessel Code (BPVC) Section IX, Welding, Brazing and Fusing Qualifications*

DIN 910 ,[5] *Hexagon head screw plugs with collar—Cylindrical thread*

EN 1092-1, *Flanges and their joints—Circular flanges for pipes, valves, fittings and accessories, PN designated—Steel flanges*

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1 American National Standards Institute, 25 West 43rd Street, 4th Floor, New York, New York 10036, www.ansi.org.

2 American Bearing Manufacturers Association, 1001 N. Fairfax Street, Suite 500, Alexandria, VA 22314, www.americanbearings.org.

3 American Gear Manufacturers Association, 1001 N. Fairfax Street, Suite 500, Alexandria, VA 22314-1587, www.agma.org.

4 ASM International, 9639 Kinsman Road, Materials Park, Ohio 44073, www.asminternational.org.

5 DIN Deutsches Institut für Normung e. V., Saatwinkler Damm 42/43, 13627 Berlin, Germany, www.din.de.

EN 13445 (all parts), *Unfired pressure vessels*

EN 13445-4, *Unfired pressure vessels—Fabrication*

HI 14.6 ,[6] *Rotodynamic Pumps for Hydraulic Performance Acceptance Tests*

IEC 60034-1,[7] *Rotating electrical machines—Part 1: Rating and performance*

IEC 60034-2-1, *Rotating electrical machines—Part 2-1: Standard methods for determining losses and efficiency from tests (excluding machines for traction vehicles)*

IEC 60079 (all parts), *Explosive atmospheres*

IEEE 841 ,[8] *Petroleum and Chemical Industry—Premium-Efficiency, Severe Duty, Totally Enclosed Fan-Cooled (TEFC) Squirrel Cage Induction Motors—Up to and Including 370 kW (500 hp)*

ISO 228-1,[9] *Pipe threads where pressure-tight joints are not made on the threads—Part 1: Dimensions, tolerances and designation*

ISO 261, *ISO general purpose metric screw threads—General plan*

ISO 262, *ISO general purpose metric screw threads—Selected sizes for screws, bolts and nuts*

ISO 281:2007, *Rolling bearings—Dynamic load ratings and rating life*

ISO 286 (all parts), *System of limits and fits*

ISO 3117, *Tangential keys and keyways*

ISO 3183, *Petroleum and natural gas industries—Steel pipe for pipeline transportation systems*

ISO 4200, *Plain end steel tubes, welded and seamless—General tables of dimensions and masses per unit length*

ISO 4386, *Plain Bearings – Metallic multilayer plain bearings*

ISO 5753 (all parts), *Rolling bearings—Internal clearance*

ISO 8501 (all parts), *Preparation of steel substrates before application of paints and related products—Visual assessment of surface cleanliness*

ISO 9606 (all parts), *Qualification testing of welders—Fusion welding*

ISO 9906, *Rotodynamic pumps—Hydraulic performance acceptance tests—Grades 1, 2 and 3*

ISO 10441, *Flexible couplings for mechanical power transmission—Special-purpose applications*

ISO 14120, *Safety of machinery—Guards—General requirements for the design and construction of fixed and movable guards*

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

7 International Electrotechnical Commission, 3 rue de Varembé, 1st Floor, PO Box 131, CH-1211 Geneva 20, Switzerland, www.iec.ch.

8 Institute of Electrical and Electronics Engineers, 445 Hoes Lane, Piscataway, New Jersey 08854, www.ieee.org.

9 International Organization for Standardization, 1, ch. de la Voie-Creuse, Case postale 56, CH-1211 Geneva 20, Switzerland, www.iso.org.

ISO 14691, *Petroleum, petrochemical and natural gas industries—Flexible couplings for mechanical power transmission—General-purpose applications*

ISO 15607, *Specification and qualification of welding procedures for metallic materials—General rules*

ISO 15609 (all parts), *Specification and qualification of welding procedures for metallic materials—Welding procedure specification*

ISO 15649, *Petroleum and natural gas industries—Piping*

ISO 21940-11, *Mechanical vibration—Rotor balancing—Procedures and tolerances for rotors with rigid behaviour*

ISO 21940-12, *Mechanical vibration—Rotor balancing—Procedures and tolerances for rotors with flexible behaviour*

MSS SP-55 ,[10] *Quality Standard for Steel Castings for Valves, Flanges, Fittings and Other Piping Components—Visual Method for Evaluation of Surface Irregularities*

NACE MR0103 ,[11] *Materials Resistant to Sulfide Stress Cracking in Corrosive Petroleum Refining Environments*

NACE MR0175, *Guidelines for Corrosion-Resistant Materials*

ISO 15156 (all parts), *Petroleum and Natural Gas Industries—Materials for Use in H_2S-containing Environments in Oil and Gas Production*

NEMA MG-1, *Motors and Generators*

SSPC SP 6 ,[12] *Commercial Blast Cleaning*

3 Terms, Definitions, Acronyms, and Abbreviations

3.1 Terms and Definitions

For the purposes of this document, the following definitions apply.

3.1.1
allowable operating region
Portion of a pump's hydraulic range over which the pump is allowed to operate, based on vibration within the upper limit of this standard or temperature rise or other limitation, specified by the manufacturer. The allowable operating region includes the preferred operating region. Outside the preferred operating region, the service life of a pump, although reduced, is within design limits.

3.1.2
axially split
Joint split with the principal casing face parallel to the shaft centerline.

3.1.3
barrel pump
Horizontal pump of the double-casing type.

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[10] Manufacturers Standardization Society of the Valve and Fittings Industry, Inc., 127 Park Street, NE, Vienna, Virginia 22180-4602, www.mss-hq.com.

[11] NACE International (formerly the National Association of Corrosion Engineers), 1440 South Creek Drive, Houston, Texas 77084-4906, www.nace.org.

[12] The Society for Protective Coatings, 40 24th Street, 6th Floor, Pittsburgh, Pennsylvania 15222, www.sspc.org.

3.1.4
best efficiency point
BEP
Flowrate at which a pump achieves its highest efficiency at rated impeller diameter.

3.1.5
bundle
Assembly of the rotor plus the internal stationary parts of a centrifugal pump, less the heads, bearing housings, and seals.

3.1.6
cartridge bundle assembly
Assembly consisting of the complete pump excluding the casing.

NOTE It is composed of the inner barrel assembly, end heads, seals, bearing housings, rotor, and bearings.

3.1.7
critical speed
Shaft rotational speed at which the rotor-bearing-support system is in a state of resonance.

3.1.8
datum elevation
Elevation to which values of net positive suction head (NPSH) are referred (refer to 6.1.13 and 6.1.14).

3.1.9
double casing
Type of pump construction in which the pressure casing is separate from the bundle inside the casing.

NOTE Double casing pumps include BB5, VS6, VS7 pumps.

3.1.10
drive-train component
Item of the equipment used in series to drive the pump.

EXAMPLES—Motor, gear, turbine, engine, fluid drive, clutch.

3.1.11
dry critical speed
Rotor critical speed calculated assuming that there are no liquid effects, that the rotor is supported only at its bearings, and that the bearings are of infinite stiffness.

3.1.12
high energy pump
Pumps with heads greater than 650 ft (200m) per stage and with more than 300 hp (225 kW) per stage.

3.1.13
high pressure seal
API 682 mechanical seal that is designed to operate at or above 600 psi (40 bar).

3.1.14
hydraulic power recovery turbine
HPRT
Turbomachine designed to recover power from a fluid stream. It can be arranged as a pump running in reverse or radial inflow liquid machine.

3.1.15
hydrodynamic bearing
Bearing that uses the principles of hydrodynamic lubrication.

3.1.16
identical
Pump of the same size, hydraulic design, number of stages, rotational speed, clearances, materials, type of shaft seal, type of bearings, coupling mass, coupling overhang, and pumping the same liquid.

3.1.17
maximum allowable working pressure
MAWP
Maximum continuous pressure for which the manufacturer has designed the pump (or any part to which the term is referred) while pumping the specified liquid at the specified maximum operating temperature (does not include mechanical seal).

3.1.18
maximum continuous speed
MCS
Highest speed at which the manufacturer's design permits continuous operation.

3.1.19
maximum discharge pressure
Highest pressure produced by the pump with the furnished impeller(s) at any flow point from zero to maximum flow while operating at any of the continuous speeds up to trip speed, with the liquid relative density (specific gravity) pumped and the maximum suction pressure provided for each respective operating case.

3.1.20
maximum operating temperature
Highest temperature of the pumped liquid, including upset conditions, to which the pump is exposed.

3.1.21
maximum suction pressure
Highest suction pressure to which the pump is subjected during operation (non-transient; does not include water hammer).

3.1.22
minimum continuous stable flow
MCSF
Lowest flow at which the pump can continuously operate without exceeding the vibration limits imposed by this standard.

3.1.23
minimum continuous thermal flow
Lowest flow at which the pump can continuously operate without its operation being impaired by the temperature rise of the pumped liquid.

3.1.24
minimum design metal temperature
Lowest mean metal temperature (through the thickness) expected in service, including operation upsets, auto-refrigeration, and temperature of the surrounding environment, for which the equipment is designed.

3.1.25
multistage pump
Pump with three or more stages. Refer to 4.2.1

3.1.26
net positive suction head
NPSH
Absolute inlet total head above the head equivalent to the vapor pressure referred to the NPSH datum plane.

3.1.27
net positive suction head available
NPSHA
NPSH determined by the purchaser for the pumping system with the liquid at the rated flow.

3.1.28
net positive suction head required
NPSH3
NPSH that results in a 3 % loss of head (first-stage head in a multistage pump) determined by the vendor by testing with water.

3.1.29
nominal pipe size
NPS
Designation, usually followed by a size designation number, corresponding approximately to the outside diameter (OD) of the pipe and is expressed in inches.

3.1.30
normal operating point
Point at which the pump typically operates under expected process conditions.

3.1.31
normal-wear part
Part normally restored or replaced at each pump overhaul.

EXAMPLES—Wear rings, inter-stage bushings, balancing device, throat bushing, seal faces, bearings, and gaskets.

3.1.32
observed
Inspection or test where the purchaser is notified of the timing of the inspection or test and the inspection or test is performed as scheduled even if the purchaser or the purchaser's representative is not present.

3.1.33
oil-mist lubrication
Lubrication provided by oil mist produced by atomization and transported to the bearing housing, or housings, by compressed air.

3.1.34
operating region
Portion of a pump's hydraulic coverage over which the pump operates.

3.1.35
overhung pump
Pump whose impeller is supported by a cantilever shaft from its bearing assembly.

3.1.36
owner
Final recipient of the equipment who may delegate another agent as the purchaser of the equipment.

3.1.37
pipeline pump
Those pumps transporting liquids between two facilities. This excludes pumps within the process plant perimeter. Refer to 6.2.16, 6.4.3.5, 6.10.1.3, 7.3.3.2, and 7.6.2.12.

3.1.38
positive material identification testing
PMI testing
Any physical evaluation or test of a material to confirm that the material which has been or will be placed into service is consistent with the selected or specified alloy material designated by the owner. These evaluations or tests may provide either qualitative or quantitative information that will verify the nominal alloy composition.

3.1.39
preferred operating region
POR
Portion of a pump's hydraulic range over which the pump's vibration is within the base limit of this standard.

3.1.40
pressure casing
Composite of all stationary pressure-containing parts of the pump, including all nozzles, seal glands, seal chambers, and auxiliary connections but excluding the stationary and rotating members of mechanical seals. The atmospheric side of the seal gland, the seal flush (piping) plan, auxiliary piping, and valves are not part of the pressure casing.

3.1.41
purchaser
Owner, or owner's agent, who issues the order and specification to the vendor.

3.1.42
pure oil-mist lubrication
(dry sump)
System in which the mist both lubricates the bearing(s) and purges the housing and there is no oil level in the sump.

3.1.43
purge oil-mist lubrication
(wet sump)
Systems in which the mist only purges the bearing housing.

3.1.44
radially split
Joint split with the principal casing face perpendicular to the shaft centerline.

3.1.45
rated operating point
Point at which the vendor certifies that pump performance is within the tolerances stated in this standard. This is also called the guaranteed point.

3.1.46
relative density
specific gravity
Property of a liquid expressed as the ratio of the liquid's density to that of water 62.4 lb/ft^3 (1000 kg/m^3)

3.1.47
rotor
Assembly of all the rotating parts of a centrifugal pump.

3.1.48
spark-resistant material
Material that is not prone to generate impact sparks under conditions of use.

3.1.49
specific speed
Ns
Index relating flow, total head, and rotational speed for pumps of similar geometry.

3.1.50
stage
One impeller and associated diffuser or volute and return channel, if required.

3.1.51
subcritical
Rotor characterized by the first dry critical speed being above the pump's maximum continuous speed plus a defined separation margin.

3.1.52
suction-specific speed
Nss or S
Index relating flow, NPSH3, and rotational speed for pumps of similar geometry.

3.1.53
system design pressure
The pressure rating of the piping system components upstream and downstream of the pump (e.g. heat exchangers, piping, valves)

3.1.54
throat bushing
Device that forms a restrictive close clearance around the sleeve (or shaft) between the seal or inner seal of a dual seal cartridge and the impeller.

3.1.55
total indicator reading
total indicated runout
TIR
Difference between the maximum and minimum readings of a dial indicator or similar device, monitoring a face or cylindrical surface, during one complete revolution of the monitored surface.

3.1.56
trip speed
(electric motor driver)
Electric motor driver-synchronous speed at maximum supply frequency.

3.1.57
trip speed
(adjustable speed driver)
Speed at which the independent emergency overspeed device operates to shut down the driver.

3.1.58
unit responsibility
Obligation for coordinating the documentation, delivery, and technical aspects of all the equipment and all auxiliary systems included in the scope of the order.

3.1.59
vendor
Manufacturer or manufacturer's agent that supplies the equipment and is normally responsible for service support.

3.1.60
vertical in-line pump
Vertical-axis, single-stage overhung pump whose suction and discharge connections have a common centerline that intersects the shaft axis.

3.1.61
vertically suspended pump
Vertical-axis pump whose liquid end is suspended from a column and mounting plate.

3.1.62
wet lateral critical speed
Rotor critical speed calculated considering the additional support, added mass, and damping produced by the action of the pumped liquid within internal running clearances at the operating conditions and allowing for stiffness and damping within the bearings.

3.1.63
witnessed
Inspection or test for which the purchaser is notified of the timing of the inspection or test and a hold is placed on the inspection or test until the purchaser or the purchaser's representative is in attendance.

3.2 Acronyms and Abbreviations

AFNOR	Association Française de Normalization
AISI	American Iron and Steel Institute
AFD	adjustable frequency drive
ASD	adjustable speed drive
AUS	austenitic stainless steel
AVM	anti-vibration mount
BB	between-bearings
BEP	best efficiency point
CR	chrome
DBSE	distance between shaft ends
DEA	diethylamine
DN	diameter nominal
EDE	electronic data exchange
FCC	fluid catalytic cracker
FEA	finite element analysis
FFT	fast Fourier transform
HPRT	hydraulic power recovery turbine
ID	inside diameter
LNG	liquefied natural gas
MAWP	maximum allowable working pressure
MEA	monoethylamine

MEK	methylethylketone
MT	magnetic particle inspection
NDT	nondestructive testing
NPS	nominal pipe size
NPSH	net positive suction head
NPSHA	net positive suction head available
NPSHi	incipient net positive suction head
NPSH3	net positive suction head at 3 % head loss
N_s	specific speed
N_{ss}	suction-specific speed
OD	outside diameter
OH	overhung
PFA/CF	polyfluoroethylene with carbon fiber
PH	precipitation hardened
PMI	positive material identification
PRE	pitting resistance equivalency
PT	liquid penetrant inspection
PTFE	polytetrafluoroethylene
PWHT	postweld heat treatment
RMS	root mean square
RT	radiographic inspection
S	suction-specific speed
SDS	safety data sheet
SFI	shaft flexibility index
SG	specific gravity
SPL	sound pressure level
STL	carbon steel
TEA	triethylamine
TFE	tetrafluoroethylene
TIR	total indicator reading; total indicated runout
UN	Unified constant pitch (a thread form)
UNC	Unified National Coarse (a thread form)
UNS	Unified Numbering System (for metal and alloys)
UT	ultrasonic examination
VDDR	vendor drawing and data requirements
VI	visual inspection
VOC	volatile organic compound
VS	vertically suspended

4 General

4.1 Unit Responsibility

The pump vendor shall assume unit responsibility and shall assure that all subvendors comply with the requirements of this standard and all reference documents. The technical aspects to be considered by the vendor include, but are not limited to, such factors as the power requirements, speed, rotation, general arrangement, couplings, dynamics, lubrication, sealing system, material test reports, instrumentation, piping, conformance to specifications, and testing of components.

4.2 Classification and Designation

4.2.1 Description of Codes

The pumps described in this standard are classified and designated by type codes, as shown in Table 1.

Table 1—Pump Classification Type Identification

Pump Type [a]			Description		Type Code
Centrifugal pumps	Overhung	Flexibly coupled	Horizontal	Foot-mounted	OH1
				Centerline-mounted	OH2
			Vertical in-line with bearing bracket	—	OH3
		Rigidly coupled	Vertical in-line	—	OH4
		Close-coupled	Vertical in-line	—	OH5
			High-speed integrally geared	—	OH6
	Between-bearings	1- and 2-stage	Axially split	Foot-mounted	BB1-A
			Axially split	Near-centerline mounted	BB1-B
			Radially split	Centerline-mounted	BB2
		Multistage	Axially split	Near-centerline mounted	BB3
			Radially split	Single casing	BB4
				Double casing	BB5
	Vertically suspended	Single casing	Discharge through column	—	VS1
				—	VS2
				—	VS3
			Separate discharge pipe	Line shaft	VS4
				Cantilever shaft	VS5
		Double casing	Radially split	—	VS6
				—	VS7

[a] Illustrations of the various types of pumps are provided in 4.2.2.

4.2.2 Pump Designations and Descriptions

4.2.2.1 General

The figures that follow in this section are generic and informative. They are not meant to show construction details or specific features.

4.2.2.2 Pump Type OH1

Foot-mounted, single-stage overhung pumps with a single bearing housing shall be designated pump type OH1 (Figure 1).

NOTE This type does not meet all the requirements of this standard; see Table 4.

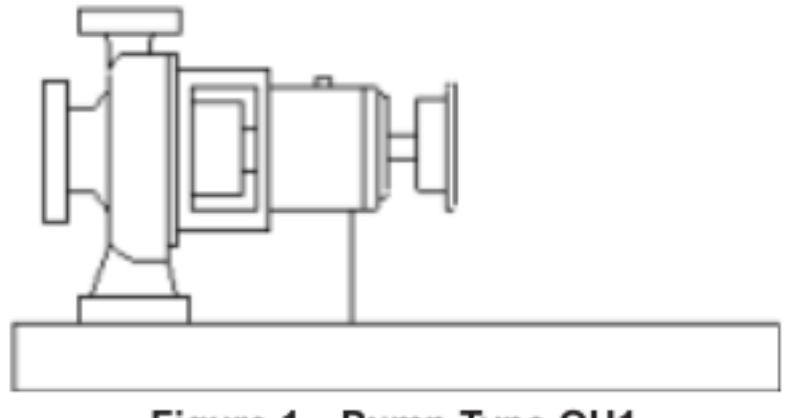


Figure 1—Pump Type OH1

4.2.2.3 Pump Type OH2

Centerline-mounted, single-stage overhung pumps with a single bearing housing shall be designated pump type OH2 (Figure 2).

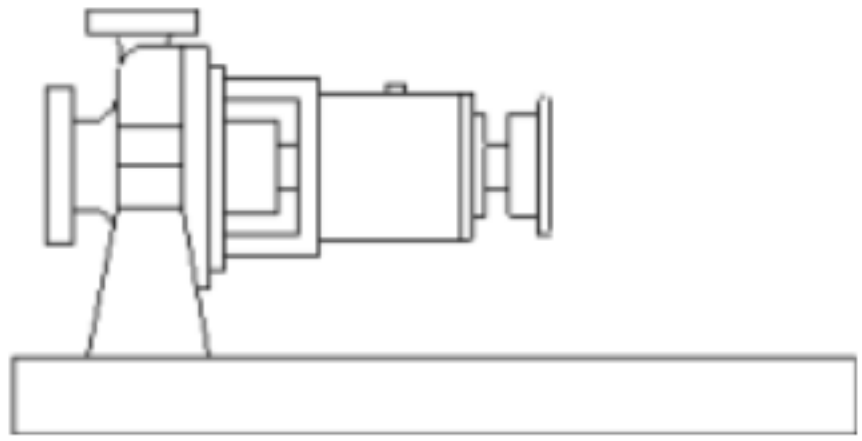


Figure 2—Pump Type OH2

4.2.2.4 Pump Type OH3

Vertical, in-line, single-stage overhung pumps with only one bearing housing and separate bearing brackets shall be designated pump type OH3 (Figure 3). The driver is usually mounted on a support integral to the pump. The pumps and their drivers are flexibly coupled. The pump may be mounted directly in the piping or support-mounted.

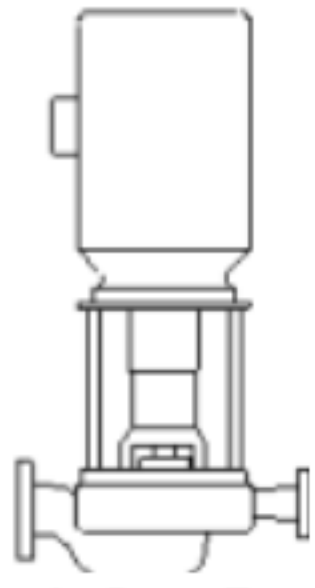


Figure 3—Pump Type OH3

4.2.2.5 Pump Type OH4

Rigidly coupled, vertical, in-line, single-stage overhung pumps shall be designated pump type OH4 (Figure 4). Rigidly coupled pumps have their shaft rigidly coupled to the driver shaft. The pump may be mounted directly in the piping or support mounted.

NOTE This type does not meet all the requirements of this standard; refer to Table 4.

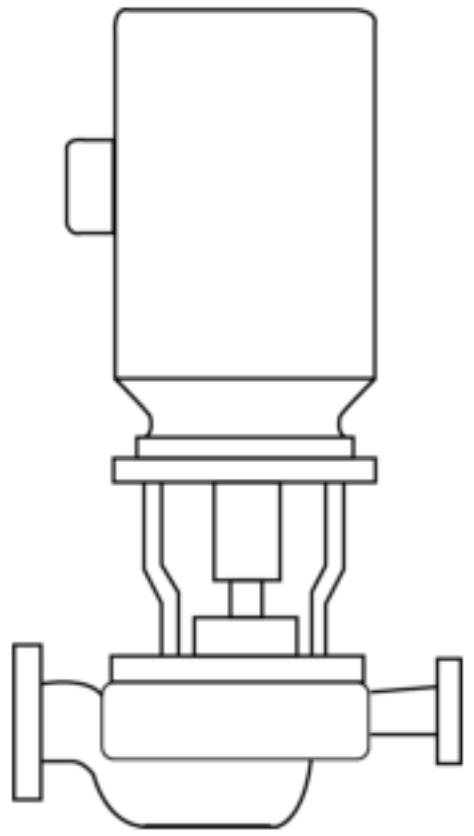


Figure 4—Pump Type OH4

4.2.2.6 Pump Type OH5

Close-coupled, vertical, in-line, single-stage overhung pumps shall be designated pump type OH5 (Figure 5). Close-coupled pumps have their impellers mounted directly on the driver shaft. The pump may be mounted directly in the piping or support mounted.

NOTE This type does not meet all the requirements of this standard; refer to Table 4.

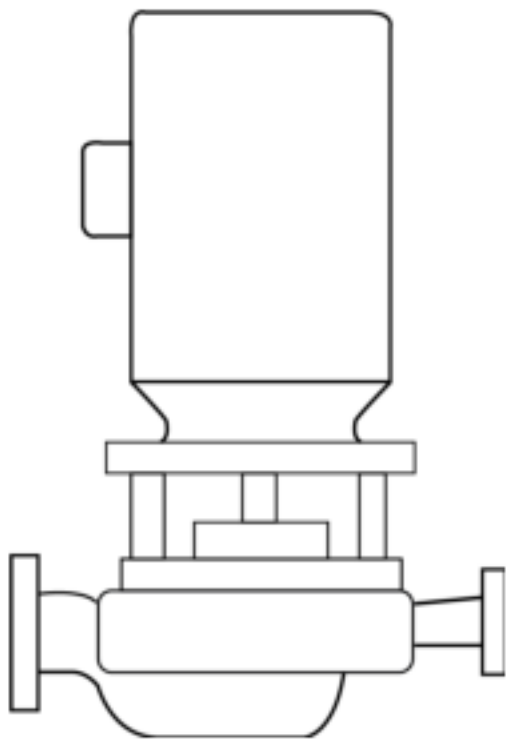


Figure 5—Pump Type OH5

4.2.2.7 Pump Type OH6

High-speed, integral, gear-driven, single-stage overhung pumps shall be designated pump type OH6 (Figure 6). These pumps have a speed-increasing gearbox integral with the pump. The impeller is mounted directly to the gearbox output shaft. There is no coupling between the gearbox and pump; however, the gearbox is flexibly coupled to its driver. The pumps can be oriented vertically or horizontally. The pump is typically support-mounted.

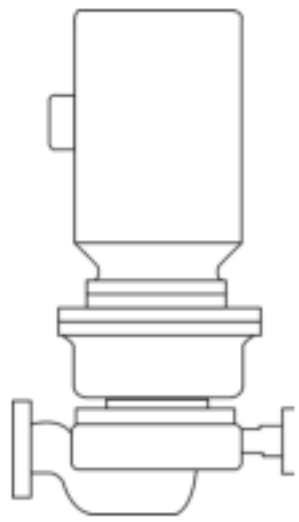
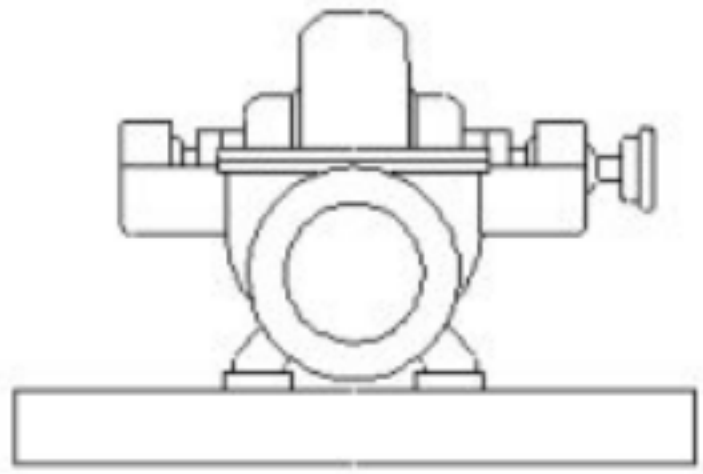


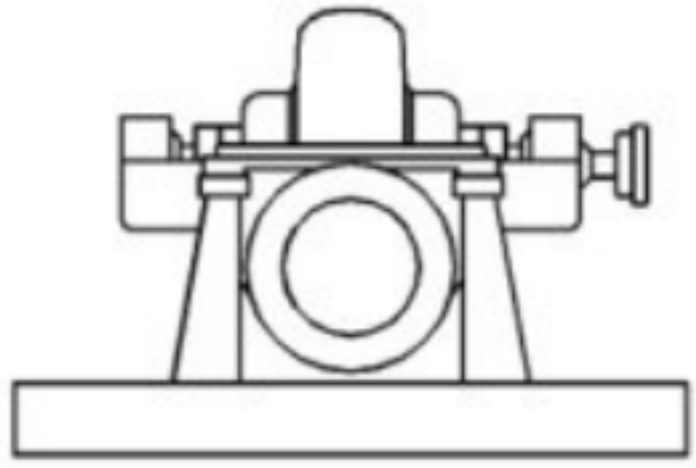
Figure 6—Pump Type OH6

4.2.2.8 Pump Type BB1

Axially split, one- and two-stage, between-bearings pumps shall be designated pump type BB1 (Figure 7).



a) Pump Type BB1-A "Foot Mounted"



b) Pump Type BB1-B "Near-centerline Mounted"

Figure 7—Pump Types BB1-A and BB1-B

4.2.2.9 Pump Type BB2

Radially split, one- and two-stage, between-bearings pumps shall be designated pump type BB2 (Figure 8). These pumps are typically centerline mounted.

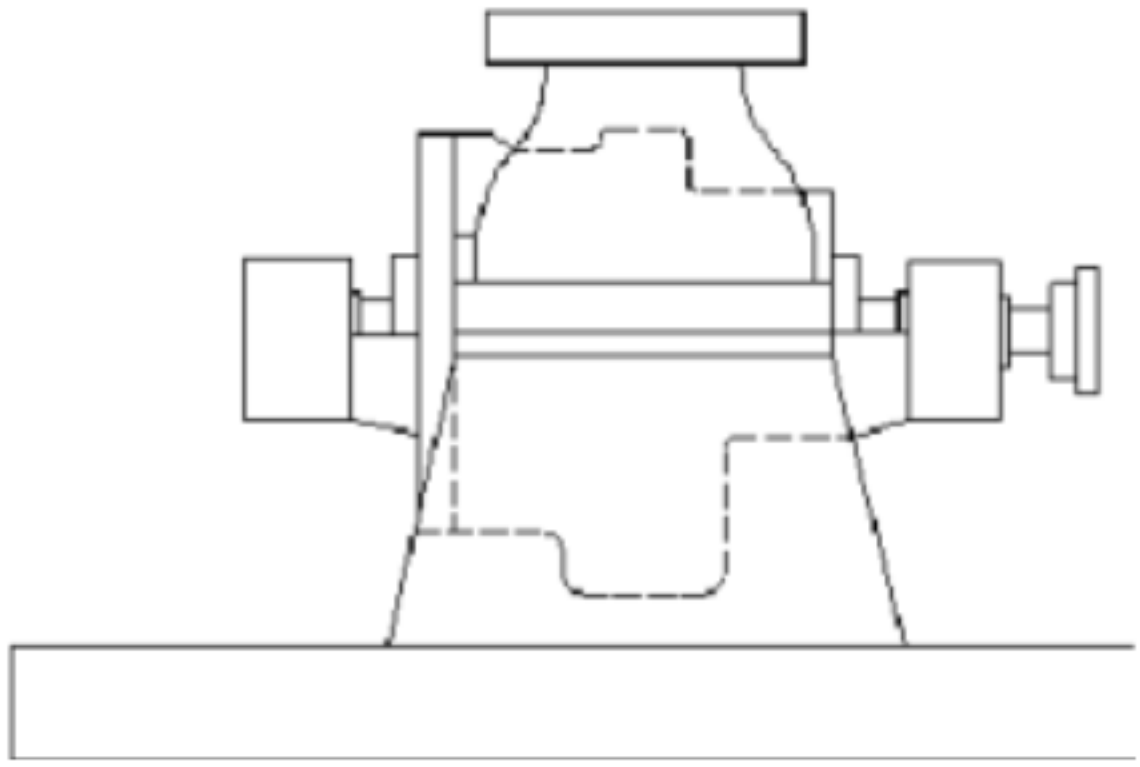


Figure 8—Pump Type BB2

4.2.2.10 Pump Type BB3

Axially split, multistage, between-bearings pumps shall be designated pump type BB3 (Figure 9). These pumps are near-centerline mounted.

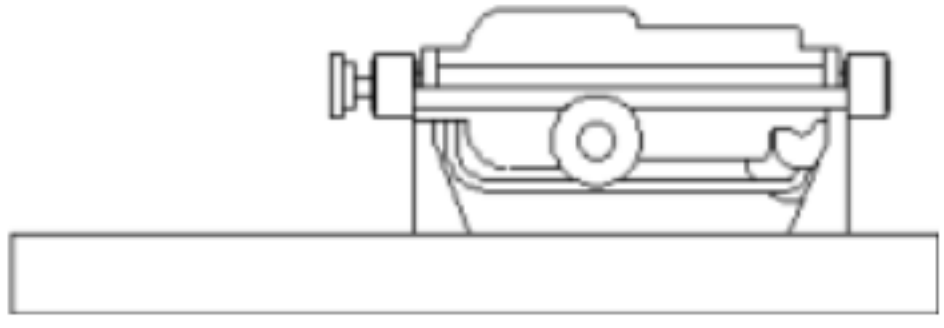


Figure 9—Pump Type BB3

4.2.2.11 Pump Type BB4

Single-casing, radially split, multistage, between-bearings pumps shall be designated pump type BB4 (Figure 10). These pumps are also called ring-section pumps, segmental-ring pumps, or tie-rod pumps. These pumps have a potential leakage path between each segment. These pumps are typically foot-mounted.

NOTE This type does not meet all the requirements of this standard; refer to Table 4.

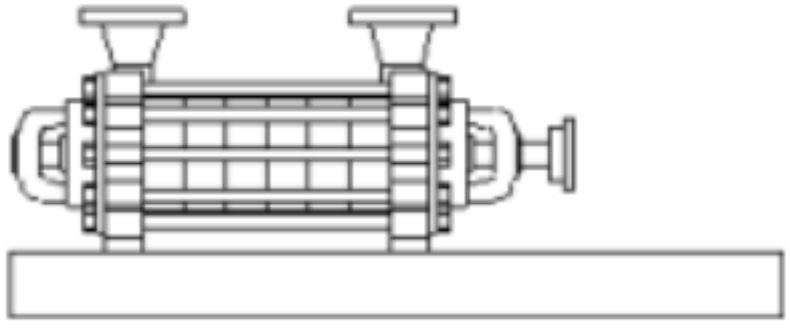


Figure 10—Pump Type BB4

4.2.2.12 Pump Type BB5

Double-casing, radially split, multistage, between-bearings pumps (barrel pumps) shall be designated pump type BB5 (Figure 11). These pumps are typically centerline mounted.

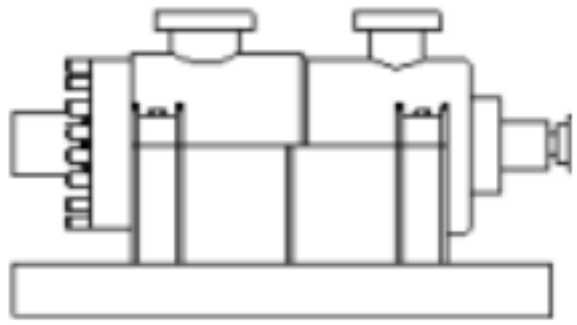


Figure 11—Pump Type BB5

4.2.2.13 Pump Type VS1

Wet pit, vertically suspended, single-casing diffuser pumps with discharge through the column shall be designated pump type VS1 (Figure 12).



Figure 12—Pump Type VS1

4.2.2.14 Pump Type VS2

Wet pit, vertically suspended, single-casing double-volute pumps with discharge through the column shall be designated pump type VS2 (Figure 13).

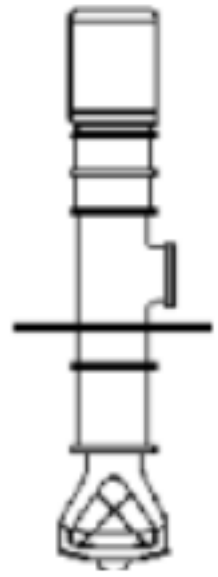


Figure 13—Pump Type VS2

4.2.2.15 Pump Type VS3

Wet pit, vertically suspended, single-casing axial-flow pumps with discharge through the column shall be designated pump type VS3 (Figure 14).



Figure 14—Pump Type VS3

4.2.2.16 Pump Type VS4

Vertically suspended, single-casing, volute, line-shaft-driven sump pumps shall be designated pump type VS4 (Figure 15).



Figure 15—Pump Type VS4

4.2.2.17 Pump Type VS5

Vertically suspended, cantilever sump pumps shall be designated pump type VS5 (Figure 16).



Figure 16—Pump Type VS5

4.2.2.18 Pump Type VS6

Double-casing, diffuser, vertically suspended pumps shall be designated pump type VS6 (Figure 17).

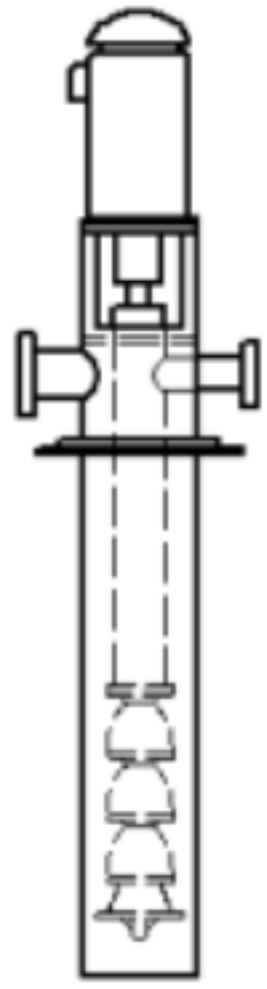


Figure 17—Pump Type VS6

4.2.2.19 Pump Type VS7

Double-casing, double-volute or double-volute plus diffuser, vertically suspended pumps shall be designated pump type VS7 (Figure 18).

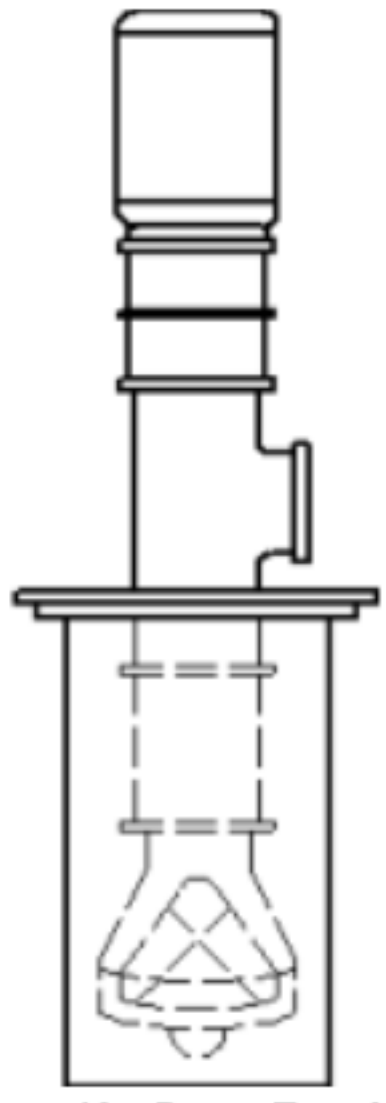


Figure 18—Pump Type VS7

5 Requirements

5.1 Units

[•] The purchaser shall specify whether data, drawings, and maintenance dimensions of pumps shall be in the U.S. customary (USC) or SI system of measurements. Use of a USC data sheet (see Figure N.1) indicates the USC system of measurements shall be used. Use of an SI data sheet (see Figure N.2) indicates the SI system of measurements shall be used.

5.2 Statutory Requirements

The purchaser and the vendor shall determine the measures to be taken to comply with any governmental codes, regulations, ordinances, or rules that are applicable to the equipment, its packaging and preservation. Equipment installed in the European Economic Area shall comply with all applicable European Union Directives.

5.3 Hierarchy of Requirements

[•] **5.3.1** The hierarchy of documents shall be specified.

5.3.2 Where requirements specific to a particular pump type in Section 9 conflict with any other section, the requirements of Section 9 shall govern.

6 Basic Design

6.1 General

6.1.1 Equipment Reliability

[•] **6.1.1.1** Only equipment that is field proven, as defined by the purchaser, is acceptable.

NOTE Purchasers can use their engineering judgment in determining what equipment is field proven. API 691 can provide guidance.

[•] **6.1.1.2** If specified, the vendor shall provide the documentation to demonstrate that all equipment and auxiliaries proposed qualify as field proven.

6.1.1.3 In the event no such equipment is available, the vendor shall submit an explanation of how their proposed equipment can be considered field proven.

NOTE A possible method of qualifying can be that all components comprising the assembled machine could satisfy the field proven definition.

6.1.2 The vendor shall advise in the proposal any component designed for a finite life.

[•] **6.1.3** The purchaser shall specify the operating conditions (including any upset, cleaning, seasonal, etc.), the liquid properties, preferred materials, site conditions, and utility conditions.

[•] **6.1.4** The purchaser shall specify if equipment should be supplied in accordance with API 691.

[•] **6.1.5** If API 691 is specified, the vendor shall identify all machinery components that are not “Field Qualified” per API 691 (First Edition), Section 4.3.2 and Table 1.

[•] **6.1.6** If the application is an HPRT, the purchaser shall specify the maximum inlet pressure. HPRTs shall comply with Annex C.

6.1.7 The equipment shall be capable of operation at the normal and rated operating points and any other operating conditions specified by the purchaser.

6.1.8 Pumps shall be capable of at least a 5 % head increase at rated flow by replacement of the impeller(s) with one(s) of larger diameter or different hydraulic design, variable-speed capability, or use of a blank stage.

NOTE The 5 % head increase at rated flow requirement is intended to prevent a change in selection caused by refinement of hydraulic requirements after the pump has been purchased.

6.1.9 Pumps shall be capable of operating at least up to the maximum continuous speed. The maximum continuous speed shall be:

a) equal to the speed corresponding to the synchronous speed at maximum supply frequency for fixed speed electrical motors,

b) at least 105 % of rated speed for variable-speed pumps and any fixed-speed pump sparing or spared by a pump whose driver is capable of exceeding rated speed.

6.1.10 Variable-speed pumps shall be designed to reach trip speed without damage.

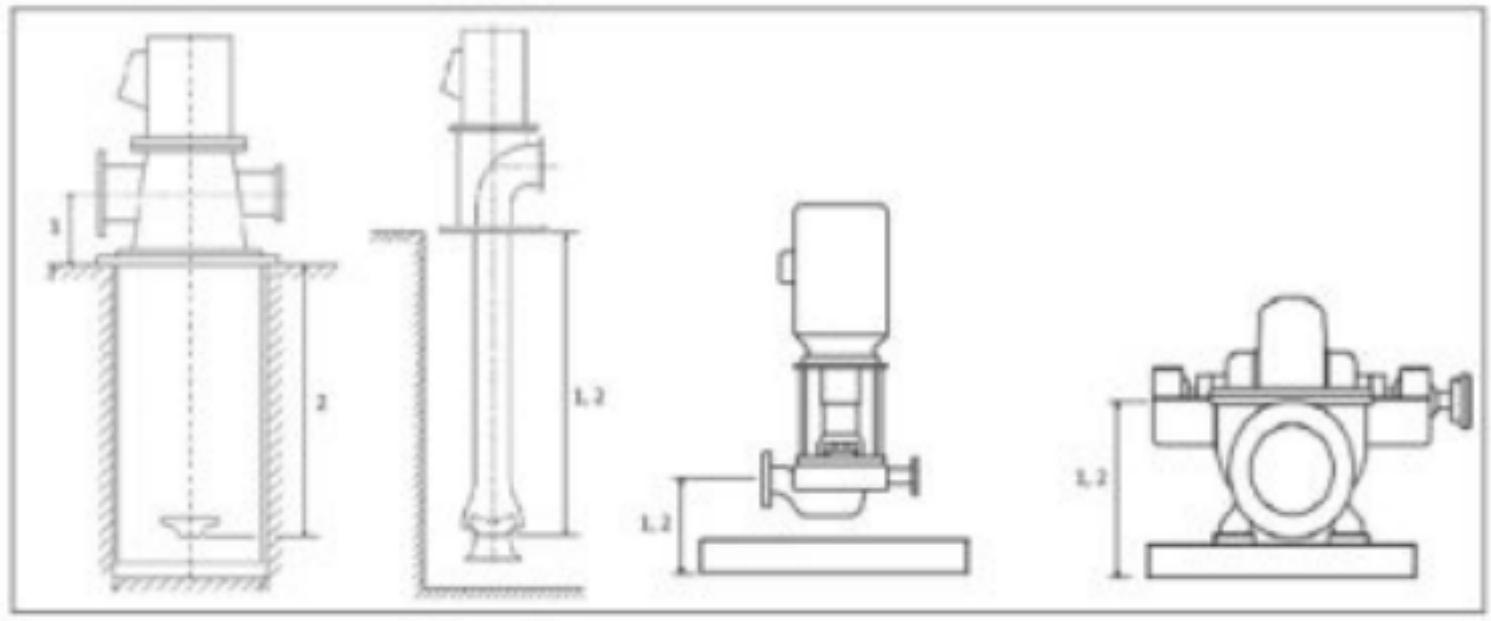
6.1.11 The seal drawing shall contain detailed data to show that purchaser's requirements for temperature, pressure, and flow, as well as vacuum service (when specified) are met by the proposed seal design. The pump vendor shall submit the seal design drawing to the purchaser for approval.

6.1.12 The vendor shall specify the net positive suction head at 3 % head loss (NPSH3) based on water at a temperature less than 130 °F (55 °C) at the rated flow and rated speed. A reduction or correction factor for liquids other than water (such as hydrocarbons) shall not be applied. NPSH3 is a performance characteristic of the first-stage impeller.

6.1.13 The datum elevations for NPSH3 shall be the shaft centerline for horizontal pumps and the impeller suction eye elevation for vertical in-line pumps and vertically suspended pumps. Refer to Figure 19.

NOTE The datum elevations defined above for NPSH3 are in accordance with HI 14.6 (ISO 9906). NPSH3 values are provided by the vendor and can be demonstrated by test.

[•] **6.1.14** The purchaser shall specify the NPSHA based on system conditions at the rated flow. The datum elevations for NPSHA shall be the estimated shaft centerline for horizontal pumps and the estimated impeller centerline elevation for vertical pumps (VS1, VS2, VS3, VS4, VS5) and estimated suction nozzle centerline for vertical pumps (VS6 and VS7). (Refer to Figure 19.)



Key
1 NPSHA Datum
2 NPSH3 Datum

Figure 19—NPSH Datum Levels

[•] **6.1.15** The minimum NPSH margin at the rated point shall be 3 ft (1 m). The purchaser shall specify if a different NPSH margin is required. Refer to HI 9.6.1 for guidance on determining NPSH margins.

[•] **6.1.16** The pump specific speed and suction-specific speed shall be calculated in accordance with Annex A. If the purchaser specifies a suction-specific speed limit, the pump shall have a calculated suction-specific speed not greater than that specified limit.

6.1.17 Pumps that handle liquids more viscous than water shall have their water performance corrected in accordance with HI 9.6.7.

6.1.17.1 Correction factors used for a viscous liquid shall be submitted with both proposal curves and final test curves.

6.1.17.2 Curves shall show the performance corrected for viscosity.

[•] **6.1.18** If parallel operation is specified, the head values of the pumps at any given flow within the preferred operating region shall be within 3 % of each other for pumps larger than 3 in. (80 mm) discharge.

[•] **6.1.19** If a system design pressure is specified by the purchaser, the pump maximum discharge pressure shall not exceed the stated system design pressure.

6.1.20 If a discharge orifice or device is used as a means of providing a continuously rising head curve, this shall be stated in the proposal, and requires purchaser's approval.

6.1.21 Pump head curves shall be continuously rising from rated point to shutoff with minimum rise of 10 %.

NOTE Pumps with a continuously rising head curve are preferred for all applications, but this is not possible with all pump types.

6.1.22 Pumps shall have a preferred operating region of 70 % to 120 % of BEP flowrate at rated impeller trim.

6.1.23 Pump rated flow shall be within the region of 80 % to 110 % of the BEP flowrate at rated impeller trim.

6.1.24 For pumps equipped with ASDs, the vendor shall determine the minimum and maximum allowable speed of the equipment and record the value.

6.1.25 The vendor shall provide maximum expected sound pressure level (SPL) for the equipment.

6.1.25.1 Control of the SPL of all equipment furnished shall be a joint effort of the purchaser and the vendor who has unit responsibility.

[•] **6.1.25.2** If specified, the equipment furnished by the vendor shall conform to a maximum allowable SPL specified by the purchaser.

6.1.26 High energy pumps can require special provisions to reduce vane passing-frequency vibration and low-frequency vibration at reduced flowrates. For these pumps, the radial clearance between the diffuser vane or volute tongue (cutwater) and the periphery of the impeller vane shall be at least 3 % of the maximum impeller vane-tip radius for diffuser designs and at least 6 % of the maximum blade-tip radius for volute designs.

NOTE In single-stage Between Bearing pumps with double suction impellers, a center rib and staggered vane design can mitigate some vane passing and acoustic vibrations. “Clocking” impeller vanes in multistage pumps can also be applied. Other methods can also be used.

6.1.27 Pumps operating above 3600 r/min and absorbing more than 400 hp (300 kW) per stage can require even larger running clearances and other special construction features. For these pumps, specific requirements shall be agreed upon by the purchaser and the vendor, considering actual operating experience with the specific pump types.

6.1.28 The need for bearing housing cooling shall be determined by the vendor, and the method shall be agreed upon by the purchaser.

6.1.28.1 Fan cooling for the bearing housing shall be the first choice.

6.1.28.2 If fan cooling is inadequate, one of the plans in Annex B shall be selected.

[•] **6.1.28.3** The cooling system shall be suitable for operation with the coolant type, pressure and temperature specified by the purchaser. The vendor shall specify the required flow.

[•] **6.1.28.4** If specified, pumps with fan cooling, wet sump lubrication and pumping temperatures over 500 °F (260 °C) shall have provision for one of the following:

a) external oil circulation system;

b) bearing housing cooling coil;

c) purge oil mist

6.1.28.5 The bearing housing connections for external oil circulation systems or bearing housing cooling coils shall be plugged unless cooling is required.

6.1.28.6 When the provision for connecting an external oil circulation system is specified, the oil return connection shall be aligned with the normal sump oil level to maintain proper oil level in the sump.

6.1.29 Jacket systems, if provided, shall have clean-out connections arranged so that the entire passageway can be mechanically cleaned, flushed, and drained.

6.1.30 Jacket systems, if provided, shall be designed to prevent the process stream from leaking into the jacket or the heat transfer medium from leaking into the process. Jacket passages shall not open into casing joints.

6.1.31 Water-cooling systems shall be designed for the conditions on the water side as given in Table 2.

6.1.32 Provisions shall be made for complete venting and draining of cooling-water systems.

6.1.33 The vendor shall notify the purchaser if the criteria for maximum temperature rise and velocity over heat exchange surfaces result in a conflict. The criterion for velocity over heat exchange surfaces is intended to

minimize water-side fouling, and the criterion for maximum temperature rise is intended to minimize the use of cooling water. If such a conflict exists, the purchaser shall approve the final selection.

Table 2—Water-cooling Systems—Conditions on the Water Side

Parameter	USC and Other Units	SI Units
Velocity over heat exchange surfaces	5 ft/s to 8 ft/s	1.5 m/s to 2.5 m/s
Maximum allowable working pressure (MAWP), gauge shall be as a minimum	100 psi; 7 bar	700 kPa
Test pressure (> 1.5 MAWP), gauge	150 psi; 10.5 bar	1050 kPa
Maximum pressure drop	15 psi, 1 bar	100 kPa
Maximum inlet temperature	90 °F	30 °C
Maximum outlet temperature	120 °F	50 °C
Maximum temperature rise	30 °F	20 °C
Fouling factor on water side	0.002 h-ft^2-°R/Btu	0.35 m^2 K/kW
Shell corrosion allowance (not for tubes)	0.125 in.	3.0 mm

6.1.34 The arrangement of the equipment, including piping and auxiliaries, shall be developed jointly by the purchaser and the vendor. The arrangement shall provide clearance between equipment and piping and auxiliaries to allow safe access for operation and maintenance.

6.1.35 The vendor drawing shall include detail drawings of all piping that will be supplied on the skid, for purchaser review.

6.1.35.1 Auxiliary pipe routing should be in accordance with API 686.

6.1.36 Electrical Classification

6.1.36.1 If an installation location is classified as hazardous, motors, electrical instrumentation, equipment, components, and electrical installations shall be suitable for the hazardous electrical area classification designation as specified. Refer to API 500 for additional information.

[•] **6.1.36.2** All applicable electrical codes shall be specified. Local electrical codes that apply shall be provided by the purchaser upon request.

6.1.37 Oil reservoirs and housings that enclose moving lubricated parts, such as bearings, shaft seals, highly polished parts, instruments and control elements, shall be designed to minimize contamination by moisture, dust, and other foreign matter while in standby or in operation.

6.1.38 Major parts, such as casing components and bearing housings, shall be designed and manufactured to ensure accurate alignment on reassembly. This can be accomplished by the use of shouldering, dowels, or keys.

6.1.39 Non-rotating parts weighing more than 50 lbs (23 kg) shall include provisions for the attachment of lifting accessories for safe mechanical handling.

6.1.40 Except for vertically suspended pumps and integrally geared pumps, pumps shall be designed to permit removal of the rotor or inner bundle without disconnecting the suction or discharge piping or moving the driver.

6.1.41 Spare and all replacement parts for the pump and all furnished auxiliaries shall, as a minimum, meet all the criteria of this standard.

6.1.42 Equipment, including all auxiliaries, shall be designed for the specified site environmental conditions. The vendor shall advise any equipment protection required for the jobsite location (e.g. winterization for low ambient temperatures, or protection against unusual humidity, dusty or corrosive conditions).

[•] **6.1.43** If specified for offshore floating applications, the proposed design shall conform to the required vessel motion design criteria.

6.1.44 Bolting and Threads

6.1.44.1 The details of threading shall conform to ASME B1.1, ASME B1.13M, or ISO 261. The vendor shall advise the type of bolting used on the pump.

6.1.44.2 If ASME B1.1 threads have been specified, the thread series shall be the varying-pitch series UNC for sizes up to and including 1 in. (25 mm) and 8 UN for sizes greater than 1 in. (25mm). All threads shall be class 2 or 3.

6.1.44.3 If ISO 261 and ISO 262 have been specified, the thread series shall be coarse. Threads shall be class 6g for bolting and studs and class 6H for nuts.

6.1.44.4 Commercial fasteners shall be manufactured in accordance with the requirements of ASME B18.18.2M or shall be procured from distributors having quality plans in accordance with ASME B18.18.2M.

6.1.44.5 Manufacturers shall specify the bolt torque and the lubricant nut factor used to calculate required tensile load.

6.1.45 Adequate clearance shall be provided at all bolting locations to permit the use of standard tools.

6.1.46 Studs shall be supplied on all main casing joints, and all other joints and connections shall be supplied with external hexagon-head bolting.

6.1.47 Fasteners (excluding washers and headless set-screws) shall have the material grade and manufacturer's identification symbols applied to one end of studs 0.38 in. (10 mm) in diameter and larger and to the heads of bolts 0.25 in. (6 mm) in diameter and larger.

6.1.47.1 If the available area is inadequate, the grade symbol may be marked on one end and the manufacturer's identification symbol marked on the other end.

6.1.47.2 Studs shall be marked on the exposed end.

6.1.48 Pressure casing fasteners shall be not less than 0.5 in. (12 mm) diameter.

6.2 Pressure Casings

6.2.1 The maximum discharge pressure shall be the highest of the maximum discharge pressures as calculated for all of the operating cases.

6.2.1.1 The calculations shall be based on the maximum differential pressure that the pump is able to develop when operating with the furnished impeller(s) when pumping the relative density (specific gravity) from respective operating cases, plus the maximum suction pressure for respective operating cases.

6.2.1.2 The pump vendor shall calculate the maximum discharge pressure for the pump and shall provide the value to the purchaser.

6.2.1.3 For variable speed pumps, the maximum discharge pressure shall be calculated by adding the additional differential pressure developed during operation to trip speed or maximum continuous speed of the driver.

[•] **6.2.2** If specified, the maximum discharge pressure shall be increased by the additional differential pressure developed during operation with temporary liquid of higher relative density as specified in the liquid characteristics section of the pump data sheet (e.g. post-construction pipe cleaning liquid);

[•] **6.2.3** If specified, the maximum discharge pressure shall be increased by the additional differential pressure developed during operation with installation of an impeller of the maximum diameter, maximum number of stages or both that the pump can accommodate.

6.2.4 The pressure casing shall be designed to:

a) operate without leakage or internal contact between rotating and stationary components while subject simultaneously to the MAWP (and maximum operating temperature) and the worst-case combination of twice the allowable nozzle loads of Table 5 applied through each nozzle;

b) withstand the hydrostatic test (see 8.3.2.3).

NOTE The twice-nozzle-load requirement is a pressure casing design criterion. Allowable nozzle loads for piping designers are the values given in Table 5, which, in addition to pressure casing design, include other factors that affect allowable nozzle loads, such as casing support and baseplate stiffness.

6.2.5 For pressure casing components, the stresses used in the design shall not exceed the allowable stress according to ASME BPVC Section IID.

6.2.5.1 If design by rule is used (ASME BPVC Section VIII Division 1), then the allowable stress in ASME BPVC Section IID Table 1 and Table 3 shall be used.

6.2.5.2 If design by analysis is used (ASME BPVC Section VIII Division 2), then ASME Tables 2, 3, 4 and 5 shall be used across the full range of specified operating temperatures.

6.2.5.3 For castings, the allowable stress shall be multiplied by the appropriate casting factor, as shown in Table 3.

NOTE Per ASME BPVC Section VIII, Division 1, Section U-1(c), pump casings are not pressure vessels.

Table 3—Casting Factors

Type of Nondestructive Examination	Casting Factor
Visual, magnetic particle, liquid penetrant	0.8
Spot radiography	0.9
Ultrasonic	0.9
Full radiography	1.0

6.2.6 The vendor's proposal shall state the source of the material properties from those listed in Table H.2 (i.e. ASTM, ISO, UNS, EN, JIS), as well as the casting factors applied. National material standards other than those listed in Table H.2 may be used with purchaser's approval.

6.2.7 The casing joint bolting shall be in accordance with 6.1.44.

6.2.7.1 For casing joint bolting, the allowable stress (as determined in 6.2.5) shall be used to determine the total bolting area based on hydrostatic load and gasket preload as applicable.

6.2.7.2 The preload stress shall not exceed 80 % of the bolting material minimum yield strength.

6.2.7.3 During the hydrostatic test, the bolting stress shall not exceed 90 % of the bolting material minimum yield strength.

NOTE Thread stress in the nut or casing can be the limiting factor in the strength of the bolting.

6.2.8 Except as noted in 6.2.11, the MAWP shall be at least the maximum discharge pressure (see 6.2.1 through 6.2.3) plus 10 % of the maximum differential pressure and shall not be less than:

a) for pump types BB1, VS1, VS2, VS3, VS4, VS5, VS6, and VS7: a pressure rating equal to that of an ASME B16.5, class 150 steel flange of material grade corresponding to that of the pressure casing;

NOTE For the purpose of this provision, ISO 7005–1 PN20 and EN 1759–1, class 150 are equivalent to ASME B16.5, class 150.

b) for all other pump types: a gauge pressure rating equal to at least 600 psi (4 MPa; 40 bar) at 100 °F (38 °C).

6.2.9 The pump-seal chamber and seal gland shall have a pressure-temperature rating at least equal to the MAWP and temperature of the pump casing to which it is attached, in accordance with API 682.

6.2.10 The entire pump pressure casing shall be designed for the MAWP.

[•] **6.2.11** If specified, pump pressure casings for pump types VS6, VS7, OH6, BB3, BB4, and BB5 may be designed for dual pressure ratings. The purchaser should consider installation of relief valves on the suction side of such installations.

6.2.12 The MAWP of pressure casings for HPRTs shall be at least equal to the maximum inlet pressure plus 10 %.

[•] **6.2.13** If a relief valve is provided by the purchaser on the downstream piping before the first block valve, the MAWP of the low-pressure region of the casing shall be at least the specified relief valve set point plus 10 %.

6.2.14 The minimum corrosion allowance for the pressure casing shall be 0.12 in. (3 mm).

NOTE The vendor can propose alternative corrosion allowances for consideration if materials of construction with superior corrosion resistance are employed without affecting safety and reliability.

6.2.15 The inner casing of double-casing pumps shall be designed to withstand the maximum differential pressure or 50 psi (350 kPa, 3.5 bar), whichever is greater.

6.2.16 Unless otherwise specified, pumps with radially split casings shall be supplied in services for any of the following operating conditions:

a) pumping temperature of 400 °F (200 °C) or higher;

b) liquids with a relative density of less than 0.7 at the specified pumping temperature;

c) liquids at a rated discharge gauge pressure above 1450 psi (10 MPa, 100 bar);

d) liquid temperature transients greater than 100 °F (55 °C) in less than 10 seconds.

NOTE Axially split casings have been used successfully beyond the limits given above, particularly for pipeline applications at higher pressure and with lower relative density (specific gravity). Transient thermal changes can have negative effects on the life of a pump if they are not accounted for in design, including warm-up or cool-down requirements.

6.2.17 Radially split casings shall have metal-to-metal fits, with confined controlled-compression gaskets, such as an O-ring or a spiral-wound type. Gaskets other than spiral-wound may be proposed and furnished if proven suitable for service and approved by the purchaser (see 9.3.2.3 for VS type pumps).

6.2.18 Centerline-mounted pump casings shall be used for all horizontal pumps except as allowed in 9.2.1.2.

6.2.18.1 OH2 pumps with maximum operating temperature above 500 °F (260 °C) shall have a radial guide, key slot or equivalent design feature between the pump and the baseplate pedestal at each support location at the center of the pressure casing.

6.2.18.2 BB3 pumps with maximum operating temperature above 300 °F (150 °C), and BB2 and BB5 pumps operating above 500 °F (260 °C) shall include the following:

a) guide, key slot or equivalent design feature between the pump and the baseplate pedestal at the non-drive end support locations;

b) drive end mounting feet pinned, doweled or transverse keyed to the support pedestals.

NOTE The purpose of 6.2.18.1 and 6.2.18.2 is to ensure that the casing is allowed to thermally expand in the direction away from the driver in a straight line.

6.2.19 O-ring sealing surfaces, including all grooves and bores, shall have a maximum surface roughness average value, Ra, of 63 µin. (1.6 µm) for static O-rings and 32 µin. (0.8 µm) for the surface against which dynamic O-rings slide.

6.2.19.1 Bores shall have a minimum 0.12 in. (3 mm) radius or a minimum 0.06 in. (1.5 mm) chamfered lead-in for static O-rings and a minimum 0.08 in. (2 mm) chamfered lead-in for dynamic O-rings.

6.2.19.2 Chamfers shall have a maximum angle of 30°.

6.2.20 Spiral-wound gasket sealing surfaces shall have a maximum surface roughness average value, Ra, of 250 µin (6.4 µm) and minimum surface roughness average value of 125 µin (3.2 µm).

6.2.21 The use of threaded holes in pressure parts shall be minimized. To prevent leakage in pressure sections of casings, metal, equal in thickness to at least half the nominal bolt or stud diameter, plus the allowance for corrosion, shall be left around and below the bottom of drilled and threaded holes.

6.2.22 Internal bolting shall be of a material fully resistant to corrosive attack by the pumped liquid.

6.2.23 If the manufacture of cast pressure casing parts requires the use of openings for core support, core removal, or waterway inspection and cleaning, these openings shall be designed so they can be closed by welding, using an appropriate, qualified weld procedure, during the completion of casting manufacture. These openings shall be identified, per 6.12.2.11.4.

[•] **6.2.24** If specified, for BB3 or BB5 pumps, the main casing joint studs and nuts shall be designed for the use of hydraulic bolt tensioning. Procedures and the extent of special tooling provided by the vendor shall be agreed.

6.2.25 Jackscrews shall be provided to facilitate disassembly of the casing. One of the contacting faces shall be relieved (counter-bored or recessed) to prevent a leaking joint or an improper fit caused by marring of the face.

6.2.26 Mounting surfaces on the equipment shall meet the following criteria (Refer to Figure 20):

a) Mounting surfaces shall be machined to a finish of 250 µin (6 µm) Ra or better.

b) Each mounting surface shall be machined within a flatness of 0.001 in. per linear foot (75 µm per linear meter) of mounting surface.

c) To prevent a soft foot, the difference between all mounting surfaces in the same horizontal plane shall be within 0.005 in. (125 µm).

d) Mounting surfaces with differing design elevations shall be parallel to each other within 0.005 in. (125 µm).

e) The upper machined or spot faced surface shall be parallel to the mounting surface.

f) Hold-down bolt holes shall be drilled perpendicular to the mounting surface or surfaces.

g) Hold-down bolt holes shall be a minimum of 0.25 in. (6 mm) larger diameter than the bolt diameter, to prevent a bolt-bound condition.

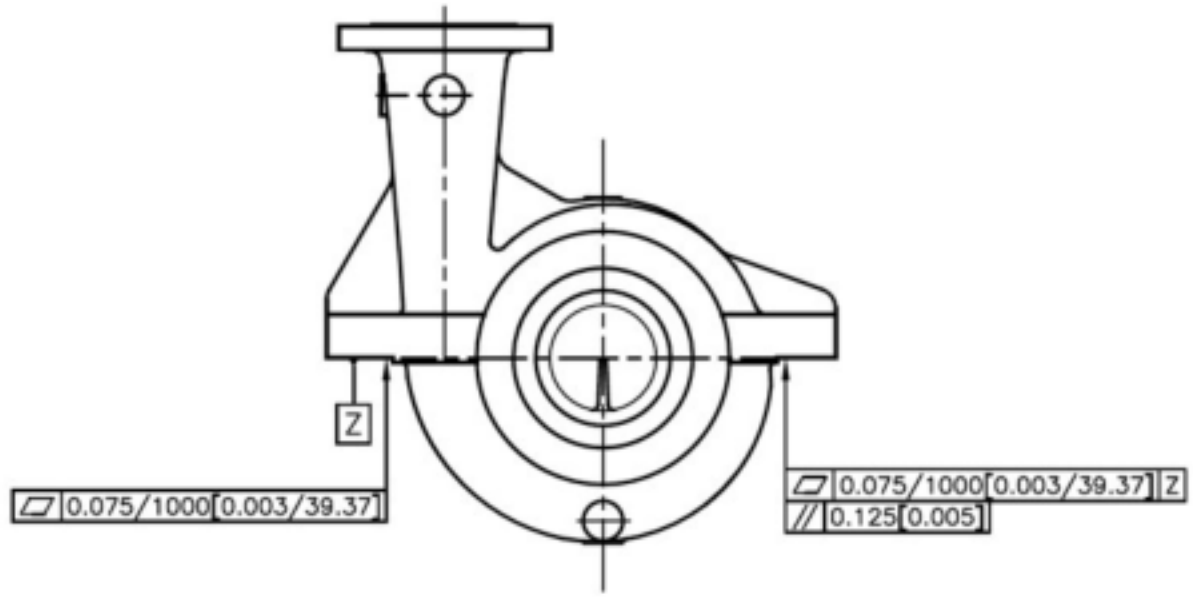


Figure 20—Pump Mounting Surface Tolerances, in./in. (mm/mm)

6.3 Pump Types

[•] **6.3.1** The pump types listed in Table 4 have special design features that prevent them from fully complying with this standard. These pumps may be offered only if specified by the purchaser and if the manufacturer has proven experience for the specific application. Table 4 lists the features requiring special consideration for these pump types and gives in parentheses the relevant subsection(s) of this standard.

Table 4—Special Design Features of Particular Pump Types

Pump Type	Features Requiring Special Consideration
Horizontal foot-mounted overhung, OH1	a) Pressure rating (6.2.8) b) Casing support (6.2.18)
Rigidly coupled vertical in-line, OH4	a) Motor construction (7.1.12, 7.1.13) b) Rotor stiffness (6.6.16) c) Product-lubricated guide bearing (6.10.1.1) d) Shaft runout at seal (6.6.9, 6.8.5)
Close-coupled (impeller mounted on motor shaft), OH5	a) Motor construction (7.1.12, 7.1.13) b) Motor bearing and winding temperature at high pumping temperatures c) Seal removal (6.8.2)
Two-stage overhung	a) Rotor stiffness (6.6.16)
Double-suction overhung	a) Rotor stiffness (6.6.16)
Ring-section casing (multistage), BB4	a) Pressure containment (6.2.4, 6.2.16, 6.2.15) b) Dismantling (6.1.40)
Built-in mechanical seal (no separable seal gland)	a) Seal removal (6.8.2)

[•] **6.3.2** If the pump is specified by the purchaser to be a special-purpose pump, refer to Annex D for guidance.

6.4 Nozzles and Pressure Casing Connections

6.4.1 Casing Opening Sizes

6.4.1.1 Openings for nozzles and other pressure casing connections shall be standard pipe sizes. Openings of NPS 1–1/4, NPS 2–1/2, NPS 3–1/2, NPS 5, NPS 7, and NPS 9 (DN 32, DN 65, DN 90, DN 125, DN 175, and DN 225) shall not be used.

6.4.1.2 Casing connections other than suction and discharge nozzles shall be at least NPS 1/2 (DN 15) for pumps with discharge nozzle openings NPS 2 (DN 50) and smaller.

6.4.1.3 Casing connections shall be at least NPS 3/4 (DN 20) for pumps with discharge nozzle openings NPS 3 (DN 80) and larger.

6.4.1.4 Casing connections for seal flush piping and gauges shall be minimum NPS 1/2 (DN 15) regardless of pump size.

6.4.1.5 Drain connections of pumps with discharge nozzles of NPS 4 (DN 100) or larger, handling fluids with a viscosity greater than or equal to 400 cP, products with higher than ambient pour point temperature, or slurries shall be at least NPS 1 (DN25).

6.4.1.6 The first segment of piping connected to the casing shall be less than 6 in. (150 mm) long and shall be a minimum of schedule 160 seamless for pipe sizes NPS 1 (DN 25) and smaller and a minimum of schedule 80 for pipe sizes NPS 1½ (DN 40) and larger.

6.4.1.7 The first segment of the piping (nipple) welded to the casing shall be straight to allow connections such as drains to be cleaned.

6.4.1.7.1 The first segment may be attached axially to avoid increasing centerline height.

6.4.1.7.2 On small pumps, if this requirement causes interference with the suction nozzle, the drain nozzle orientation may be changed with purchaser approval.

6.4.2 Casing Nozzle Connections

6.4.2.1 All casing connection nozzles shall be flanged, except for the seal gland connection, or those on pumps with forged casings, which shall be flanged or machined and studded.

6.4.2.2 If the pump is supplied with machined and studded connections, the pump vendor shall provide drawings showing the dimensions of break-out spool pieces to allow the pump to be removed from the piping.

6.4.2.3 Break-out spool pieces shall be provided by the purchaser.

6.4.2.4 All pumps shall have suction and discharge flanges of equal rating, unless casing is designed for dual pressure rating, per 6.2.11.

6.4.2.5 All nozzle flanges shall conform to the pressure ratings and dimensional requirements of ASME B16.5 and ASME B16.47 Series B, except as noted in 6.4.2.5.1 through 6.4.2.5.3, 6.4.2.6, and 6.4.2.7.

NOTE For the purpose of this standard, ISO 7005–1 and EN 1759.1 are equivalent to ASME B16.5 and ASME B16.47. Additional resources for flange quality are MSS SP 44 (for large flanges) and MSS SP 75 (for fittings).

6.4.2.5.1 For ASME B16.5 flange sizes from $^{1}/_{2}$ NPS (12 mm) to 10 NPS (250 mm), the actual outside diameters (ODs) of the flanges shall be not less than their nominal OD minus 0.16 in. (4 mm).

6.4.2.5.2 For ASME B16.5 flange sizes from 12 NPS (300 mm) to 24 NPS (600 mm), the actual ODs of the flanges shall be not less than their nominal OD minus 0.21 in. (5 mm).

NOTE These tolerances are consistent with the negative tolerances stated on EN 1092–1, Table 22.

6.4.2.5.3 For ASME B16.47 flanges, the actual ODs of the flanges shall be not less than their nominal OD minus 0.25 in. (6 mm).

6.4.2.6 Flanges in all materials that are thicker or have a larger OD than required by the relevant ISO or ASME standards in this standard are acceptable.

6.4.2.6.1 Nonstandard (oversized) flanges shall be completely dimensioned on the arrangement drawing.

6.4.2.7 If nonstandard flanges require studs or bolts of nonstandard length, this requirement shall be identified on the arrangement drawing along with required torque value and lubricant (k-) factor.

6.4.2.8 If nonstandard stud lengths are required for extra-thick flanges, the vendor shall provide studs of the appropriate length.

6.4.2.9 Flanges shall be full faced or spot faced on the back and shall be designed for through bolting, except for jacketed casings.

6.4.2.10 To minimize nozzle loading and facilitate installation of piping, machined faces of pump flanges shall be parallel to the plane of the flange as shown on the general arrangement drawing within 0.5°.

6.4.2.11 Bolt holes or studs shall straddle centerlines parallel to the main axes of the pump.

6.4.2.12 Connections, including gusseting, welded to the casing (up to the first flange) shall meet or exceed the material and the pressure-temperature requirements of the casing, including impact values, rather than the requirements of the connected piping.

6.4.2.12.1 For C6 construction, 316L pipe and fittings connected to the casing are required for process temperatures up to and including 500 °F (260 °C), above which UNS N06625 materials shall be used.

6.4.2.12.2 For C6 construction, gussets shall be a minimum of 316L.

6.4.2.13 All of the purchaser's connections shall be accessible for disassembly without requiring the pump, or any major part of the pump, to be moved.

6.4.3 Auxiliary Connections

[•] **6.4.3.1** Auxiliary connections shall be integrally flanged, socket welded, or butt welded as specified by the purchaser.

6.4.3.2 Threaded connections and / or seal welding of threaded connections on the casing are not permitted, except as addressed in 7.6.2.12 or 6.4.3.5.

6.4.3.3 Full penetration butt welds shall be used when any of the following apply:

a) The pump nozzle rating is ASME B16.5 class 900 and above;

b) The minimum pumping temperature is 32 °F (0 °C) and below;

c) Due to the nature of the pumped fluid, any of the following apply:

1) Pump specified to be in hot service (>400 °F (>200 °C));

2) Service classed as any of the following:

i) extremely hazardous;

ii) highly corrosive or highly erosive;

iii) compliant with either NACE MR0175 or NACE MR0103.

NOTE Extremely hazardous and highly corrosive or highly erosive services are defined by the purchaser and typically include those as described in API 570, class 1 or API 682, Annex A.

6.4.3.4 Table 14 shall govern the NDT required for butt welded connections.

[•] **6.4.3.5** If specified, for pumps in pipeline service with a maximum operating temperature of 130 °F (55 °C) or less, auxiliary connections may be threaded.

6.4.3.5.1 Threaded openings shall be tapered threads in accordance with ASME B1.20.1 or ISO 7–1 or alternatively cylindrical threads in accordance with ISO 228–1.

6.4.3.5.2 Cylindrical threads shall be sealed with a contained face gasket of suitable material sealing to a machined face (see Figure 21). The chosen method and details of threaded connection shall be approved by the purchaser.

6.4.3.5.3 Threaded connections shall be brought to a remote flange that can be disassembled for maintenance, to require the least amount of assembly/disassembly of the threaded connection.

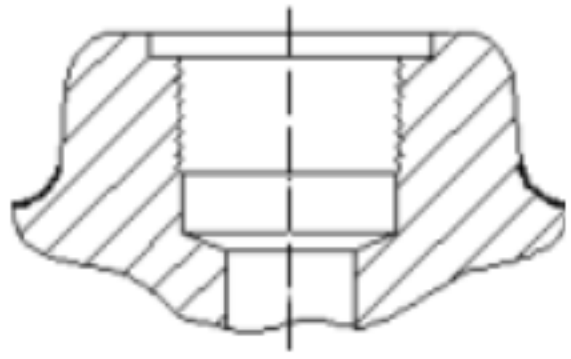


Figure 21—Machined Face Suitable for Gasket Containment if Using Cylindrical Threads

[•] **6.4.3.6** If specified, special threaded fittings for transitioning from the casing to tubing for seal flush piping shall be used, provided that a secondary sealing feature, such as O-rings, are used and that the joint does not depend on the thread contact alone to seal the fluid. The connection boss shall have a machined face suitable for sealing contact.

6.4.3.7 Threaded openings, which are allowed in seal glands and in pump casings for some pipeline services (if specified in 6.4.3.5), shall be plugged.

6.4.3.7.1 Plugs for tapered threads shall be long-shank solid round-head, or long-shank hexagon-head, bar stock plugs in accordance with ASME B16.11.

6.4.3.7.2 Plugs for cylindrical threads shall be solid hexagon-head plugs in accordance with DIN 910.

6.4.3.7.3 Plugs shall meet the material requirements of the casing, except as noted in 6.4.2.12.1.

6.4.3.7.4 A process compatible thread lubricant of proper temperature specification shall be used on all threaded connections to ensure that the threads are vapor-tight.

6.4.3.7.5 Thread tape shall not be used.

6.4.3.7.6 Plastic plugs are not permitted.

[•] **6.4.3.8** If specified, auxiliary connections to the pressure casing shall be machined and studded. These connections shall conform to the facing and drilling requirements of 6.4.2.5.

6.4.3.8.1 Studs and nuts shall be furnished installed.

6.4.3.8.2 If the stud engagement does not control the depth of the threads in the mating hole, the first 1.5 threads at both ends of the stud shall be removed.

6.4.3.9 All pumps shall be provided with vent connections, except that vent connections may be omitted if the pump is made self-venting by the arrangement of the nozzles. A pump is considered functionally self-venting if

the nozzle arrangement and the casing configuration permit venting of gases from the first-stage impeller and volute area to prevent loss of prime during the starting sequence.

NOTE Pumps with a top-discharge, such as OH2 pumps, are generally considered self-venting.

6.4.3.10 All pumps shall be provided with drain connections.

6.4.3.10.1 If the pump cannot be completely drained for geometrical reasons, this shall be stated in the proposal.

6.4.3.10.2 The operating manual shall include a drawing indicating the quantity and location(s) of the liquid remaining in the pump.

6.4.3.11 For OH2 and BB2 pumps, with double volute construction or with a casing orientation resulting in an internal liquid trap, a drain hole shall be provided at the low point of the inner volute.

6.4.3.12 For socket-welded construction, there shall be a $^{1}/_{16}$ in. (1.5 mm) gap between the pipe end and bottom of the socket before welding.

6.4.3.13 Piping NPS 2 (DN 50) and smaller shall be gusseted in two orthogonal planes to increase the rigidity of the piped connection, in accordance with the following stipulations, except connections for seal flush piping and gauges, or integral flanges. Two possible arrangements for the gusseting of piping are depicted in Figure 22.

a) Gussets shall be of a material compatible with the pressure casing and the piping and shall be made of either flat bar with a minimum equivalent cross section of 1 in. by 0.12 in. (25 mm by 3 mm) or round bar with a minimum diameter of 0.38 in. (9 mm).

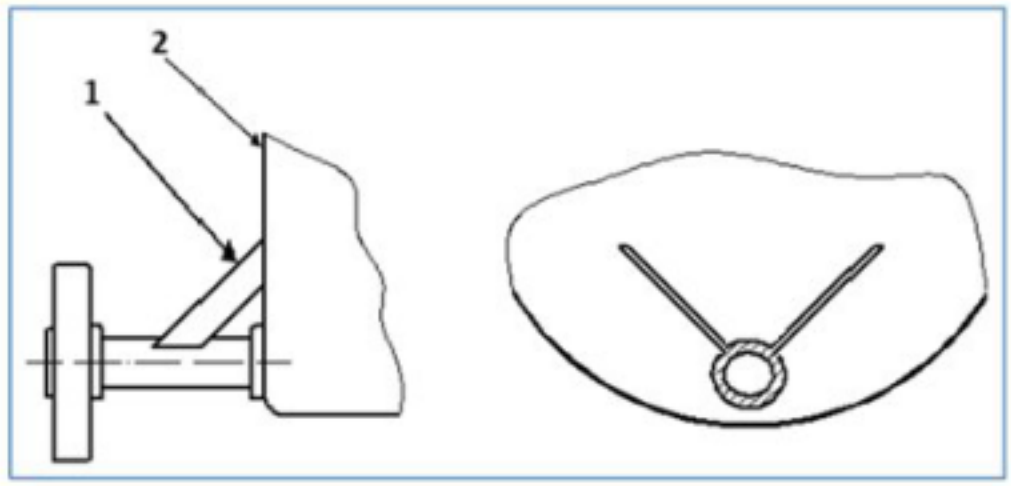
b) Gussets shall be located at or near the connection end of the piping or to the back of the flange and fitted to the closest convenient location on the casing to provide maximum rigidity. The long width of gussets made with bar shall be perpendicular to the pipe and shall be located to avoid interference with the flange bolting or any maintenance areas on the pump.

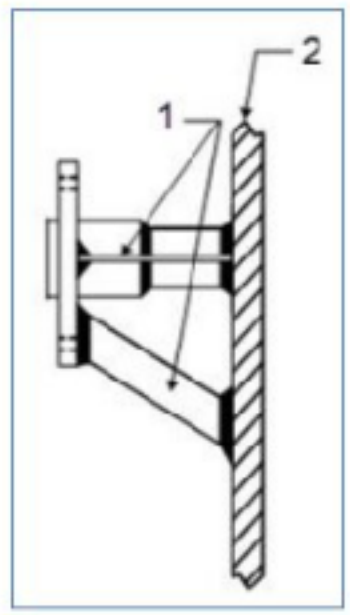
c) Gusset welding shall meet the fabrication requirements (see 6.12.3), and the inspection requirements (see 8.2.1.6) of this standard.

d) Gussets may also be bolted to the casing if drilling and tapping is done prior to hydrostatic test. Proposals to use bolted gusset designs shall be submitted to the purchaser for approval.

e) Gussets welded to the back of the flange shall not overhang the flange edge and shall allow space for installation and tightening of flange fasteners.

f) Welds on the gussets shall not overlap pressure containing weld heat affected zones.





Key
1 two-plane gusseting
2 equipment wall

Figure 22—Gusset Design Examples

6.5 External Nozzle Forces and Moments

6.5.1 Steel and alloy-steel horizontal pumps and their baseplates, vertical in-line pumps with supports anchored to the foundation, and vertically suspended pumps shall be designed to operate if subjected to the forces and moments in Table 5 applied simultaneously to both suction and discharge nozzles in the worst-case combination for the pump.

6.5.2 The design margin between the pump case design capabilities (2-times Table 5, per 6.2.4) and the allowable nozzle load limits (1-times Table 5) shall not be reduced without purchaser's approval.

NOTE For horizontal pumps, two effects of nozzle loads are considered: distortion of the pump casing (see 6.2.3 and 6.2.4) and misalignment of the pump and driver shafts (see 7.4.26). Design margins are intended to account for tolerances allowed in piping design, fabrication, and installation. Exceeding these limits can significantly reduce the pump's mean time between failure as piping system components age and fail to perform as designed and is discouraged.

6.5.3 Allowable nozzle load limits for vertical in-line pumps with supports not anchored to the foundation may be 2-times the values in Table 5 for side nozzles.

6.5.4 For pump casings constructed of materials other than steel or alloy steel, or for pumps with nozzles larger than NPS 24 (DN 600), the vendor shall submit allowable nozzle loads corresponding to the format in Table 5.

6.5.5 The coordinate system(s) shown in Figure 23, Figure 24, Figure 25, Figure 26, and Figure 27 shall be used to apply the forces and moments in Table 5.

6.5.6 Annex F gives methods of qualifying nozzle loads in excess of those in Table 5. The purchaser should be aware that the use of the methods in Annex F can result in a misalignment up to 50 % greater than that based on the loads given in Table 5 and can impact equipment installation criteria. The use of the methods in Annex F requires purchaser's approval.

6.5.6.1 The use of the methods in Annex F does not eliminate the requirement of 2-times margin between external nozzle loads and pump case design limits.

6.5.6.2 The pump vendor shall provide direction to the piping designers with regard to how to apply the equations in Annex F to the system design.

Table 5a—Nozzle Loadings (USC Units)

Location/ Orientation	Nozzle-loading Force as a Function of Flange Size—USC Units											
	Lbf											
	Nominal Size of Flange (NPS)											
	≤ 2	3	4	6	8	10	12	14	16	18	20	24
Each top nozzle												
FX	160	240	320	560	850	1200	1500	1600	1900	2100	2400	2900
FY	130	200	260	460	700	1000	1200	1300	1500	1700	1900	2300
FZ	200	300	400	700	1100	1500	1800	2000	2300	2600	2900	3500
FR	290	430	570	1010	1560	2200	2600	2900	3300	3700	4200	5100
Each side nozzle												
FX	160	240	320	560	850	1200	1500	1600	1900	2100	2400	2900
FY	200	300	400	700	1100	1500	1800	2000	2300	2600	2900	3500
FZ	130	200	260	460	700	1000	1200	1300	1500	1700	1900	2300
FR	290	430	570	1010	1560	2200	2600	2900	3300	3700	4200	5100
Each end nozzle												
FX	200	300	400	700	1100	1500	1800	2000	2300	2600	2900	3500
FY	160	240	320	560	850	1200	1500	1600	1900	2100	2400	2900
FZ	130	200	260	460	700	1000	1200	1300	1500	1700	1900	2300
FR	290	430	570	1010	1560	2200	2600	2900	3300	3700	4200	5100
Moment												
ft-lbf												
Each nozzle												
MX	340	700	980	1700	2600	3700	4500	4700	5400	6100	6800	8300
MY	170	350	500	870	1300	1800	2200	2300	2700	3100	3400	4100
MZ	260	530	740	1300	1900	2800	3400	3500	4000	4500	5100	6100
MR	460	950	1330	2310	3500	5000	6100	6300	7200	8200	9200	11100

NOTE 1 See Figure 23 through Figure 27 for orientation of nozzle loads (X, Y, and Z).

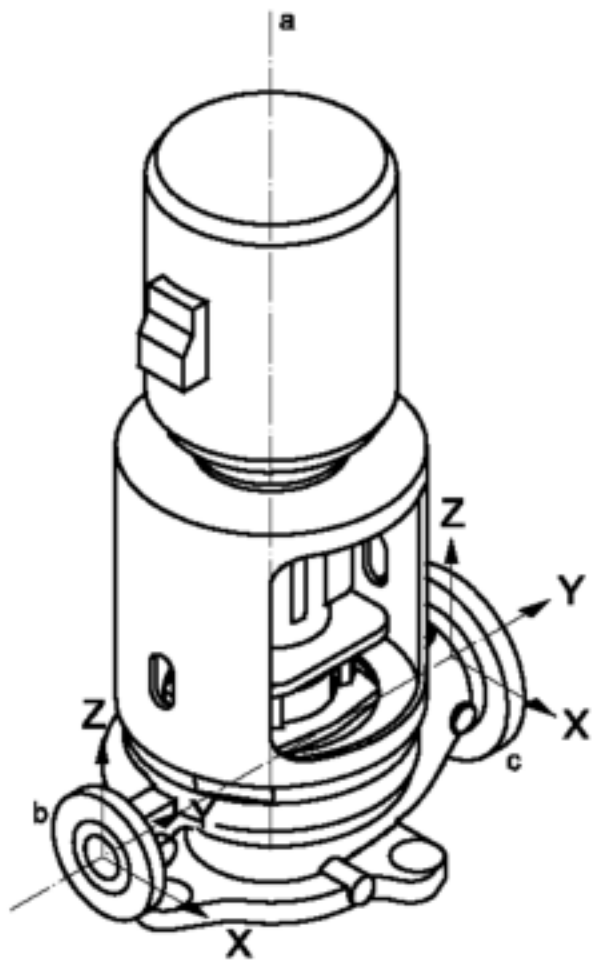
NOTE 2 Each value shown above indicates range from minus that value to plus that value; e.g. 160 indicates a range from −160 to +160.

Table 5b—Nozzle Loadings (SI Units)

Location/ Orientation	Nozzle-loading Force as a Function of Flange Size—SI Units											
	N											
	Nominal Size of Flange (DN)											
	≤ 50	80	100	150	200	250	300	350	400	450	500	600
Each top nozzle												
FX	710	1070	1420	2490	3780	5340	6670	7120	8450	9340	10680	12780
FY	580	890	1160	2050	3110	4450	5340	5780	6670	7560	8450	10230
FZ	890	1330	1780	3110	4890	6670	8000	8900	10230	11560	12780	15570
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850	16460	18680	22690
Each side nozzle												
FX	710	1070	1420	2490	3780	5340	6670	7120	8450	9340	10680	12780
FY	890	1330	1780	3110	4890	6670	8000	8900	10230	11560	12780	15570
FZ	580	890	1160	2050	3110	4450	5340	5780	6670	7560	8450	10230
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850	16460	18680	22690
Each end nozzle												
FX	890	1330	1780	3110	4890	6670	8000	8900	10,230	11560	12780	15570
FY	710	1070	1420	2490	3780	5340	6670	7120	8450	9340	10680	12780
FZ	580	890	1160	2050	3110	4450	5340	5780	6670	7560	8450	10230
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850	16460	18680	22690
Moment												
N-m												
Each nozzle												
MX	460	950	1330	2300	3530	5020	6100	6370	7320	8270	9220	11250
MY	230	470	680	1180	1760	2440	2980	3120	3660	4200	4610	5560
MZ	350	720	1000	1760	2580	3800	4610	4750	5420	6100	6910	8270
MR	620	1280	1800	3130	4710	6750	8210	8540	9820	11100	12470	15050

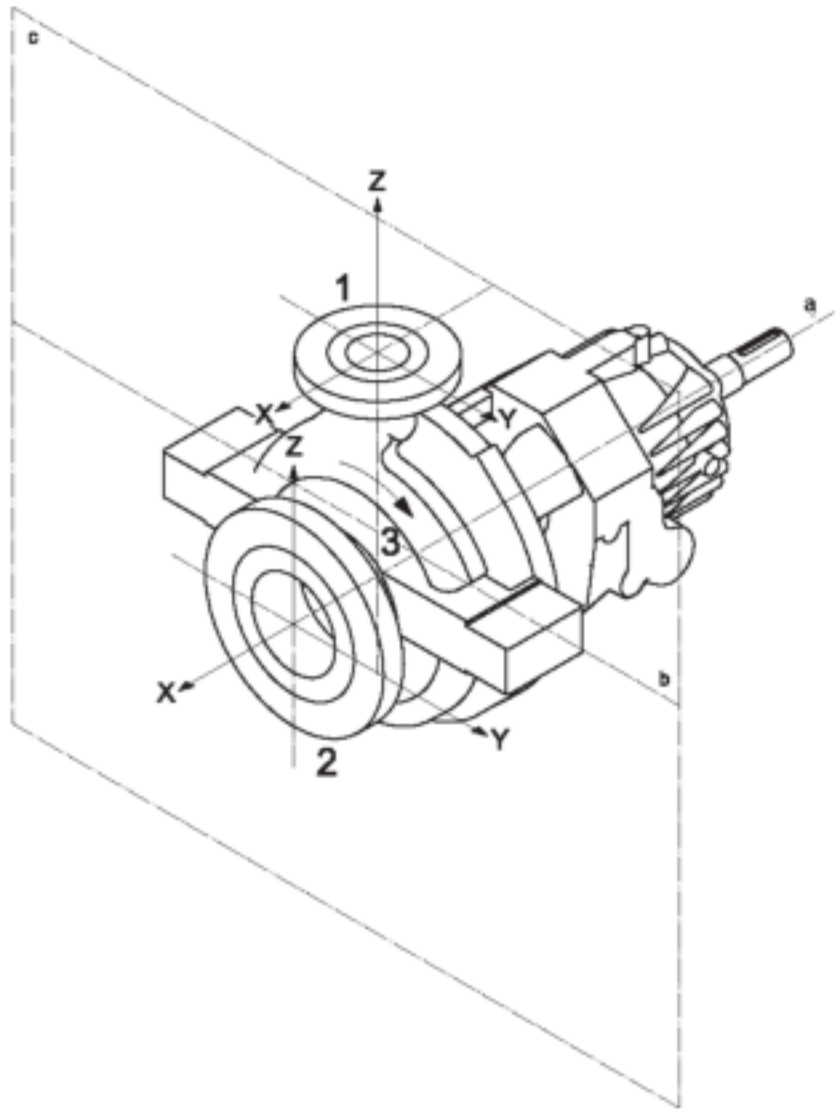
NOTE 1 See Figure 23 through Figure 27 for orientation of nozzle loads (X, Y, and Z).

NOTE 2 Each value shown above indicates range from minus that value to plus that value; e.g. 710 indicates a range from −710 to +710.



[a] Shaft centreline.
[b] Discharge.
[c] Suction.

Figure 23—Coordinate System for the Forces and Moments in Table 5, Vertical In-line Pumps

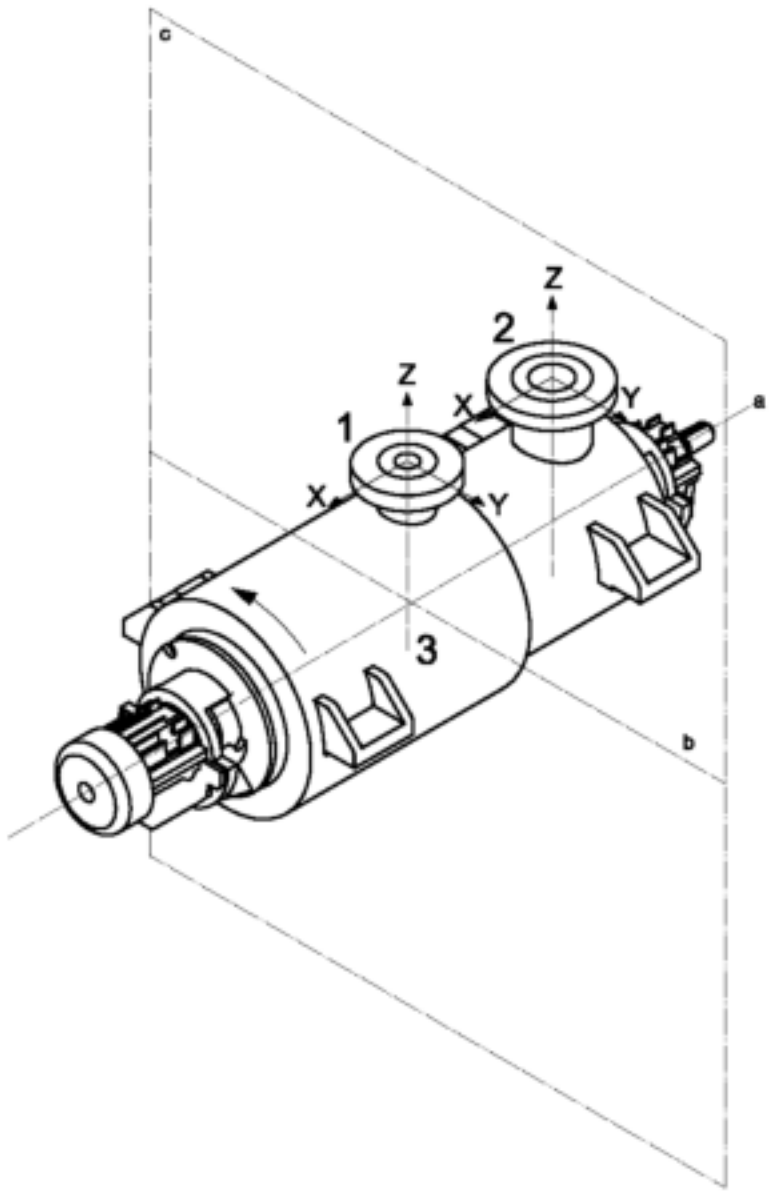


Key

1 discharge nozzle
2 suction nozzle
3 centre of pump

[a] Shaft centreline.
[b] Pedestal centreline.
[c] Vertical plane.

Figure 24—Coordinate System for the Forces and Moments in Table 5, Horizontal Pumps with End Suction and Top Discharge Nozzles

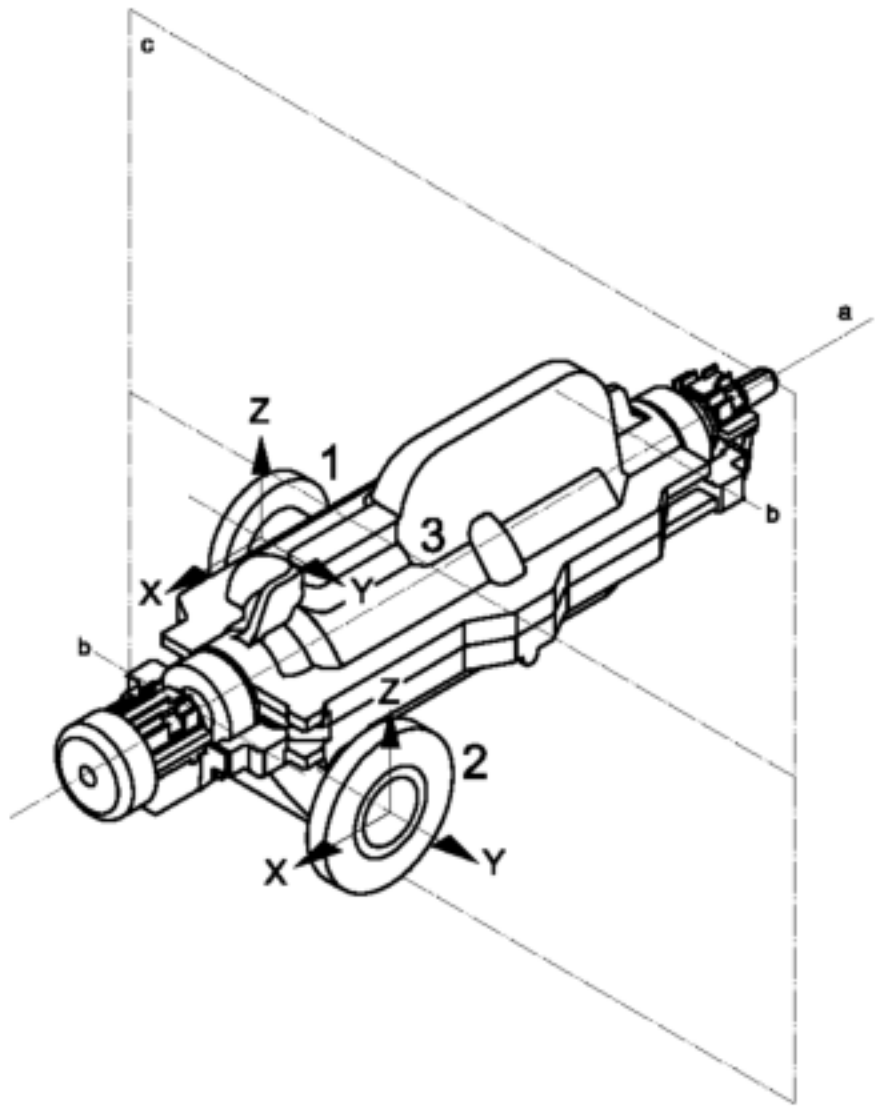


Key

1 discharge nozzle
2 suction nozzle
3 centre of pump

[a] Shaft centreline.
[b] Pedestal centreline.
[c] Vertical plane.

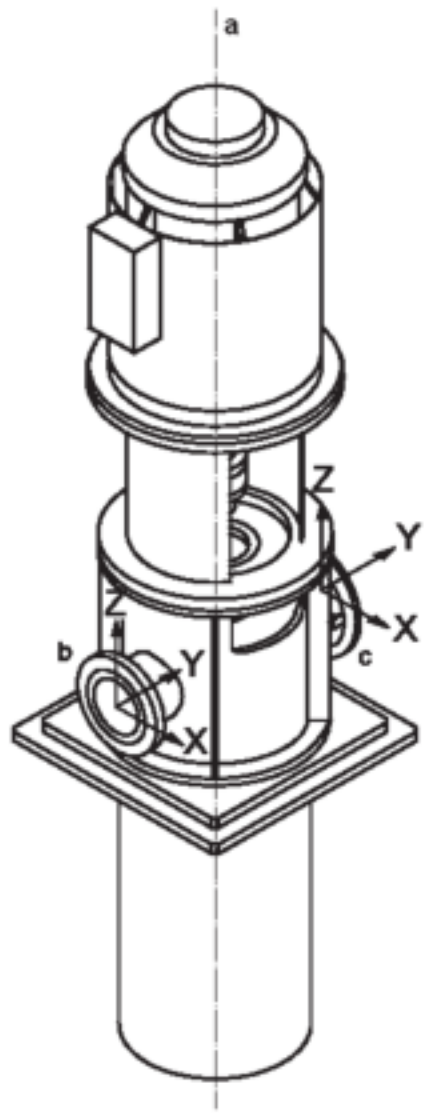
Figure 25—Coordinate System for the Forces and Moments in Table 5, Horizontal Pumps with Top Nozzles



Key

1 discharge nozzle
2 suction nozzle
3 centre of pump
[a] Shaft centreline.
[b] Pedestal centreline.
[c] Vertical plane.

Figure 26—Coordinate System for the Forces and Moments in Table 5, Horizontal Pumps with Side Suction and Side Discharge Nozzles



[a] Shaft centreline.

[b] Discharge.

[c] Suction.

Figure 27—Coordinate System for the Forces and Moments in Table 5, Vertically Suspended, Double-casing Pumps

6.6 Rotors

6.6.1 Impellers shall be of the fully enclosed, semi-open, or open type.

6.6.2 Impellers shall be single-piece castings, forgings, or fabrications.

6.6.3 Impellers shall be keyed to the shaft.

6.6.3.1 Pinning of impellers to the shaft is not acceptable.

6.6.3.2 With the purchaser's approval, collets may be used in vertically suspended pumps.

6.6.3.3 Overhung impellers shall be secured to the shaft by a cap screw or cap nut that does not expose shaft threads.

6.6.3.3.1 The impeller securing device shall be threaded to tighten by liquid drag on the impeller during normal rotation.

6.6.3.3.2 A positive mechanical locking method (e.g. a staked and corrosion-resistant set-screw or a tongue-type washer) is required.

6.6.3.4 Cap screws shall have fillets and a reduced-diameter shank to reduce stress concentrations.

6.6.4 All shaft keyways shall have fillet radii in accordance with ASME B17.1 or ISO 3117.

6.6.5 Impellers shall have solid hubs.

6.6.6 For shafts that require sleeve gaskets to pass over threads, at least 0.06 in. (1.5 mm) radial clearance shall be provided between the threads and the internal diameter of the gasket, and the diameter transition shall be chamfered in accordance with 6.2.19.1.

6.6.7 The shaft-to-seal sleeve fit(s) shall be F7/h6 up to 4.3 in. (110 mm) and F8/h6 for above 4.3 in. (110 mm) to 6.0 in. (150 mm) in accordance with ISO 286 (all parts).

6.6.8 Areas of shafts that can be damaged by set-screws shall be relieved to facilitate the removal of sleeves or other components.

6.6.9 Shafts shall be single-piece construction, machined, and finished throughout their length so that the total indicated runout (TIR) is not more than 0.001 in. (25 µm).

6.6.10 Shafts constructed of multiple pieces assembled together, such as stub shaft extensions to drive lube oil pumps or fans, shall require purchaser's approval.

6.6.11 If non-contacting shaft vibration probes are furnished in accordance with 7.5.2.1, the shaft sensing areas (both radial vibration and axial position) for observation by non-contacting probes shall:

a) be concentric with the bearing journals;

b) be free from stencil and scribe marks or any other surface discontinuity, such as an oil hole or a keyway, for a minimum distance of one probe tip diameter on each side of the probe;

c) not be metallized, sleeved, or plated on rotors made of materials that exhibit consistent electrical properties;

d) have a final surface finish of 32 µin. (0.8 µm) Ra or smoother, preferably obtained by honing or burnishing;

e) be properly degaussed to the levels specified in API 670, or otherwise treated so that the combined total electrical and mechanical runout does not exceed the following:

   1) for areas observed by radial vibration probes, 25 % of the allowed peak-to-peak vibration amplitude or 0.25 mil (6 µm), whichever is greater;

   2) for areas to be observed by axial position probes, 0.5 mil (13 µm).

f) Probe target area width shall ensure that the radial probe's observed surface remains within the burnished probe area for all positions of rotor axial float and thermal expansion with a minimum side clearance of no less than half of the diameter of the probe tip.

   NOTE "All positions of rotor axial float" is inclusive of the maximum allowable rotor travel (i.e. rotor position at the danger setpoint) with the maximum allowable thrust clearance from the supplier.

g) Probe tracking area of the shaft shall be enclosed within the bearing housing for weather protection of probe tip in the shaft sensing area.

6.6.12 Purchaser approval is required for any modification to the shaft sensing areas.

6.6.12.1 If the shaft is made of material that exhibits inconsistent electrical properties, the shaft-sensing areas may be produced by metallizing, plating, or shrink-fitting sleeves (target rings) on the shaft.

NOTE Materials known to exhibit inconsistent electrical properties are high-chromium alloys such as 17–4 PH, duplex stainless steel, and many austenitic stainless steels.

6.6.12.2 The target area shall be finished in accordance with 6.6.11 and modifications and the probe area surface material shall be identified in the technical documentation.

6.6.13 If provisions for mounting non-contacting shaft vibration probes are specified (7.5.2.1), the shaft shall be prepared in accordance with the requirements of 6.6.11 and API 670.

6.6.14 If non-contacting shaft vibration probes are furnished, accurate records of electrical and mechanical runout for the full 360° at each probe location shall be included in the mechanical test report. Electrical and mechanical runout records shall be phase referenced to a permanent location on the shaft such as a key slot.

6.6.15 Repairs to machining errors on rotating components requires purchaser approval prior to performing the repair.

6.6.16 To obtain satisfactory seal performance, the shaft stiffness shall limit the total deflection, under the most severe dynamic conditions over the allowable operating region of the pump with maximum diameter impeller(s) and the specified speed and fluid conditions, to 0.002 in. (50 µm) at the primary seal faces. This shaft-deflection limit may be achieved by a combination of shaft diameter, shaft span or overhang, and casing design (including the use of dual volutes or diffusers).

6.6.17 The number of impeller vanes, as well as the type of volute (single or double), or number of diffuser vanes furnished on impellers and volutes or diffusers shall be provided to the purchaser.

6.7 Wear Rings and Internal Running Clearances

6.7.1 Radial running clearances shall be used to limit internal leakage and, where necessary, balance axial thrust. Impeller pump-out vanes or close axial clearances shall not be used to balance axial thrust.

6.7.2 Renewable wear rings shall be provided in the pump casing.

6.7.3 Impellers shall have either integral wear surfaces or renewable wear rings.

6.7.4 Mating wear surfaces of hardenable materials shall have a difference in Brinell hardness number of at least 50 unless both the stationary and the rotating wear surfaces have Brinell hardness numbers of at least 400.

6.7.5 Renewable metallic wear rings, if used, shall be held in place by a press fit with at least three equally spaced axial or radial screws or by tack welding in at least three equally spaced places. Equal spacing is an approximate location.

6.7.5.1 For axially split pumps (BB1 and BB3), anti-rotation pins on casing wear rings are acceptable.

6.7.5.2 The diameter of a hole in a wear ring for a radial screw shall not be more than one-third the thickness of the wear ring.

6.7.5.3 The center of the axial screw shall be a minimum of 1.5 diameters from the edge of the wear ring or impeller eye.

6.7.6 Nonmetallic wear rings may be installed in holders or directly into the casing. If suitable data exists showing that a press fit will prevent rotation and with purchaser approval, other mechanical anti-rotation features are not required.

6.7.7 All internal running clearances shall meet the following requirements.

a) When establishing internal running clearances between wear rings and other moving parts, consideration shall be given to pumping temperatures, suction conditions, the liquid properties, the thermal expansion and galling characteristics of the materials, and pump efficiency.

b) Clearances shall ensure dependability of operation and freedom from seizure under all specified operating conditions.

c) For cast iron, hardened martensitic stainless steel, and other materials with similarly low galling tendencies, the minimum diametral clearances given in Table 6 shall be used. For materials with higher galling tendencies or for all materials operating at temperatures above 500 °F (260 °C), 0.005 in. (125 µm) shall be added to these diametral clearances.

NOTE If materials with higher galling tendencies are operated at temperatures above 500 °F (260 °C), adding 0.005 in. (125 µm) to the diametral clearances in Table 6 is typically done only once.

d) For nonmetallic wear-ring materials with very low or no galling tendencies (see Table H.3), as well as other wear-ring material combinations with proven operating experience, clearances less than those given in Table 6 may be proposed by the manufacturer to improve pump efficiency. Factors such as distortion and thermal gradients shall be considered to ensure clearances will provide dependability of operation and freedom from seizure under all specified operating conditions.

NOTE There is published data showing successful applications of nonmetallic wear-ring materials with API clearances reduced by 50 %. Reasonable reductions in clearances are believed to be dependent on the materials applied and other service conditions, such as cleanliness and temperature.

Table 6—Minimum Internal Running Clearances

Diameter of Rotating Member at Clearance	Minimum Diametral Clearance	Diameter of Rotating Member at Clearance	Minimum Diametral Clearance
in.	in.	mm	mm
< 2.000	0.010	< 50	0.25
2.000 to 2.499	0.011	50 to 64.99	0.28
2.500 to 2.999	0.012	65 to 79.99	0.30
3.000 to 3.499	0.013	80 to 89.99	0.33
3.500 to 3.999	0.014	90 to 99.99	0.35
4.000 to 4.499	0.015	100 to 114.99	0.38
4.500 to 4.999	0.016	115 to 124.99	0.40
5.000 to 5.999	0.017	125 to 149.99	0.43
6.000 to 6.999	0.018	150 to 174.99	0.45
7.000 to 7.999	0.019	175 to 199.99	0.48
8.000 to 8.999	0.020	200 to 224.99	0.50
9.000 to 9.999	0.021	225 to 249.99	0.53
10.000 to 10.999	0.022	250 to 274.99	0.55
11.000 to 11.999	0.023	275 to 299.99	0.58
12.000 to 12.999	0.024	300 to 324.99	0.60
13.000 to 13.999	0.025	325 to 349.99	0.63
14.000 to 14.999	0.026	350 to 374.99	0.65
15.000 to 15.999	0.027	375 to 399.99	0.68
16.000 to 16.999	0.028	400 to 424.99	0.70
17.000 to 17.999	0.029	425 to 449.99	0.73
18.000 to 18.999	0.030	450 to 474.99	0.75
19.000 to 19.999	0.031	475 to 499.99	0.78
20.000 to 20.999	0.032	500 to 524.99	0.80
21.000 to 21.999	0.033	525 to 549.99	0.83
22.000 to 22.999	0.034	550 to 574.99	0.85
23.000 to 23.999	0.035	575 to 599.99	0.88
24.000 to 24.999	0.036	600 to 624.99	0.90
25.000 to 25.999	0.037	625 to 649.99 [a]	0.95

[a] For diameters greater than 25.999 in. (649.99 mm), the minimum diametral clearances shall be 0.037 in. (0.95 mm) plus 0.001 in. for each additional 1.0 in. of diameter or fraction thereof (1 µm for each additional 1.0 mm of diameter).

6.8 Mechanical Shaft Seals

6.8.1 Pumps shall be equipped with mechanical seals and sealing systems in accordance with API 682.

6.8.1.1 Pump and seal interface dimensions shall be in accordance with Table 7A for Category 2 and Category 3 seals, and Table 7B for High Pressure Seals.

6.8.1.1.1 Figure 28 of this standard shall apply to both Tables 7A and 7B.

[•] **6.8.1.1.2** The purchaser shall specify the category of seal required.

NOTE API 610, 13th edition introduces chamber sizes 11–14 for larger pump sizes.

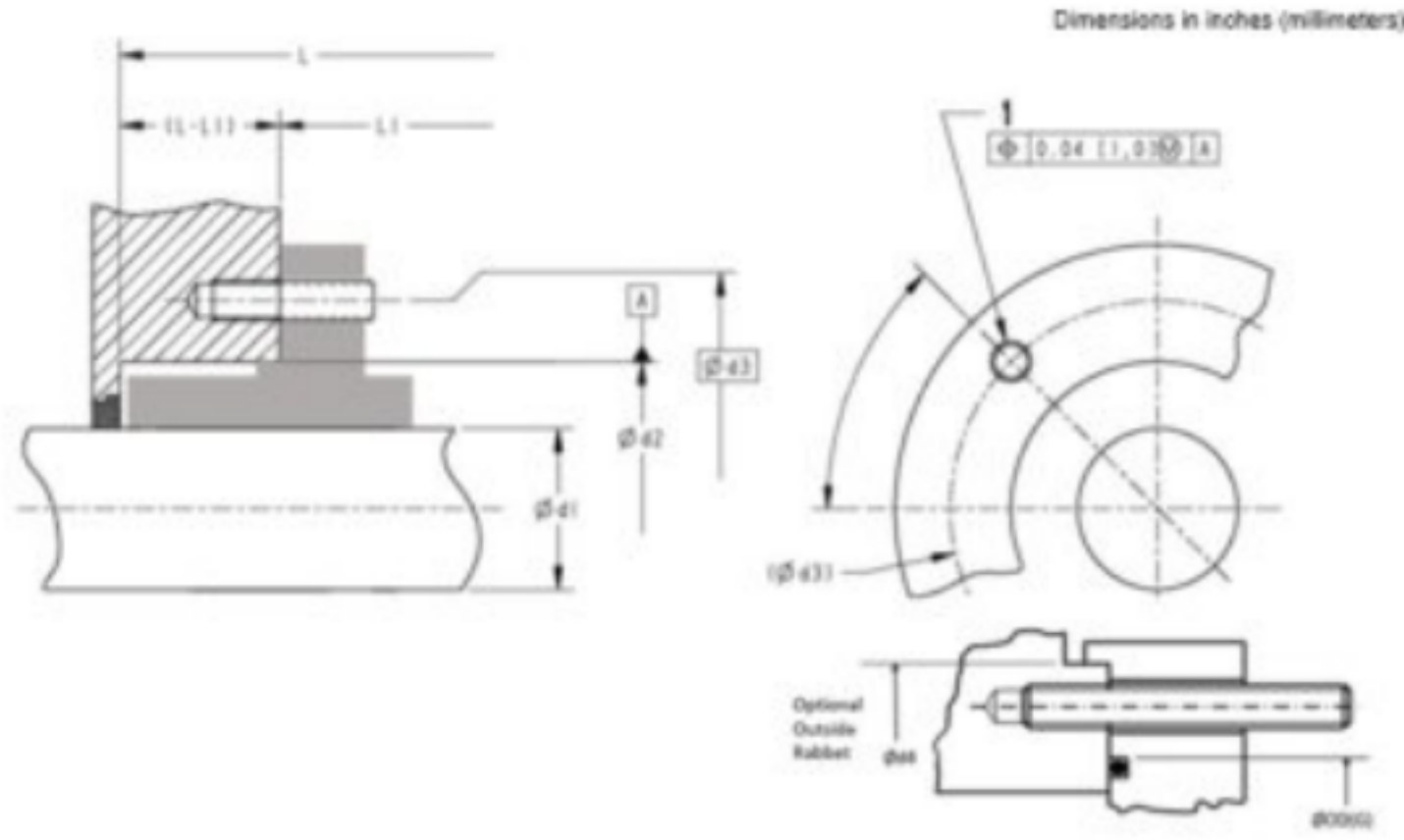
6.8.1.1.3 Seal mounting studs shall straddle centerlines of the main axes of the pump, as shown in Figure 28.

6.8.1.1.4 Alternative seal chamber designs may be proposed with proven experience and purchaser approval.

6.8.1.2 The vendor shall advise purchaser of any non-compliance with Table 7A or Table 7B.

6.8.2 The seal cartridge shall be removable without disturbing the driver except for vertical pumps type OH5 and for vertical pumps type OH6 with the motor mounted directly on the pump gearbox.

6.8.3 For pumps with flange and pump pressure ratings in excess of the minimum values in 6.2.8, the gland stud size and bolt circle diameter may be increased. Larger studs shall only be used to meet the stress requirements of 6.2.5 or to compress spiral-wound gaskets enough in accordance with manufacturer's specifications.



Key

1 gland studs

L total length to nearest obstruction

L_1 length from seal chamber face to nearest obstruction

Figure 28—Chamber Diagrams

Table 7a—Standard Dimensions for Category 2 and 3 Seal Chambers, Seal Gland Mounting, and Cartridge Mechanical Seal Sleeves

Seal Chamber Size No.	Shaft Diameter Max.[a] d1		Seal Chamber Bore [b] d2		Gland Stud Circle d3		Outside Gland Rabbet [c] d4		Total Length Min. L		Clear Length Min. [d] L1		No. of Studs [e] N(stud)	Stud Size Ø	
	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	—	mm	in.
1	20	0.7874	70	2.7559	105	4.134	125	4.921	150	5.906	100	3.937	4	12	0.500
2	30	1.1811	80	3.1496	115	4.528	135	5.315	155	6.102	100	3.937	4	12	0.500
3	40	1.5748	90	3.5433	125	4.921	145	5.709	160	6.299	100	3.937	4	12	0.500
4	50	1.9685	100	3.9370	140	5.512	165	6.496	165	6.496	110	4.331	4	16	0.625
5	60	2.3622	120	4.7244	160	6.299	185	7.283	170	6.693	110	4.331	4	16	0.625
6	70	2.7559	130	5.1181	170	6.693	195	7.677	175	6.890	110	4.331	4	16	0.625
7	80	3.1496	140	5.5118	180	7.087	205	8.071	180	7.087	110	4.331	4	16	0.625
8	90	3.5433	160	6.2992	205	8.071	235	9.252	185	7.283	120	4.724	4	20	0.750
9	100	3.9370	170	6.6929	215	8.465	245	9.646	190	7.480	120	4.724	4	20	0.750
10	110	4.3307	180	7.0866	225	8.858	255	10.039	195	7.677	120	4.724	4	20	0.750
11	120	4.7244	195	7.6772	250	9.843	280	11.024	200	7.874	135	5.315	4	20	0.750
12	130	5.1181	205	8.0709	260	10.236	290	11.417	205	8.071	140	5.512	4	20	0.750
13	140	5.5118	215	8.4646	270	10.630	300	11.811	210	8.268	145	5.709	4	20	0.750
14	150	5.9055	225	8.8583	280	11.024	310	12.205	215	8.465	150	5.906	4	20	0.750

NOTE Dimensions in inches (millimeters).

[a] Dimensions to tolerance class h6.

[b] Dimensions to tolerance class H7; For axially split pumps, an additional tolerance of 0.003 in. (75 µm) to allow for gasket thickness

[c] Dimensions to tolerance class f7.

[d] Shaft deflection criteria (6.6.16) may require the reduction of dimensions L and L1 on size 1 and size 2 seal chambers to below the minimum values listed, depending on specific pump construction and casing design. Size 1 and size 2 seal chambers are not usually found on Type OH2 and OH3 pumps.

[e] B7 studs were used for all calculation basis for this table, users are cautioned that if other materials are selected stud stress calculation are required to confirm suitability.

Table 7b—Standard Dimensions for High Pressure Seal Chambers, Seal Gland Mounting, and Cartridge Mechanical Seal Sleeves

Seal Chamber Size No.	Shaft Diameter Max.[a] d1		Seal Chamber Bore [b] d2		Gland Stud Circle d3		Gasket Diameter Max. OD(G)		Total Length Min. L		Clear Length Min. L1		No. of Studs [c] N(stud)	Stud Size Ø	
	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	—	mm	in.
1	20	0.7874	90	3.5433	130	5.118	106	4.173	175	6.890	100	3.937	4	16	0.625
2	30	1.1811	100	3.9370	150	5.906	122	4.803	175	6.890	100	3.937	4	20	0.750
3	40	1.5748	110	4.3307	160	6.299	132	5.197	175	6.890	100	3.937	4	20	0.750
4	50	1.9685	120	4.7244	170	6.693	141	5.551	175	6.890	110	4.331	4	20	0.750
5	60	2.3622	130	5.1181	185	7.283	154	6.063	185	7.283	110	4.331	4	22	0.875
6	70	2.7559	140	5.5118	195	7.677	164	6.457	200	7.874	110	4.331	4	22	0.875
7	80	3.1496	150	5.9055	205	8.071	174	6.850	200	7.874	120	4.724	4	22	0.875
8	90	3.5433	170	6.6929	230	9.055	191	7.520	220	8.661	120	4.724	4	24	1.000
9	100	3.9370	180	7.0866	240	9.449	206	8.110	220	8.661	120	4.724	8	24	1.000
10	110	4.3307	190	7.4803	250	9.843	216	8.504	220	8.661	120	4.724	8	24	1.000
11	120	4.7244	220	8.6614	280	11.024	246	9.685	240	9.449	130	5.118	8	24	1.000
12	130	5.1181	230	9.0551	290	11.417	256	10.079	240	9.449	130	5.118	8	24	1.000
13	140	5.5118	240	9.4488	300	11.811	266	10.472	240	9.449	150	5.906	8	24	1.000
14	150	5.9055	250	9.8425	310	12.205	276	10.866	240	9.449	150	5.906	8	24	1.000

[a] Dimensions to tolerance class h6.

[b] Dimensions to tolerance class H7; For axially split pumps, an additional tolerance of 0.003 in. (75 µm) to allow for gasket thickness

[c] B7 studs were used for all calculation basis for this table, users are cautioned that if other materials are selected stud stress calculation are required to confirm suitability.

High pressure seals are generally intended for BB1, BB3, BB5, VS1, VS2, VS6, and VS7.

6.8.4 Provisions shall be made to center the seal gland and chamber with either an ID or OD register fit.

6.8.4.1 The register fit surface shall be concentric to the shaft and shall have a TIR of not more than 0.005 in. (125 µm).

6.8.4.2 Using the seal-gland bolts to center mechanical seal components is not acceptable (see Figure 29).

6.8.4.3 To account for bearing internal clearances, TIR measurements shall be taken with the assembly in the vertical position for overhung pumps.

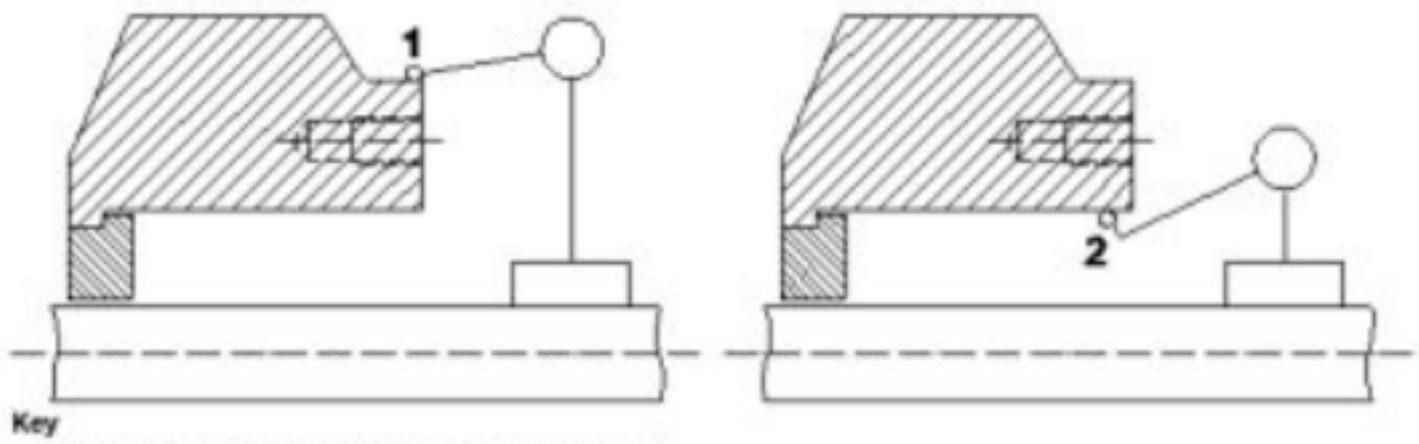
6.8.5 Seal chamber face runout (TIR), in relation to shaft centerline, shall not exceed 0.0005 in./in. (0.5 µm/mm) of seal chamber bore (see Figure 30).

6.8.6 The mating joint between the seal gland and the seal chamber face shall incorporate a confined gasket to prevent blowout.

6.8.6.1 The gasket shall be of the controlled-compression type, e.g. an O-ring or a spiral-wound gasket with a metal-to-metal joint contact.

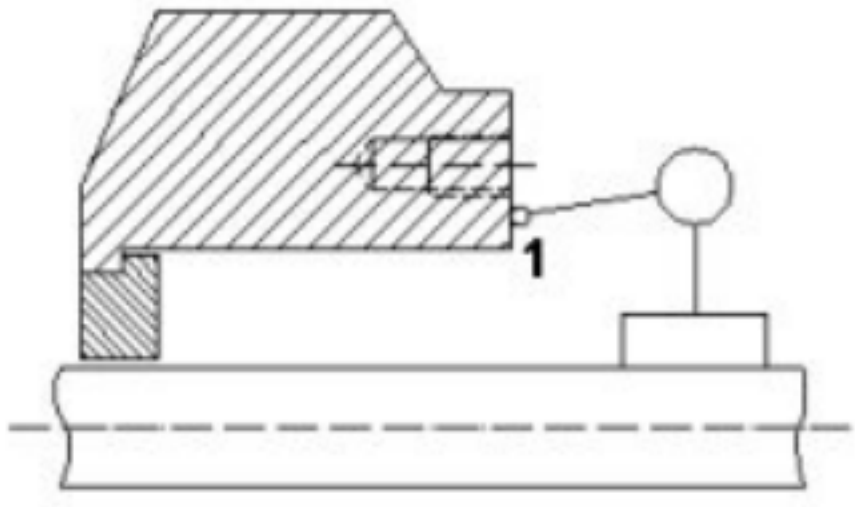
6.8.6.2 If space or design limitations make this requirement impractical, an alternative seal gland design shall be submitted to the purchaser for approval.

6.8.7 Specified seal and pump connections shall be identified by symbols in accordance with API 682 that are permanently marked on the components using stamping, casting, electro-chemical etching, or similar method.



Key
1 location of outside diameter register fit measurement
2 location of inside diameter register fit measurement

Figure 29—Seal Chamber Concentricity



Key
1 location of face runout measurement

Figure 30—Seal Chamber Face Runout

6.8.8 Seal glands and seal chambers shall only have those connections that are required by the seal flush plan. Any additional unused connections shall be plugged in accordance with 6.4.3.7.

6.8.9 Provision shall be made to ensure complete venting of the seal chamber.

[•] **6.8.10** If specified or if required by the vendor or seal vendor, jackets shall be provided on seal chambers for heating. Heating requirements shall be agreed upon by the purchaser, vendor, and seal manufacturer for high-melting-point products.

6.8.11 Mechanical seals and glands for all pumps, except vertically suspended pumps shipped without mounted drivers, shall be installed in the pump before shipment and shall be clean and ready for initial service.

6.8.12 If seals require final adjustment or installation in the field, the vendor shall attach a metal tag warning of this requirement.

6.8.13 The maximum static and dynamic sealing pressures that can be anticipated to occur in the seal chamber shall be determined per API 682 and agreed between the vendor and purchaser.

[•] **6.8.14** If specified, seal chambers shall be provided with an additional flush port at approximately the center of the chamber and extending vertically upward. This additional flushing port shall be designed in accordance with casing auxiliary connection requirements (refer to 6.4.3).

6.9 Dynamics

6.9.1 General

6.9.1.1 The topics of critical speed and lateral analysis are covered under each specific pump type in Section 9.

6.9.1.2 The rotor of one- and two-stage horizontal pumps shall be designed so the first dry-bending critical speed (subcritical) is at least 20 % above the pump's maximum continuous operating speed. Alternatively, an analysis per Annex I may be performed.

6.9.2 Torsional Analysis

6.9.2.1 The flowchart in Figure 31 may be used to determine the torsional analysis performed.

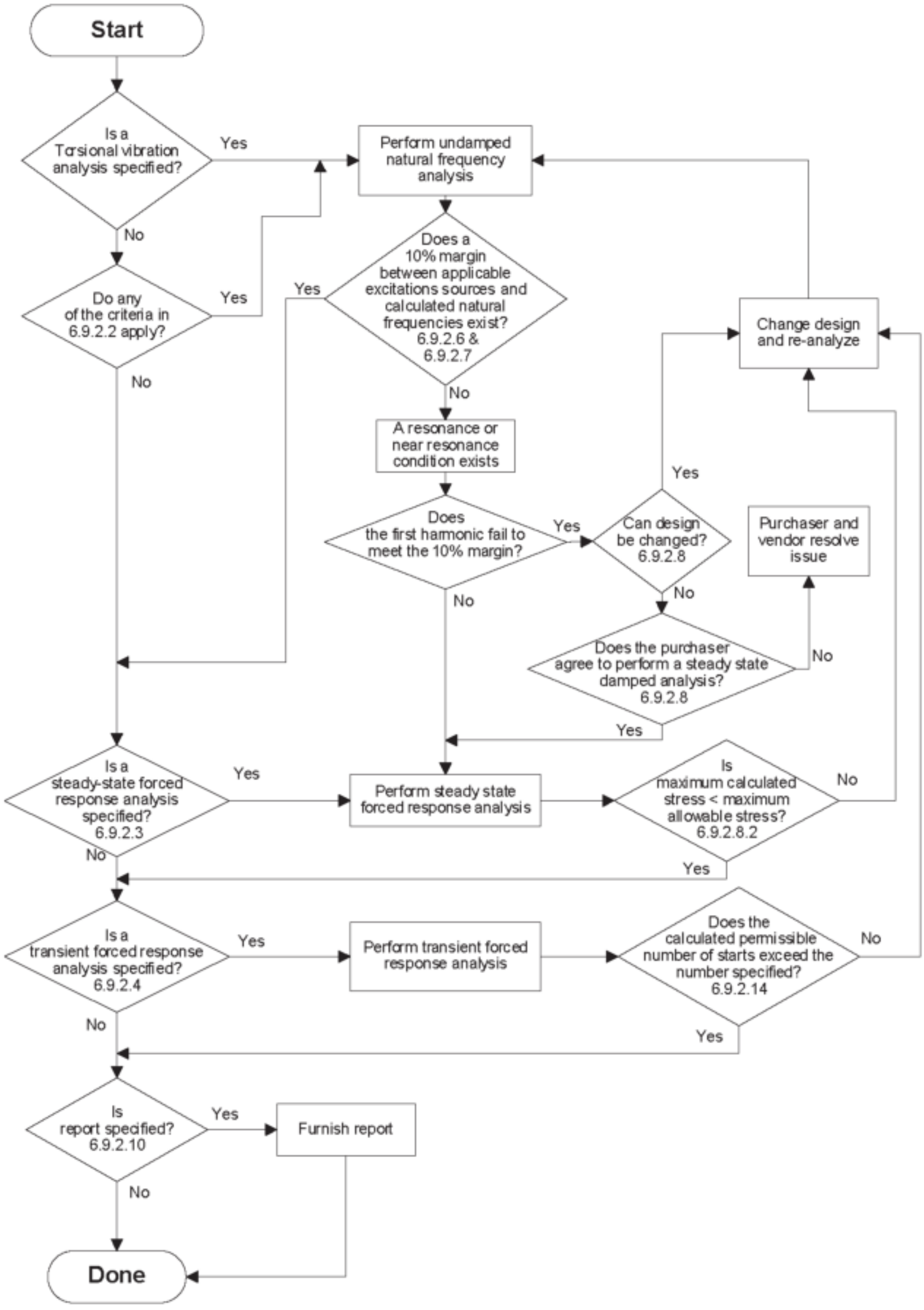


Figure 31—Torsional Analysis Flowchart

6.9.2.2 For trains meeting the following conditions, the vendor having unit responsibility shall ensure that a torsional vibration analysis of the complete coupled train is performed and be responsible for directing any modifications necessary to meet the requirements of 6.9.2.3 through 6.9.2.8.

a) multiple couplings: trains comprised of three or more coupled machines rated 2000 hp (1500 kW) or higher;

b) gears: rated 2000 hp (1500 kW) or higher;

c) internal combustion engines: trains with engines rated 335 hp (250 kW) or higher;

d) synchronous motors: trains with synchronous motors rated 750 hp (550 kW) or higher;

e) AFDs rated 750 hp (550 kW) or higher;

f) vertically suspended pumps: with drivers rated 1000 hp (750 kW) or larger;

g) induction motors: trains with induction (asynchronous) motors rated 2000 hp (1500 kW) or higher.

[•] **6.9.2.3** If specified, for AFDs meeting 6.9.2.2.e), a steady-state, forced response analysis shall be performed.

[•] **6.9.2.4** If specified, a transient forced response analysis shall be performed. If performed, the time-transient analysis shall meet the requirements of 6.9.2.10 through 6.9.2.14.

6.9.2.5 Excitation of torsional natural frequencies may come from many sources and should be considered in the analysis. These sources shall include but are not limited to the following:

a) pumps and HPRTs: impeller vane and cutwater pass frequencies

b) gear characteristics such as mesh frequencies;

c) torsional excitation resulting from electric motors and engines;

d) one and two times electrical line frequency;

e) one and two times operating speed(s);

f) harmonic frequencies from AFD's

NOTE Operating speed of centrifugal pumps driven by an adjustable speed driver is inclusive of all speeds from minimum speed to maximum continuous speed for AFD motor driven and from minimum governor to trip speed on steam turbine, gas turbine, and reciprocating engine driven assets.

6.9.2.6 The intersection of the first harmonic shall be at least 10 % above or 10 % below the specified operating speed range (from minimum to maximum continuous speed).

6.9.2.6.1 If the separation is within these margins, the pump train shall be modified to meet this requirement.

6.9.2.6.2 If the natural frequency cannot be moved, it shall be shown to have no adverse effect by performing a steady state forced response analysis.

6.9.2.7 The intersection of all other torsional natural frequencies with any possible excitation frequency should be at least 10 % above or 10 % below the specified operating speed range (from minimum to maximum continuous speed). If the frequency falls within this margin, the vendor shall perform a steady state forced response analysis to demonstrate that the resonance has no adverse effect on the complete train.

6.9.2.8 The assumptions made in this analysis regarding the magnitude of excitation and the degree of damping shall be clearly stated.

6.9.2.8.1 The acceptance criteria for this analysis shall be agreed upon by the purchaser and the vendor.

6.9.2.8.2 The analysis shall show that all shaft sections, couplings, and gear mesh have infinite life using agreed criteria.

NOTE Typically, steady-state forced response torsional analyses of pumps driven by pulse-width-modulated adjustable frequency drives have shown acceptably low stresses at the resonant conditions due to AFD excitations; these have no adverse effects on the machinery train.

6.9.2.9 If only an undamped torsional analysis is performed, a Campbell diagram with a tabulation of the mass elastic data and a brief explanation of the calculation method may be furnished to the purchaser in lieu of a report.

[•] **6.9.2.10** If specified, or if either a steady-state or transient forced response analysis is performed, the manufacturer shall furnish a detailed report of the torsional analysis. The report shall include the following:

a) description of the method(s) used to calculate the train torsional natural frequencies;

b) diagram of the mass elastic system;

c) table of the mass moment and torsional stiffness of each element of the mass elastic system;

d) Campbell diagram;

e) mode-shape diagram with peak stresses shown for each resonant frequency.

6.9.2.11 In addition to the parameters used to perform an undamped torsional natural frequency analysis as specified in 6.9.2.2, the following shall be included in the transient forced response analysis:

a) motor average torque, as well as pulsating torque (direct and quadrature axis) vs speed characteristics;

b) load torque vs speed characteristics;

c) electrical system characteristics affecting the motor terminal voltage or the assumptions made concerning the terminal voltage, including the method of starting, such as across the line, or some method of reduced voltage starting.

6.9.2.12 A forced response analysis shall generate the maximum torque and, in case of a transient forced response analysis, a torque vs time history for each of the shafts in the train.

6.9.2.12.1 The maximum torques shall be used to evaluate the peak torque capability of coupling components, gearing and interference fits of components, such as coupling hubs, and shaft locations with stress raisers such as keyways.

6.9.2.12.2 The torque vs time history shall be used to develop a cumulative damage-fatigue analysis of shafting, keys, and coupling components.

6.9.2.13 Appropriate fatigue properties and stress concentrations shall be used.

6.9.2.14 An appropriate cumulative fatigue algorithm shall be used to develop a value for the safe number of starts and short-circuit events. The purchaser and vendor shall agree as to the safe number of such events.

NOTE Values used depend on the analytical model used and the vendor's experience. Values of 1000 to 1500 starts are common. API 541 requires 5000 starts, which is a reasonable assumption for a motor. The driven equipment, however, would be overdesigned to meet this requirement.

6.9.3 Balancing

6.9.3.1 Impellers, balancing drums, and similar major rotating components shall be dynamically balanced to ISO 21940-11, Grade G2.5.

6.9.3.1.1 The mass of the arbor used for balancing shall not exceed the mass of the component being balanced.

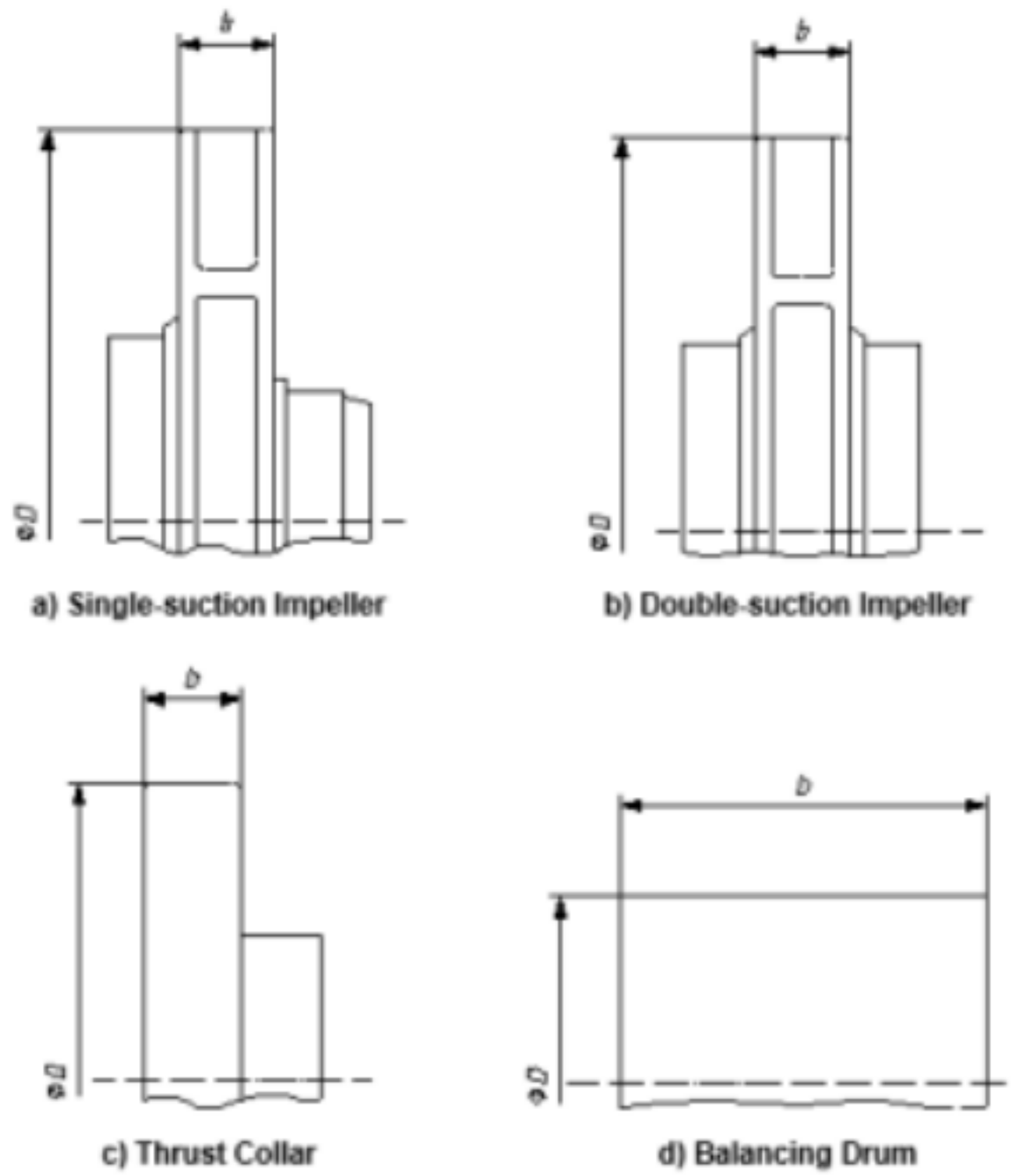
6.9.3.1.2 Shafts are not required to be balanced.

6.9.3.1.3 For single-stage BB1 and BB2 pump rotors with interference fit components, the vendor may choose to balance the assembled rotor (in accordance with 9.2.4.2) instead of balancing major rotating components individually.

6.9.3.2 Component balancing may be single-plane if the ratio D/b (see Figure 32) is 6 or greater.

6.9.3.3 Rotor balancing shall be performed as required in the specific pump sections.

[•] **6.9.3.4** If specified, impellers, balancing drums, and similar rotating components shall be dynamically balanced to ISO 21940–11, Grade G1.



Key

b width

D diameter

Figure 32—Rotating Component Dimensions to Determine if Single-plane Balancing Is Allowable

[•] **6.9.3.5** If specified, impellers, balancing drums, and similar rotating components shall be dynamically balanced to:

$$U = 4W/n \quad (2)$$

In USC unitswhere

U is the unbalance per plane, expressed in ounce-inches;

W is the component mass (for components), expressed in pounds; or the load per balancing machine journal (for rotors), expressed in pounds;

N is the rotational speed of the pump, expressed in revolutions per minute.

NOTE 4 W/n is a balance tolerance denominated solely in USC units. With modern balancing machines, it is feasible to balance components mounted on their arbors to $U = 4W/n$ (USC units), or even lower depending upon the mass of the assembly, and to verify the unbalance of the assembly with a residual unbalance check. However, balancing to ISO Grade G1 or $U = 4W/n$ (USC units) is generally not repeatable when the components are dismantled and remounted on their arbors, because of the minute mass eccentricity values associated with these fine levels of unbalance requirements. See 9.2.4.2.2.

6.9.4 Vibration

6.9.4.1 The allowable operating region shall be stated in the proposal and meet the criteria in Table 8 and Table 9 for testing.

6.9.4.2 If the allowable operating region is limited by a factor other than vibration, that factor shall also be stated in the proposal. Figure 33 illustrates typical vibration behavior with the allowable and preferred operating region.

6.9.4.3 During the performance test, overall vibration measurements and discrete vibration measurements using fast Fourier transform (FFT) spectra shall be made at a minimum of five test points. These points shall include the rated flow point and the points defining the minimum and maximum flows for the allowable and preferred operating ranges. Vibration measurements are not required at shutoff.

6.9.4.4 The upper frequency range limit (F_{max}) of the vibration analyzer should be set based on the pump bearing configuration and pump operating speed.

6.9.4.4.1 For pumps equipped with hydrodynamic radial and thrust bearings, an F_{max} of 1,000 Hz shall be used.

6.9.4.4.2 Pumps equipped with antifriction radial or thrust bearings operating at 1,800 rpm and 3,600 rpm shall have an F_{max} of no less than 1,000 Hz and 2,000 Hz respectively, with a recommended F_{max} of 5,000 Hz.

6.9.4.5 Vibration measurements shall be made at the following locations:

a) on the bearing housing(s) or equivalent location(s) of all pumps, at the positions shown on Figure 34, Figure 35, and Figure 36 (it is recommended that vibration readings be taken in both horizontal and vertical planes at the rated point and at all other points in the same plane as the higher reading). Refer to API-670 and HI 9.6.4 for mounting surface dimensions.

b) on the shaft of pumps with hydrodynamic bearings both X and Y, if the pump has proximity probes.

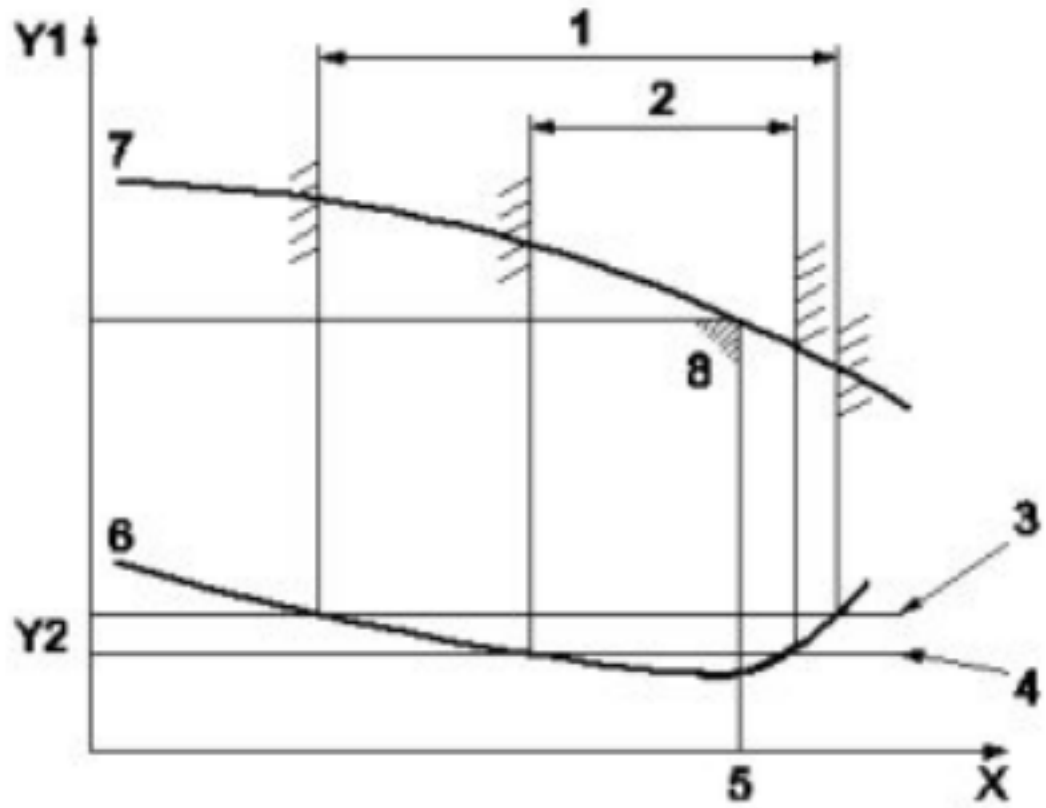
6.9.4.6 The FFT spectra shall include the range of frequencies from 5 Hz to 2Z times running speed (where Z is the number of impeller vanes; in multistage pumps with different impellers, Z is the highest number of impeller vanes in any stage). The plotted spectra shall be included with the pump test results.

NOTE The discrete frequencies 1.0, 2.0, and Z times running speed are associated with various pump phenomena and are, therefore, of particular interest in the spectra.

6.9.4.7 For pumps equipped with antifriction bearings (radial or thrust), the kinematic fault frequencies associated with bearing defects shall be identified in the spectra.

6.9.4.8 Bearing-housing overall vibration measurements shall be made in root mean square (RMS) velocity, expressed in inches per second (millimeters per second).

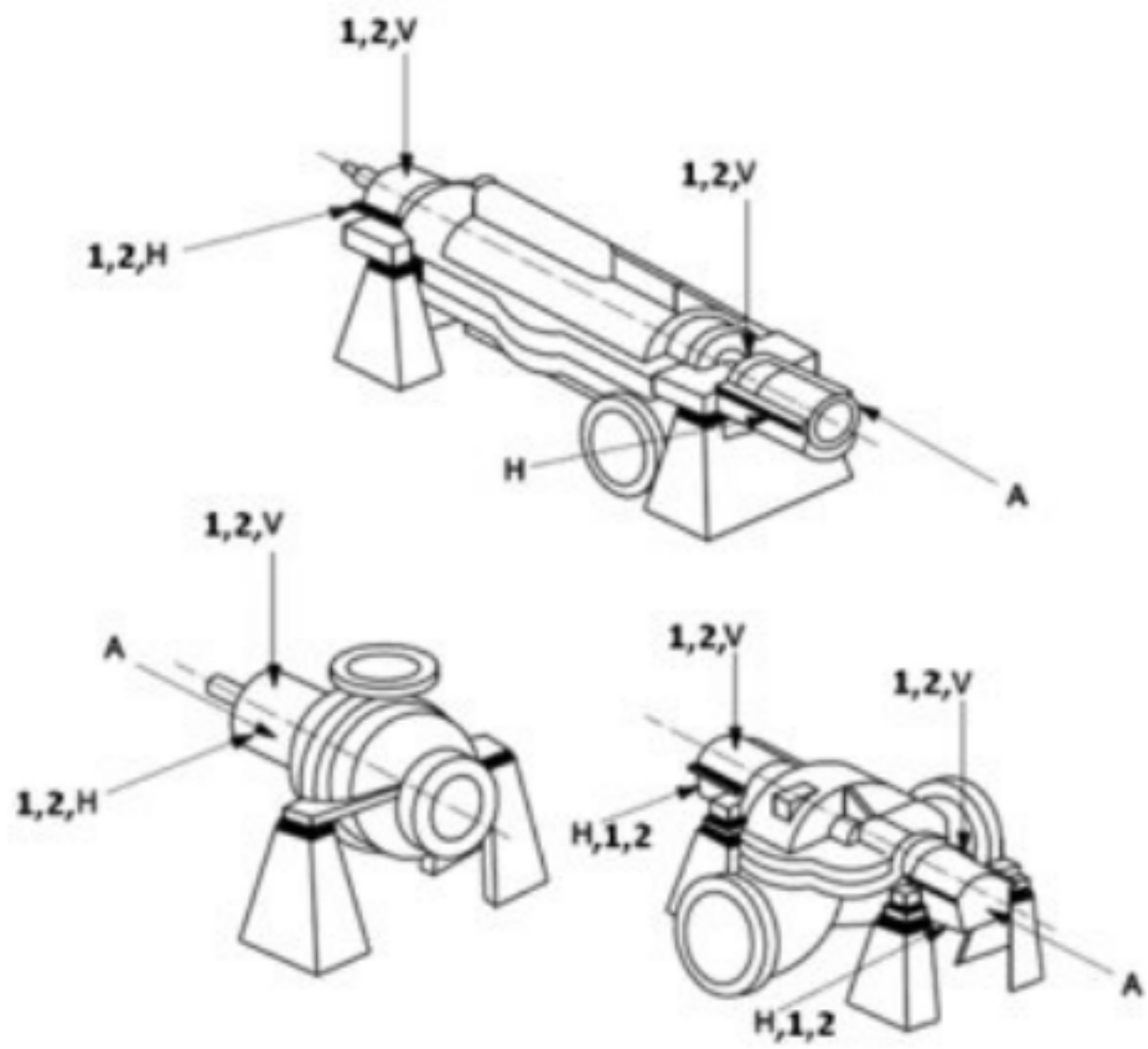
6.9.4.9 Shaft vibration measurement shall be peak-to-peak displacement, in mils (micrometers).



Key

X flowrate
Y1 head
Y2 vibration
1 allowable operating region of flow
2 preferred operating region of flow
3 maximum allowable vibration limit at flow limits
4 basic vibration limit
5 best efficiency point, flowrate
6 typical vibration vs flowrate curve showing maximum allowable vibration
7 head-flowrate curve
8 best efficiency point, head and flowrate

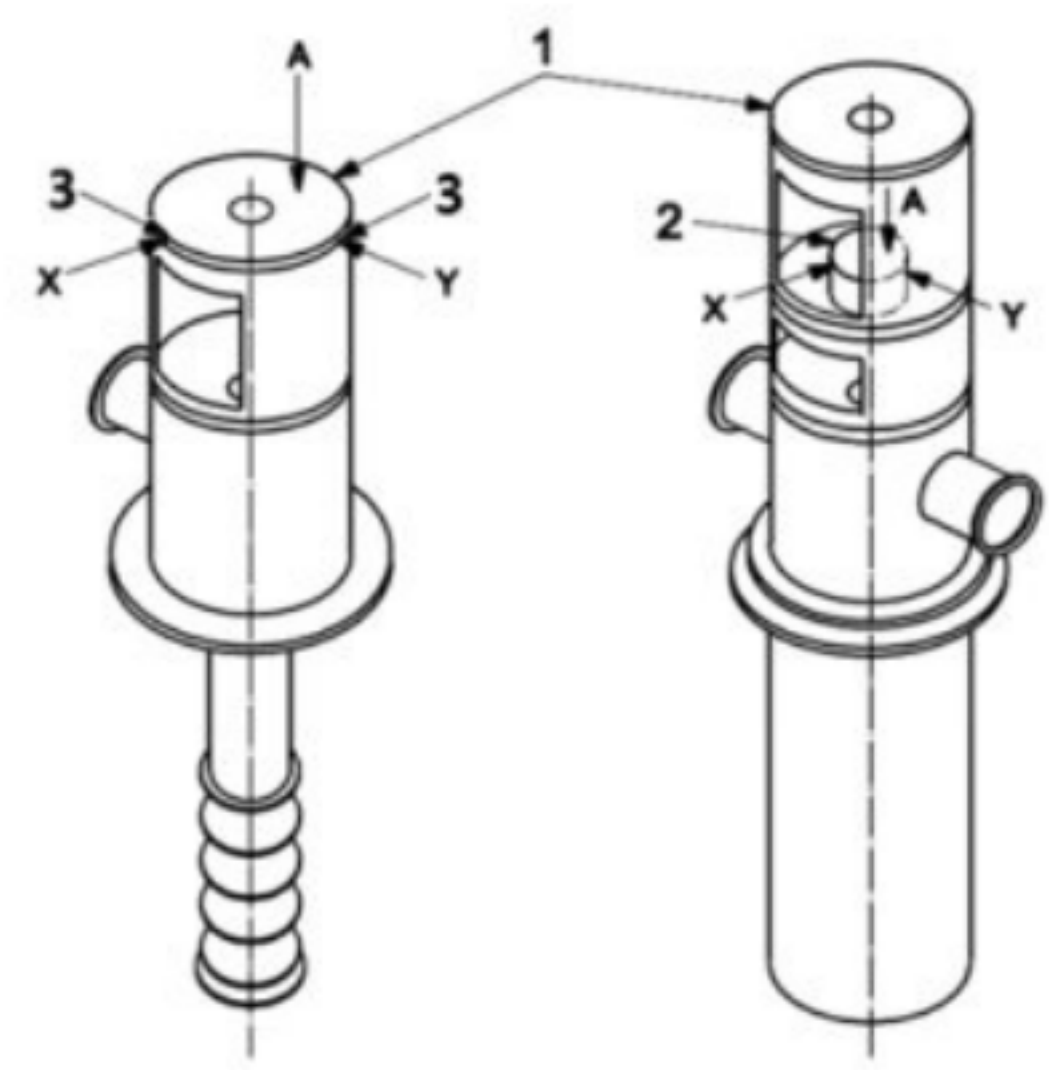
Figure 33—Relationship Between Flow and Vibration



Key

1 spot face (see 6.10.2.12)
2 optional arrangement for mounting vibration-measuring equipment (see 6.10.2.13)
A axial
H horizontal
V vertical

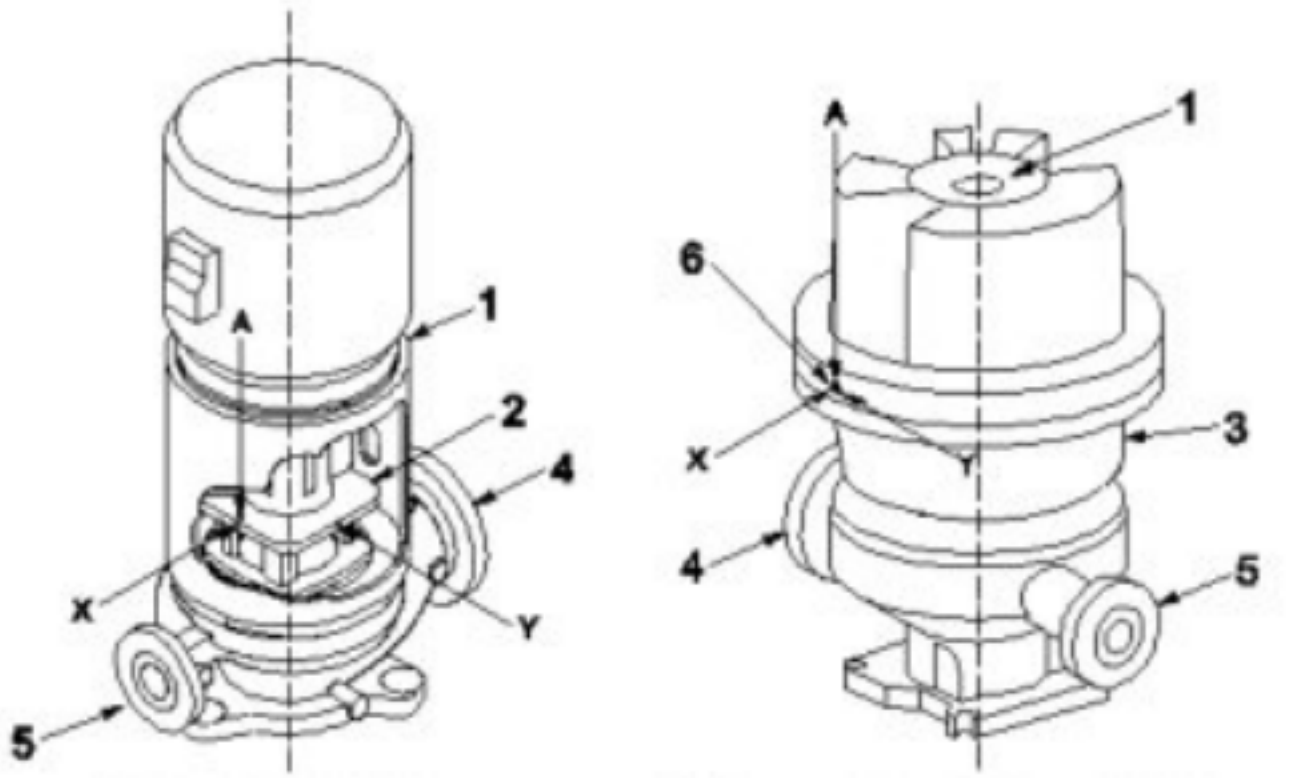
Figure 34—Locations for Taking Vibration Readings on OH and BB Type Pumps



Key

1 driver mounting surface
2 pump bearing housing
3 spot face (see 6.10.2.12)
4 optional arrangement for mounting vibration-measuring equipment (see 6.10.2.13)
A axial

Figure 35—Locations for Taking Vibration Readings on Vertically Suspended (VS) Pumps



a) Vertical In-line (OH3) Pump b) High-speed Integrally Geared (OH6) Pump

Key
1 driver mounting surface
2 pump bearing housing
3 gearbox housing
4 suction flange
5 discharge flange
6 spot face or threaded connection for stud-mounting vibration sensor (6.10.2.12 and 6.10.2.13)
A axial

Figure 36—Locations for Taking Vibration Readings on Vertical In-line (OH3) and High-speed Integrally Geared (OH6) Pumps

6.9.4.10 The vibration measured shall not exceed the values shown in the following:

a) Table 8 for overhung and between-bearing pumps,

b) Table 9 for vertically suspended pumps,

c) Figure 37 for high-energy pumps.

6.9.4.11 Pumps furnished with proximity probes shall meet both bearing-housing and shaft-vibration limits.

NOTE Bearing housing overall vibration limits are defined for RMS measurements only.

Table 8—Vibration Limits for Overhung and Between-bearings Pumps

Criteria	Location of Vibration Measurement	
	Bearing Housing (see Figure 34 and Figure 36)	**Pump Shaft (adjacent to bearing)**
	Pump Bearing Type	
	All	Hydrodynamic journal bearings
	Vibration at Any Flowrate within the Pump's Preferred Operating Region	
Overall	For pumps running at up to 3600 r/min and absorbing up to 400 hp (300 kW) per stage: v_u < 0.12 in./s RMS (3.0 mm/s RMS)	$A_u < (8000/n)^{0.5}$ mils peak-to-peak $(5.2 \times 10^6/n)^{0.5}$ µm peak-to-peak Not to exceed: A_u < 2.0 mils peak-to-peak 50 µm peak-to-peak
Discrete frequencies	For pumps running at up to 3600 r/min and absorbing up to 400 hp (300 kW) per stage: v_f < 0.08 in./s RMS (2.0 mm/s RMS)	For all frequencies except running speed (in Hz): $A_f < 0.33A_u$
Allowable increase in vibration at flows outside the preferred operating region but within the allowable operating region	30 %	30 %

Power calculated for BEP of rated impeller with liquid relative density (specific gravity) equal to 1.0.

Vibration velocity and amplitude values calculated from the basic limits shall be rounded off to two significant figures,

where

v_u is the measured overall velocity;

v_f is the discrete frequency velocity, measured with a FFT spectrum using a Hanning window and a minimum frequency resolution of 1600 lines;

A_u is the amplitude of measured overall displacement;

A_f is the amplitude of displacement at discrete frequencies, measured with a FFT spectrum using a Hanning window and a minimum frequency resolution of 1600 lines;

n is the rotational speed, expressed in revolutions per minute.

For pumps running above 3600 r/min or absorbing more than 400 hp (300 kW) per stage, see Figure 35.

Table 9—Vibration Limits for Vertically Suspended Pumps

Criteria	Location of Vibration Measurement	
	Pump thrust bearing housing or motor mounting flange (see Figure 35)	Pump shaft (adjacent to bearing)
	Pump Bearing Type	
	All	Hydrodynamic guide bearing adjacent to accessible region of shaft
	Vibration at Any Flow Within the Pump's Preferred Operating Region	
Overall	v_u < 0.20 in./s RMS (5.0 mm/s RMS)	$A_u < (10{,}000/n)^{0.5}$ mils peak-to-peak $<(6.2\times10^6/n)^{0.5}$ µm peak-to-peak Not to exceed: A_u < 4.0 mils peak-to-peak (100 µm peak-to-peak)
Discrete frequencies	v_f < 0.13 in./s RMS (3.4 mm/s RMS)	For all frequencies except running speed (in Hz): $A_f < 0.33A_u$
Allowable increase in vibration at flows outside the preferred operating region but within the allowable operating region	30 %	30 %

Vibration velocity and amplitude values calculated from the basic limits shall be rounded off to two significant figures,

where

v_u is the measured overall velocity;

v_f is the discrete frequency velocity, measured with a FFT spectrum using a Hanning window and a minimum frequency resolution of 1600 lines;

A_u is the amplitude of measured overall displacement;

A_f is the amplitude of displacement at discrete frequencies, measured with a FFT spectrum using a Hanning window and a minimum frequency resolution of 1600 lines;

n is the rotational speed, expressed in r/min.

6.9.4.12 The pump and its driver shall perform on their test stands and on their permanent foundation within the vibration acceptance criteria specified in 6.9.4.10. After installation in accordance with the vendor's requirements, the performance of the pump and driver as a combined unit shall be the joint responsibility of the purchaser and the vendor.

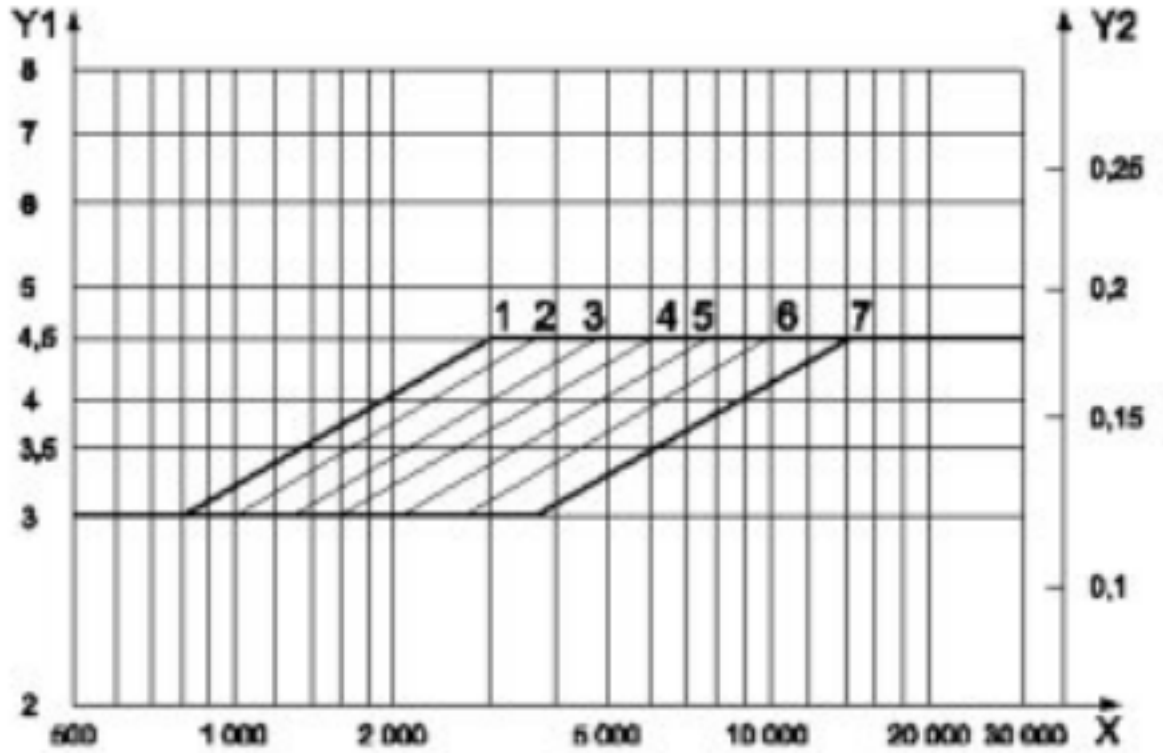
6.9.4.13 At any speed greater than the maximum continuous speed, up to and including the trip speed of the driver, the vibration shall not exceed 150 % of the maximum value recorded at the maximum continuous speed. Vibration exceeding this limit, but below the allowable limit in the table may be accepted with purchaser's approval.

6.9.4.14 Variable-speed pumps shall operate over their specified speed range without exceeding the vibration limits of this standard.

6.9.4.15 If the vendor can demonstrate that electrical or mechanical runout is present, the demonstrated amount of runout can be vectorially subtracted from the measured vibration during the factory test as long as it does not exceed 25 % of the allowed peak-to-peak vibration amplitude or 0.25 mil (6.5 µm), whichever is less.

NOTE 1 The equation for transition from 3.0 mm/s to 4.5 mm/s is: $v_u = 3.0(n/3600)^{0.30}[P/300]^{0.21}$.

NOTE 2 The allowable vibration limit for discrete frequencies is: $v_f < 0.67v_u$ from Table 8 and Table 9.



Key

X rotational speed, n expressed in revolutions per minute
Y1 vibrational velocity, v_u expressed in mm per second, RMS
Y2 vibrational velocity, v_u expressed in inches per second, RMS
1 $P \geq 4000$ HP/stage (3000 kW/stage)
2 $P = 2700$ HP/stage (2000 kW/stage)
3 $P = 2000$ HP/stage (1500 kW/stage)
4 $P = 1300$ HP/stage (1000 kW/stage)
5 $P = 950$ HP/stage (700 kW/stage)
6 $P = 700$ HP/stage (500 kW/stage)
7 $P \leq 400$ HP/stage (300 kW/stage)

NOTE 1 The equation for transition from 3.0 mm/s to 4.5 mm/s is: $v_u = 3.0(n/3600)^{0.30}[P/300]^{0.21}$.

NOTE 2 The allowable vibration limit for discrete frequencies is: $v_i < 0.67 v_u$ from Table 8 and Table 9.

Figure 37—Bearing Housing Vibration Limits for Horizontal Pumps Running Above 3600 r/min or Absorbing More Than 300 hp (225 kW) per Stage

6.10 Bearings and Bearing Housings

6.10.1 Bearings

6.10.1.1 Each shaft shall be supported by two radial bearings and one double-acting axial (thrust) bearing.

6.10.1.2 Bearings, shall be one of the following arrangements:

a) rolling-element radial and thrust;

b) hydrodynamic radial and rolling-element thrust;

c) hydrodynamic radial and thrust.

6.10.1.3 The bearing type and arrangement shall be selected in accordance with the limitations in Table 10. With purchaser approval, antifriction bearings can be used beyond the energy density specified.

6.10.1.3.1 For all bearing types, the bearing manufacturer's published nominal speed limitations shall not be exceeded. For back-to-back 40 degree (0.7 rad) angular contact ball bearings, factor nd_m for individual bearings shall not exceed 500,000 for oil-lubricated and 350,000 for grease-lubricated bearings, where:

D_m is the mean bearing diameter $[(d + D)/2)]$, expressed in millimeters;

N is the rotational speed, expressed in revolutions per minute.

NOTE 1 The bearing temperature limits in 6.10.2.7 can limit nd_m factors to even lower values.

NOTE 2 Roller and spherical bearings generally have lower speed limitations than ball bearings.

6.10.1.3.2 Rolling-element bearing life shall be determined in accordance with 6.10.1.14 or 6.10.1.15.

6.10.1.3.3 For non-pipeline pumps, hydrodynamic radial and thrust bearings shall be used if the energy density [i.e. the product of pump rated power, hp (kW), and rated speed, r/min] is 5.4×10^6 hp/min (4.0×10^6 kW/min) or greater.

6.10.1.3.4 For pipeline pumps, hydrodynamic radial and hydrodynamic thrust bearings shall be used if the energy density is greater than 10.7×10^6 hp/min (8.0×10^6 kW/min).

6.10.1.3.5 For pipeline pumps, with energy density values between 5.4×10^6 hp/min (4.0×10^6 kW/min) and 10.7×10^6 hp/min (8.0×10^6 kW/min), hydrodynamic radial bearings shall be used with either rolling-element or hydrodynamic thrust bearings.

6.10.1.4 For pumps that will be installed in oilfield and pipeline applications, rolling-element radial and thrust bearings may be applied exceeding Table 10 limits up to the manufacturer's favorable experience limits with purchaser's approval.

Table 10—Bearing Selection

Condition	Bearing Type and Arrangement
Radial and thrust bearing speed and life within limits for rolling-element bearings and Pump energy density below limit	Rolling-element radial and thrust
Radial bearing speed or life outside limits for rolling-element bearings and Thrust bearing speed and life within limits and Pump energy density below limit	Hydrodynamic radial and rolling-element thrust
Radial and thrust bearing speed or life outside limits for rolling-element bearings or Pump energy density above limit	Hydrodynamic radial and thrust

6.10.1.5 Thrust bearings shall be sized for continuous operation under all specified conditions, including maximum differential pressure, and comply with the following:

a) All loads shall be determined at design internal clearances and also at twice design internal clearances.

b) Thrust forces for flexible metal-element couplings shall be calculated on the basis of the maximum allowable deflection permitted by the coupling manufacturer.

c) If a sleeve-bearing motor (without a thrust bearing) is directly connected to the pump shaft with a coupling, the coupling-transmitted thrust shall be assumed to be the maximum motor thrust.

d) In addition to thrust from the rotor and any internal gear reactions due to the most extreme allowable conditions, the axial force transmitted through flexible couplings shall be considered a part of the duty of any thrust bearing.

[•] e) If specified, the angular contact thrust bearing may be mounted on a removable sleeve that is retained on the shaft by a locking nut and tab washer. This arrangement allows the thrust bearing to be easily removed for mechanical seal maintenance and then be reused.

f) Hydrodynamic thrust bearings shall be selected such that under any operating condition the load does not exceed 50 % of the bearing manufacturer's ultimate load rating, including twice design internal clearances. Inline diffuser-style BB5 pumps may exceed this criteria with purchaser's approval.

6.10.1.6 Rolling-element bearings shall not have filling slots.

6.10.1.7 Rolling-element bearings shall not have nonmetallic cages.

6.10.1.8 Single-row, deep-groove ball bearings shall have a radial internal clearance in accordance with ABMA 20, Group 3 (larger than "N" or Normal internal clearance). Greater internal clearances can reduce the temperature rise of the lubricant. However, vibration velocities can be increased with greater clearances. The vendor shall ensure that the values for temperature rise and vibration meet the requirements of this standard.

NOTE For the purpose of this provision, ISO 5753, Group 3 is equivalent to ABMA 20, Group 3.

6.10.1.9 Ball thrust bearings shall be of the paired, single-row, 40° (0.7 rad) angular contact type (7000 series) with machined brass cages in a back-to-back arrangement. Pressed steel cages may be used if approved by the purchaser.

6.10.1.10 The need for bearing clearance or preload shall be determined by the vendor to suit the application and meet the bearing life requirements of this standard.

6.10.1.11 If loads exceed the capability of paired, angular-contact bearings as described in 6.10.1.9, or where bearings operate continuously with minimal axial loads, alternative rolling-element arrangements may be proposed.

6.10.1.12 Rolling-element bearing life (basic rating life, L_{10h}, for each bearing or bearing pair) shall be calculated in accordance with ABMA 9 (ISO 281).

NOTE ABMA 9 defines basic rating life, L10, in units of millions of revolutions. Industry practice is to convert this to hours and to refer to it as L_{10h} . For the purpose of this provision, ISO 281 is equivalent to ABMA 9.

6.10.1.13 Bearing system life (the calculated life of the combined system of bearings in the pump) shall be equivalent to at least 25,000 h with continuous operation at rated conditions, and at least 16,000 h at maximum radial and axial loads and rated speed. The system life shall be calculated as given in Equation (3):

$$L_{10h,system} = \left[\left(1/L_{10hA}\right)^{3/2} + \left(1/L_{10hB}\right)^{3/2} + \ldots + \left(1/L_{10hN}\right)^{3/2}\right]^{-2/3} \quad (3)$$

where

L_{10hA} is the basic rating life, L_{10h}, in accordance with ABMA 9 for bearing A;

L_{10hB} is the basic rating life, L_{10h}, in accordance with ABMA 9 for bearing B;

L_{10hN} is the basic rating life, L_{10h}, in accordance with ABMA 9 for bearing *N*;

N is the number of bearings.

[•] **6.10.1.14** If specified, the bearing system life calculations shall be furnished. Refer to K.2 for a discussion of bearing system life.

NOTE A bearing system L_{10h} life of 25,000 h and 16,000 h indicates that the L_{10h} life of each individual bearing is significantly higher. Further to meet these requirements, two bearings would each have to have an L_{10h} of ~40,000 h.

6.10.1.15 Rolling-element bearings shall be located, retained, and mounted in accordance with the following.

a) Bearings shall be retained on the shaft with an interference fit and fitted into the housing with a diametral clearance, both in accordance with ABMA 7.

b) Bearings shall be mounted directly on the shaft, if not otherwise specified [see 6.10.1.5 e)].

c) Bearings shall be located on the shaft using shoulders, collars or other positive locating devices. Snap rings and spring-type washers are not acceptable.

d) The device used to lock thrust bearings to shafts shall be restricted to a nut with a tongue-type lock washer.

e) For certain roller bearings, such as cylindrical roller types with separable races, bearing-to-housing diametral clearances in accordance with ABMA 7 may not be appropriate.

6.10.2 Bearing Housings

6.10.2.1 Bearing housings shall be arranged so that bearings can be replaced without disturbing pump drives or mountings.

[•] **6.10.2.2** If specified, bearing housings shall have provisions to measure bearing temperatures.

6.10.2.3 Bearing housings for oil-lubricated non-pressure-fed bearings shall be provided with the following features:

a) threaded and plugged fill and drain connections of at least NPS $^1/_2$ (DN 15);

b) means, such as a bulls-eye or sight glass, for detecting overfilling of the housings shall be located such that the proper oil level elevation is at the midpoint (at 50 % up or down the viewing area);

c) a permanent indication of the proper oil level shall be accurately located and clearly marked on the outside of the bearing housing with permanent metal tags, marks inscribed in the castings, or other durable means;

d) a bearing housing vent to atmosphere, which may be located at the fill connection;

e) a “vented to atmosphere” constant-level sight feed oilers at least 4 fluid oz (1.2 dl) in volume, heat-resistant glass containers and protective wire cages.

NOTE VS pumps can have an alternate arrangement for accessibility or sump volume.

[•] **6.10.2.4** If specified, bearing housings for oil-lubricated non-pressure-fed bearings shall be provided with a “vented to bearing housing” constant level oiler in lieu of a “vented to atmosphere” constant level oiler.

[•] **6.10.2.5** If specified, bearing housings for oil-lubricated non-pressure-fed bearings shall be provided with an oil sump collection container.

6.10.2.5.1 The container shall be transparent and shall be located on the bottom of the oil sump to collect bearing housing contaminants such as water.

6.10.2.5.2 The container shall be fitted with a spring-loaded drain pet cock.

6.10.2.5.3 The container materials of construction shall be suitable for the lubricant used and the minimum ambient temperature of the pump location.

NOTE If ambient air temperature is below 32 °F (0 °C), residual water in the collection site glass can cause it to rupture. This is taken into consideration for installations that can be in cold weather climates

6.10.2.6 Bearing housings for pressure-lubricated hydrodynamic bearings shall be arranged to minimize foaming. The drain system shall be adequate to maintain the oil and foam level below shaft end seals.

6.10.2.7 Enough cooling, including an allowance for fouling, shall be provided to maintain oil and bearing temperatures as follows during shop testing and in field operation under the most adverse specified operating condition:

a) for pressurized circulating oil systems, the bearing-oil temperature rise shall not exceed 50 °F (28 K), and if bearing-temperature sensors are supplied, bearing metal temperatures shall not exceed 200 °F (93 °C);

b) for non-pressurized and oil mist systems, the bearing housing surface temperature shall not exceed 160 °F (70 °C), and if bearing-temperature sensors are supplied, bearing metal temperatures shall not exceed 200 °F (93 °C).

NOTE Solar radiation can cause bearing housing temperature to be higher than specified above.

6.10.2.8 If cooling with water or product is required, tube-type cooling should be provided.

6.10.2.8.1 The tubing (including fittings) shall be austenitic stainless steel and shall have no internal pressure joints. Alternate metallurgy may be required for some services, such as high-chloride content cooling media, with purchaser's approval.

6.10.2.8.2 Tubing or pipe shall have a minimum thickness of 0.040 in. (1.0 mm) and shall be at least 0.50 in. (12 mm) OD.

6.10.2.9 Bearing housings for rolling-element bearings shall be designed to prevent contamination by moisture, dust, and other foreign matter. This shall be achieved without the requirement for external service, e.g. air purge.

6.10.2.9.1 Bearing housings shall be equipped with replaceable labyrinth-type or magnetic-type end seals and deflectors where the shaft passes through the housing.

6.10.2.9.2 Lip-type seals shall not be used.

6.10.2.9.3 Seals and deflectors shall be made of spark-resistant materials.

NOTE Many users consider pure aluminum and aluminum alloys with a maximum content of 2 % magnesium or 0.2 % copper, all copper, and copper-based alloys (e.g. brass, bronze) to be spark-resistant. However, local standards, such as ISO 80079-36, might not allow aluminum or nonmetallic materials within potentially explosive atmospheres.

6.10.2.9.4 The design of the seals and deflectors shall effectively retain oil in the housing and prevent entry of foreign material into the housing while running and in standstill conditions.

6.10.2.10 If oil-mist lubrication is specified, the requirements of 6.10.2.10.1 or 6.10.2.10.2 shall apply.

6.10.2.10.1 For pure oil-mist lubrication, bearings and bearing housings shall meet the following requirements:

a) a threaded NPS $^{1}/_{4}$ (DN 8) oil-mist inlet connection shall be provided on the housing or end cover for each of the spaces between the rolling-element bearing or bearing set and the bearing housing end seal;

b) oil-mist fitting connections shall be located so that oil mist can flow through rolling-element bearings;

c) oil rings or flingers and constant-level oilers shall not be provided, and a mark indicating the oil level is not required;

d) drain-back and any other (feed hole) oil passages in the bearing housing shall be plugged to prevent the oil mist from bypassing the bearings(s);

e) water cooling systems shall not be provided.

f) Permanent markings at each connection location (cast, stamped, or stamped SS tag affixed with bolts or screws) shall be used to identify all oil mist IN, VENT, and DRAIN connections.

6.10.2.10.2 For purge oil-mist lubrication, bearings and bearing housings shall meet the following requirements.

a) A threaded NPS $^1/_4$ or NPS $^1/_2$ (DN 8 or DN 15) oil-mist connection shall be located in the top half of the bearing housing to act also as a vent-and-fill connection.

b) Constant-level oilers shall be provided, and a mark indicating the oil level shall be provided on the bearing housing. Bearing lubrication is by a conventional oil bath, flinger, or oil ring system.

c) Constant-level sight feed oilers shall be equipped with overflow control to allow excess coalesced oil from the mist system to drain from the bearing housing so that oil level in the sump is maintained at proper level. The oil shall be contained to prevent it from draining onto the baseplate.

d) Constant-level sight feed oilers shall be piped so that they operate at the internal pressure of the bearing housing, do not vent excess mist at the bearing housing, or allow oil to drip to the baseplate.

6.10.2.10.3 For both pure and purge mist applications, a drain connection shall be located on the bottom of the bearing housing to provide complete oil drainage (see 6.10.2.10.5).

6.10.2.10.4 Shielded or sealed bearings shall not be used in conjunction with either pure or purge oil-mist systems.

6.10.2.10.5 The oil-mist supply, reclassifier and drain fittings shall be provided by the purchaser (refer to API 686).

6.10.2.10.6 Directional reclassifiers, if required, shall be provided by the pump manufacturer.

6.10.2.11 Housings for ring oil-lubricated bearings shall be provided with (plugged) ports positioned to allow visual inspection of the oil rings while the pump is running.

6.10.2.12 All bearing housings shall be spot faced at the locations shown on Figure 34, Figure 35, and Figure 36 to facilitate consistent vibration measurements. It shall be 1.125 in. (29 mm) diameter spot face, with a surface finish of 32 microinches (0.8 µm) R_a or better and surface flatness below 1 mil (25 µm).

[•] **6.10.2.13** If specified, bearing housings shall have a threaded connection(s) for permanently mounting vibration transducers in accordance with API 670. Refer to Figure 34, Figure 35, and Figure 36.

6.10.2.14 On pumps without a 360 degree bearing housing mount, the bearing housings shall be doweled by the vendor after verification of seal chamber runout.

6.11 Lubrication

6.11.1 Bearings and bearing housings shall be designed for oil lubrication using a mineral (hydrocarbon) oil.

[•] **6.11.2** The purchaser shall specify whether synthetic oil shall be used.

6.11.3 The operation and maintenance manual shall describe how the lubrication system circulates oil.

[•] **6.11.4** If specified, provisions shall be made for either pure oil-mist or purge oil-mist lubrication (see 6.10.2.10.1 and 6.10.2.10.2 for requirements).

[•] **6.11.5** If specified, rolling-element bearings shall be grease-lubricated in accordance with the following:

a) grease life (re-lubrication interval) shall be estimated using the method recommended by the bearing manufacturer or an alternative method approved by the purchaser;

b) grease lubrication shall not be used if the estimated grease life is less than 2000 h;

c) if the estimated grease life is 2000 h or greater but less than 25,000 h, provision shall be made for re-greasing the bearings in service and for the effective discharge of old or excess grease, and the vendor shall advise the purchaser of the required re-greasing interval;

d) if the estimated grease life is 25,000 h or more, grease nipples or any other system for the addition of grease in service shall not be fitted.

6.11.6 The vendor shall ensure that bearing-housing internal paint, if used, is compatible with the specified oil.

6.12 Materials

6.12.1 General

[•] **6.12.1.1** The purchaser shall specify the material class for pump parts from Annex H, Table H.1.

6.12.1.1.1 Annex G, Table G.1 provides a guide for material classes that may be appropriate for various services.

6.12.1.1.2 Pump materials shall be in accordance with Table H.1.

6.12.1.1.3 Alternative materials, including materials that can improve life and performance in service, may also be included in the vendor's proposal and listed on the data sheets.

[•] **6.12.1.2** If specified, wear parts of nonmetallic materials from Table H.3 shall be provided.

6.12.1.3 If nonmetallic wear parts are provided, the material class from Table H.1 shall be modified by addition of the letter C, such as S-5C. The nonmetallic materials shall be agreed by the vendor and purchaser.

6.12.1.4 The material specification of all components listed in Table H.1 shall be clearly stated in the vendor's proposal.

6.12.1.4.1 Materials shall be identified by reference to applicable international standards, including the material grade (Table H.2 and Table H.3 may be used for guidance).

6.12.1.4.2 If standard materials are not available, then national standards may be used. If no such designations are available, the vendor's material specification, giving physical properties, chemical composition, and test requirements, shall be included in the proposal.

6.12.1.5 The material specification of all gaskets and O-rings exposed to the pumped liquid shall be identified in the proposal. O-rings shall be selected and their application limited in accordance with API 682, Annex B.

6.12.1.6 Pump parts having strength or pressure-integrity requirements are designated as "full compliance" materials in Table H.1 and shall meet all the requirements of the agreed specifications. For any other part (e.g. if corrosion resistance is the primary concern), it is necessary to comply only with the specified chemical composition. Auxiliary piping materials are covered in 7.6.

[•] **6.12.1.7** Purchasers shall specify optional test and inspection requirements that are necessary to ensure that materials are satisfactory for the service.

6.12.1.8 If austenitic stainless steel parts exposed to conditions that can promote intergranular corrosion are fabricated, hard-faced, overlaid, or repaired by welding, they shall be made of low-carbon or stabilized grades.

NOTE Overlays or hard surfaces that contain more than 0.10 % carbon can sensitize both low-carbon and stabilized grades of austenitic stainless steel unless a buffer layer that is not sensitive to intergranular corrosion is applied.

[•] **6.12.1.9** If specified, the vendor shall furnish material certificates that include chemical analysis and mechanical properties for the heats from which the material is supplied for pressure-containing castings and forgings, impellers, and shafts. Piping nipples, auxiliary piping components, gaskets, and bolting may be included in this requirement, as specified by the purchaser.

[•] **6.12.1.10** The purchaser shall specify any erosive or corrosive agents (including trace quantities) present in the process liquids and in the site environment, including constituents that can cause stress-corrosion cracking or attack elastomers.

[•] **6.12.1.11** If specified, coatings of a type agreed between the purchaser and the vendor shall be applied to impellers and other wetted parts.

6.12.1.12 The sequence of procedures and acceptance criteria for balancing and coating of rotating components shall be agreed between the purchaser and the vendor.

6.12.1.13 If mating parts, such as studs and nuts of austenitic stainless steel or materials with similar galling tendencies, are used, they shall be lubricated with an anti-seizure compound compatible with the material(s) and specified process liquid(s) operating conditions.

NOTE The torque loading values required to achieve the necessary preload can vary considerably depending upon the thread lubricant (see 6.4.3.7.4).

[•] **6.12.1.14** The purchaser shall specify the amount of wet H_2S that can be present in the process, considering normal operation, start-up, shutdown, idle standby, upsets, or unusual operating conditions such as catalyst regeneration.

[•] **6.12.1.15** The purchaser shall specify if reduced-hardness materials are required. Reduced-hardness materials shall be supplied in accordance with NACE MR0103. NACE MR0103 applies to oil refineries, liquefied natural gas (LNG) plants, and chemical plants. NACE MR0103 applies to materials potentially subject to sulfide stress-corrosion cracking.

[•] **6.12.1.16** If specified, reduced-hardness materials shall be supplied in accordance with NACE MR0175 in lieu of NACE MR0103. NACE MR0175 (all parts), applies to material potentially subject to sulfide and chloride stress-corrosion cracking in oil and gas production facilities and natural gas sweetening plants.

NOTE For the purposes of this provision, ISO 15156–1 is equivalent to NACE MR0175.

6.12.1.16.1 Reduced-hardness ferrous materials not covered by NACE MR0103 or NACE MR0175 shall have yield strength not greater than 90,000 psi (620 N/mm^2) and hardness not greater than HRC 22.

6.12.1.16.2 Components that are fabricated by welding shall be postweld heat treated, if required, so that both the welds and heat-affected zones meet the yield strength and hardness requirements.

6.12.1.16.3 Reduced hardness materials shall be provided for the following components, as a minimum:

a) pressure casing,

b) shafting (including wetted shaft nuts),

c) pressure-retaining mechanical seal components (excluding the seal ring and mating ring),

d) wetted bolting,

e) bowls.

NOTE 1 Double-casing-pump inner-casing parts that are in compression, such as diffusers, are not considered pressure casing parts. Casing bolting for pumps with axially split casings (such as BB1 and BB3) are not considered to be wetted.

6.12.1.16.4 Renewable impeller wear rings that are through-hardened shall not have hardness greater than HRC 22, if reduced-hardness materials are specified.

6.12.1.16.5 Renewable impeller wear rings that are either hard-coated or surface-hardened may have hardness greater than HRC 22 if their substrate material hardness is not greater than HRC 22.

6.12.1.16.6 If approved by the purchaser, integral wear surfaces on the impellers may be surface-hardened or hardened by the application of a suitable coating.

6.12.1.17 Only fully killed, normalized steels made to fine-grain practice shall be used.

6.12.1.18 The vendor shall select materials to avoid conditions that can result in galvanic corrosion. Where such conditions cannot be avoided, the purchaser and the vendor shall agree on the material selection and any other precautions necessary.

6.12.1.19 Bearing housings, load-carrying bearing housing covers, and brackets between the pump casings or heads and the bearing housings shall be steel.

6.12.1.20 Driver supports for vertical pumps that use thrust bearings in the driver to support the shaft shall be steel.

6.12.1.21 Any part that is manufactured by Additive Manufacturing ("3D Printing") shall be identified by the manufacturer, prior to fabrication.

6.12.1.21.1 The vendor shall provide a listing of the technical information in accordance with API 20S, 1st Edition, Section 4.2.

6.12.1.21.2 Feedstock shall meet the requirements of Additive Manufacturing Specification Level (AMSL) 3, per API 20S.

6.12.2 Castings

6.12.2.1 Surfaces of castings shall be cleaned by sandblasting, shot blasting, chemical cleaning, or any other standard method to meet the visual requirements of MSS SP-55. Mold-parting fins and remains of gates and risers shall be chipped, filed, or ground flush.

6.12.2.2 The use of chaplets in pressure castings shall be held to a minimum.

6.12.2.2.1 Chaplets shall be clean and corrosion-free (plating is permitted) and of a composition compatible with the casting.

6.12.2.2.2 Chaplets shall not be used in impeller castings.

6.12.2.3 Ferrous pressure boundary and impeller castings shall not be repaired by peening, plugging, burning in, or impregnating.

6.12.2.4 Weldable grades of steel castings may be repaired by welding in accordance with 6.12.3. Weld repairs shall be inspected according to the same quality standard used to inspect the casting.

6.12.2.5 Fully enclosed, cored voids that become fully enclosed by methods such as plugging, welding, or assembly shall not be used.

6.12.2.6 Casting repairs shall be performed in accordance with a weld repair procedure compliant with the component material specification and with the requirements of Table 11.

[•] **6.12.2.7** If specified, for major repairs as defined in 6.12.2.11, repair procedures and weld maps shall be subject to purchaser's acceptance prior to any repair commencing.

6.12.2.8 Castings shall be heat-treated following major weld repairs in accordance with the applicable material standard specification.

6.12.2.9 With the exception of the requirements of 6.12.2.11.1 through 6.12.2.11.4, casting repairs made at the foundry level shall be controlled by the casting material specification ("producing specification").

6.12.2.10 Pressure-containing castings of carbon steel shall be furnished in the normalized and tempered, or quenched and tempered condition.

6.12.2.11 A weld repair that involves any of the following at the foundry level or at the Supplier's shop shall be considered a major weld repair.

6.12.2.11.1 The depth of the cavity prepared for repair welding on a pressure containing casting under repair exceeds 20 % of the component wall thickness or 1 in. (25mm), whichever is smaller.

6.12.2.11.2 The total length of the cavity prepared for repair welding is greater than 6 in. (150mm).

6.12.2.11.3 The total area of all repairs to a pressure containing casting under repair exceeds 10 % of the surface area of the pressure containing casting or 10 in^2 (65 cm^2) whichever is smaller.

6.12.2.11.4 Closure of any through-wall cavity or opening, for any reason, by any means, on a pressure containing casting, that meets class II or class III Inspection criteria (per Table 14).

[•] **6.12.2.11.5** If specified, closure of any through-wall cavity or opening, for any reason, by any means, on a pressure containing casting.

6.12.2.11.6 A casting that leaks during hydrostatic testing.

[•] **6.12.2.11.7** If specified, a welding repair performed on impeller castings.

6.12.2.12 The following shall be provided for purchaser's approval for major repairs at the foundry level or at the Supplier's shop.

6.12.2.12.1 Sketches or photographs showing the affected area.

6.12.2.12.2 Method of repair, including WPS, PQR and non-destructive examination requirements.

6.12.2.12.3 Extent of testing or retesting to prove the effectiveness of the repaired casting, if applicable.

6.12.2.13 Major repairs or welds at the foundry level or at the Supplier's shop shall conform to material and welding requirements of the applicable material specification.

6.12.3 Welding

6.12.3.1 Welding and weld repairs shall be performed by operators and in accordance with procedures qualified to the requirements of Table 11. Alternative standards may be proposed by the vendor for the purchaser's approval.

6.12.3.2 The vendor shall be responsible for the review of all repairs and repair welds to ensure they are properly heat-treated and nondestructively examined for soundness and compliance with the applicable qualified procedures (see 6.12.3.1 and 8.2.1.6, Table 14).

6.12.3.3 Pressure-containing casings made of wrought materials or combinations of wrought and cast materials shall conform to the conditions specified in Items a) through d) as follows (these requirements do not apply to casing nozzles and auxiliary connections; see 6.12.3.4):

a) accessible surfaces of welds shall be inspected by magnetic-particle or liquid-penetrant examination after back chipping or gouging and again after PWHT or, for austenitic stainless steels, after solution annealing;

b) pressure-containing welds, including welds of the casing to axial-joint and radial-joint flanges, shall be full-penetration welds;

c) if dimensional stability of such a casing component is required for the integrity of pump operation, then PWHT shall be performed regardless of thickness;

d) plate edges shall be inspected by magnetic-particle or liquid-penetrant examination as required by recognized standards, such as ASME BPVC Section VIII, Division 1, UG-93 (d)(3).

Table 11—Welding Requirements

Requirement	Applicable Code or Standard
Welder/operator qualification	ASME BPVC Section IX or ISO 9606 for welder qualification or ISO14732 for welder operator qualification
Welding procedure qualification	Applicable material specification or, where weld procedures are not covered by the material specification, ASME *BPVC* Section IX, ASME B31.3, ISO 15607, or ISO 15609 (all parts)
Non-pressure-retaining structural welding, such as baseplates or supports	AWS D1.1/D1.1M
Magnetic-particle or liquid-penetrant examination of the plate edges	ASME *BPVC* Section VIII, Division 1, UG-93(d)(34)
Postweld heat treatment	Applicable material specification, ASME *BPVC* Section VIII, Division 1, UW 40, ASME B31.3, or EN 13445–4
Postweld heat treatment of casing fabrication welds	Applicable material specification, ASME *BPVC* Section VIII, Division 1 or EN 13445–4
NOTE For the purpose of this provision, ISO 10721–2 is equivalent to AWS D1.1/D1.1M.	

6.12.3.4 Connections welded to pressure casings shall be installed as specified in Items a) through d):

a) attachment of suction and discharge nozzles shall be by means of full-fusion, full-penetration welds using welding neck flanges and shall not be dissimilar metal weldments;

b) auxiliary piping welded to alloy steel casings shall be of a material with the same nominal properties as the casing material (see 6.4.2.12);

c) PWHT, if required, shall be performed after all welds, including piping welds, have been completed;

d) suction and discharge nozzle welds shall be inspected by magnetic-particle or liquid-penetrant examination after back chipping or gouging and again after PWHT or, for austenitic stainless steels, after solution annealing.

[•] **6.12.3.4.1** The purchaser shall specify if the following additional examinations shall be performed on connections welded to pressure casings (if not already required in 8.2.1.6, Table 14):

a) magnetic-particle or liquid-penetrant examination of auxiliary connection welds;

b) ultrasonic or radiographic examination of any casing welds.

[•] **6.12.3.4.2** The purchaser shall specify if the proposed connection design shall be submitted for approval before fabrication. The submittal drawing shall show weld designs, sizes, materials, and pre-weld and postweld heat treatments.

6.12.4 Low-temperature Service

[•] **6.12.4.1** The purchaser shall specify the minimum design metal temperature to which the pump will be subjected in service. This temperature shall be used to establish impact test requirements. Normally, this is the lower of the minimum surrounding ambient temperature or minimum liquid pumping temperature. However, the purchaser may specify a minimum design metal temperature based on pump liquid properties, such as auto-refrigeration at reduced pressures.

6.12.4.2 To avoid brittle failures, materials of construction for low-temperature service shall be suitable for the minimum design metal temperature in accordance with the codes and other requirements specified. The purchaser and the vendor shall agree on any special precautions necessary, with regard to conditions that can occur during operation, maintenance, transportation, erection, commissioning, and testing.

[•] **6.12.4.3** The purchaser shall specify whether ASME BPVC, Section VIII Division 1 or EN 13445 (all parts) shall apply with regard to impact-testing requirements.

6.12.4.4 The governing thickness used to determine impact-testing requirements shall be the greater of the following:

a) nominal thickness of the largest butt-welded joint;

b) largest nominal section for pressure containment, excluding:

   1) structural support sections, such as feet or lugs,

   2) sections with increased thickness required for rigidity to mitigate shaft deflection,

   3) structural sections required for attachment or inclusion of mechanical features such as jackets or seal chambers;

c) one-fourth of the nominal flange thickness, including parting flange thickness for axially split casings (in recognition that the predominant flange stress is not a membrane stress).

6.12.4.5 If ASME BPVC Section VIII, Division 1 is specified (see 6.12.4.3), the following shall apply:

a) all pressure-retaining steels applied at a specified minimum design metal temperature below −20 °F (−29 °C) shall have a Charpy V-notch impact test of the base metal and the weld joint unless they are exempt in accordance with ASME *BPVC* Section VIII, Division 1, UHA-51;

b) carbon steel and low-alloy steel pressure-retaining parts applied at a specified minimum design metal temperature between −20 °F (−29 °C) and 100 °F (38 °C) shall require impact testing as follows:

   1) impact testing is not required for parts with a governing thickness of 1 in. (25 mm) or less;

   2) impact testing exemptions for parts with a governing thickness greater than 1 in. (25 mm) shall be established in accordance with ASME *BPVC* Section VIII, Division 1, UCS-66.

   3) Minimum design metal temperature without impact testing may be reduced as shown in Figure UCS-66.1.

4) If the material is not exempt, Charpy V-notch impact test results shall meet the minimum impact energy requirements of ASME *BPVC* Section VIII, Division 1, UG-84.

6.13 Nameplates and Rotation Arrows

6.13.1 A nameplate shall be securely attached at a readily visible location on the equipment and on any other major piece of auxiliary equipment. Riveted or screwed-on nameplates shall be affixed to a low-stress area or away from the pressure boundary.

6.13.2 The nameplate shall be stamped with the following information, in units consistent with the purchase order:

a) purchaser's item number,

b) vendor's size and model number,

c) pump serial number,

d) rated flow,

e) rated head,

f) casing hydrostatic test pressure,

g) speed,

h) manufacturer's bearing identification numbers with prefix and suffix (if applicable),

i) MAWP,

j) temperature basis for MAWP,

k) purchaser's purchase order number,

l) bare shaft pump weight.

m) year of manufacture

n) shaft lift (for vertical pumps)

6.13.3 In addition to being stamped on the nameplate, the pump serial number shall be plainly and permanently marked on the pump casing.

6.13.4 Rotation arrows shall be cast in or attached to each major item of rotating equipment at a readily visible location.

6.13.5 Nameplates and rotation arrows (if attached) shall be of austenitic stainless steel or of nickel-copper alloy UNS N04400 [equivalent to Monel [13]].

6.13.6 Attachment pins shall be of the same material as the nameplate or rotation arrow. Welding is not permitted as an attachment method for nameplates or rotation arrows.

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[13] Monel™ is an example of a suitable product available commercially. This information is given for the convenience of users of this standard and does not constitute an endorsement by API of this product.

7 Accessories

7.1 Drivers

7.1.1 The driver shall be sized to meet the maximum specified operating conditions, including bearing, mechanical seal, external gear, and coupling losses, as applicable, and shall be in accordance with the applicable specifications, as stated in the inquiry specification, data sheets, and order.

7.1.2 The driver shall be suitable for satisfactory operation under the utility and site conditions specified.

7.1.3 The driver shall be sized to accommodate specified process variations such as changes in pressure, temperature, or properties of the liquid handled, as well as specified special plant start-up conditions.

7.1.4 For drive-train components that have a mass greater than 500 lb (225 kg), the equipment feet shall be provided with vertical jackscrews.

7.1.5 Electric motor drives shall conform to guidelines of 7.1.5.1 through 7.1.5.4, or other standard as approved by the purchaser.

7.1.5.1 Low-voltage induction motors shall be in accordance with IEEE 841 [up to 500 hp (370 kW)].

7.1.5.2 General-purpose medium-voltage induction motors shall be in accordance with API 547 [250 hp (186 kW) and larger].

7.1.5.3 Special-purpose medium- and high-voltage induction motors shall be in accordance with API 541 [500 hp (373 kW) and larger].

[•] **7.1.5.4** The purchaser shall specify if IEC motors are required.

NOTE API 541 and API 547 are applicable to either NEMA or IEC and ISO, as specified. IEEE 841 is available as NEMA only, as there is no equivalent IEC or ISO standard to IEEE 841 at this time. The purchaser and vendor agree on the requirements in this case.

7.1.6 Motors shall have nameplate power ratings, excluding the service factor (if any), of the power at the pump rated conditions (bhp/bkW), inclusive of the multiplier given in Table 12.

[•] **7.1.7** If specified, the motor nameplate power rating shall be equal to the power required at end of the pump curve or per Table 12, whichever is greater.

7.1.8 The smallest acceptable motor power rating to be supplied is 5 hp (4 kW). If it appears that this procedure leads to oversizing of the motor, an alternative proposal shall be submitted for the purchaser's approval.

7.1.9 The motor power rating shall be rounded up to the next standard motor nameplate size after the multiplier in Table 12 is applied.

Table 12—Power Ratings for Motor Drives

Rated Pump Power bhp	Rated Pump Power bkW	Multiplier Overrated Pump Power
< 30	< 22	1.25
30 to 75	22 to 55	1.15
> 75	> 55	1.10

[•] **7.1.10** The purchaser shall specify the type of motor, its characteristics, and the accessories, including the following:

a) electrical characteristics;

b) starting conditions (including the expected voltage drop on starting);

c) type of enclosure;

d) SPL;

e) area classification, based on IEC 60079 or API 500;

f) type of insulation;

g) required service factor;

h) ambient temperature and elevation above sea level;

i) transmission losses;

j) temperature detectors, vibration sensors and heaters, if these are required;

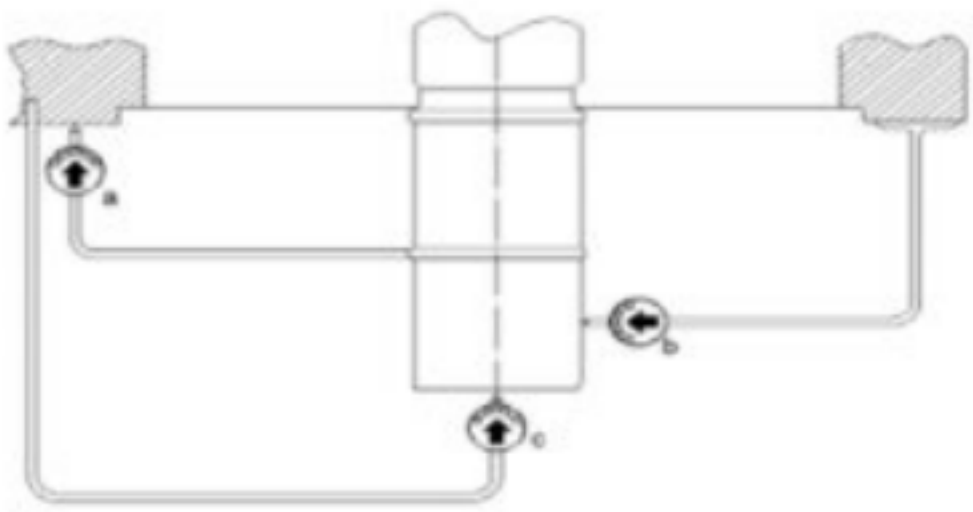
k) vibration acceptance criteria;

l) applicability of IEC 60034-1, IEC 60034-2-1, API 541, API 547, or IEEE 841.

7.1.11 The motor shall be capable of accelerating the pump to rated speed at 80 % voltage against a closed discharge valve.

7.1.12 Motors for vertical-axis pumps shall meet the shaft and base tolerances shown in Figure 38.

All measurements shall be taken with the assembled driver in the vertical position.



a	Shaft-to-driver mating face perpendicularity and surface flatness.	0.002 in./ft (0.17 mm/m)
b	Maximum shaft runout with rotor rotating freely.	0.001 in. (25 µm) TIR
c	Maximum axial float.	0.005 in. (125 µm) TIR

All measurements shall be taken with the assembled driver in the vertical position.

Figure 38—Vertical-Axis Pump Drivers—Tolerances Required for the Driver Shaft and Base

7.1.13 Bearings in the drive systems designed for radial or axial loads transmitted from the pump shall meet the following requirements.

a) Rolling-element bearings shall be selected to give a basic rating life, in accordance with ISO 281, equivalent to at least 25,000 h with continuous operation at pump rated conditions.

b) Rolling-element bearings shall be selected to give a basic rating life equivalent to at least 16,000 h when carrying the maximum loads (radial or axial or both) imposed with internal pump clearances at twice the design values and when operating at any point between minimum continuous stable flow and rated flow.

c) Vertical motors of 1000 hp (750 kW) and larger that are equipped with spherical or taper roller bearings may have less than 16,000 h life at worst conditions to avoid skidding in normal operation. In such cases, the vendor shall state the shorter design life in the proposal.

d) For vertical motors and right-angle gears, the thrust bearing shall be in the nondrive end and shall limit axial float to 0.005 in. (125 µm).

e) Single-row, deep-groove ball bearings shall have radial internal clearance in accordance with ISO 5753, Group 3 [larger than "N" (Normal) internal clearance]. Single- or double-row bearings shall not have filling slots.

f) Thrust bearings shall be designed to carry the maximum thrust that the pump can develop while starting, stopping, or operating at any flowrate.

g) Hydrodynamic thrust bearings shall be selected at no more than 50 % of the bearing manufacturer's ultimate load rating at twice the pump internal clearances specified in 6.7.7.

7.1.14 Steam turbine drivers shall conform to API 611. Steam turbine drivers shall be sized to deliver continuously 110 % of the pump rated power at normal steam conditions.

[•] **7.1.15** If specified, API 612 shall be used for special purpose steam turbine applications.

7.1.16 Gears shall conform to API 677.

[•] **7.1.17** If specified, API 613 shall be used for special purpose gears.

7.1.18 Motors shall be tested bolted on a rigid mounting surface, per Section 7.6.3 of NEMA MG-1 (2021).

7.2 Couplings

7.2.1 Couplings between drivers and driven equipment shall be supplied and mounted by the vendor with unit responsibility. Coupling spacers may be shipped loose for installation at the site.

7.2.2 Couplings shall be all-metal, flexible disc, spacer-type manufactured in accordance with AGMA 9000, class 9. Additionally, couplings shall comply with the following:

a) Flexible elements shall be non-lubricated metal type of corrosion-resistant material.

b) Couplings shall be designed to positively retain the spacer if a flexible element ruptures. The use of bolt heads or flexible element fasteners alone to retain the spacer if a flexible membrane ruptures may not provide adequate support because they are subject to wear if and when failure occurs.

c) Coupling hubs shall be steel.

d) The distance between the pump and driver shaft ends (distance between shaft ends, or DBSE) shall be greater than the longest component that will be removed between the shaft ends (such as coupling hub length, mechanical seal and drive collar total length), without disturbing the driver, driver coupling hub, pump coupling hub, or the suction and discharge piping. For OH Pumps, the DBSE shall be at least 5 in. (125 mm).

e) Provision shall be made for the attachment of alignment equipment without the requirement to remove the spacer or dismantle the coupling in any way.

f) Couplings operating at speeds in excess of 3800 r/min shall meet the requirements of API 671 for component balancing and assembly balance check.

[•] g) If specified, major coupling components (hubs, spacer/flex element, and weight-matched hardware) shall be balanced in accordance with ISO 21940–11, to the balance grade specified by the purchaser.

7.2.3 Couplings and coupling to shaft junctures shall be rated for the driver name plate power.

7.2.4 The coupling service factor shall be a minimum of 1.5.

[•] **7.2.5** If specified, couplings shall meet the requirements of API 671, ISO 14691, or ISO 10441.

7.2.6 Information on shafts, keyway dimensions (if any), and shaft end movements due to end play and thermal effects shall be furnished to the vendor supplying the coupling.

7.2.7 Flexible couplings shall be keyed to the shaft.

7.2.7.1 Keys, keyways, and fits shall conform to AGMA 9002, Commercial class.

7.2.7.2 Shaft coupling keyways shall be cut to accommodate a rectangular cross section key.

7.2.7.3 Sled-runner type keys and keyways shall not be provided.

7.2.7.4 Keys shall be fabricated and fitted to minimize unbalance.

7.2.8 For shaft diameters greater than 2.5 in. (60 mm) and if it is necessary to remove the coupling hub to service the mechanical seal, the hub shall be mounted with a taper fit.

7.2.8.1 The coupling fit taper for keyed couplings shall be 1 in 16 (0.75 in./ft, 60 mm/m diametral).

7.2.8.2 Required amount of pull up for tapered fit couplings shall be supplied.

7.2.8.3 Other mounting methods and tapers shall be agreed upon by the purchaser and the vendor.

7.2.9 Coupling hubs with cylindrical bores may be supplied with slip fits to the shaft and set screws that bear on the key.

7.2.10 Coupling hubs designed for interference fits to the shaft shall be furnished with tapped puller holes at least 0.38 in. (10 mm) in diameter to facilitate removal.

[•] **7.2.11** If specified, couplings shall be fitted hydraulically.

[•] **7.2.12** If specified, couplings shall be fitted with a proprietary clamping device. Acceptable clamping devices may include tapered bushings, frictional locking assemblies, and shrink discs. The vendor responsible for the final machining of the hub bores shall select a suitable rating/size device to suit the coupling and the application.

7.2.13 If the vendor is not required to mount the driver, the fully machined half-coupling shall be delivered to the driver manufacturer's plant or any other designated location, together with the necessary instructions for mounting the half-coupling on the driver shaft.

7.3 Guards

7.3.1 Guards between drivers and driven equipment and between the bearing housing and seal gland shall be supplied and mounted by the vendor with unit responsibility.

7.3.2 Each coupling shall have a coupling guard that is removable without disturbing the coupled elements. Each coupling guard shall meet the following requirements:

a) enclose the coupling and the shafts to prevent personnel from contacting moving parts during operation of the equipment train; allowable access dimensions shall conform with ISO 14120 or ANSI B11.19;

b) be constructed with enough stiffness (rigidity) to withstand a 200 lbf (900 N) static point load in any direction without the guard contacting moving parts;

c) be fabricated from sheet (solid or perforated), plate, or expanded metal; any openings shall conform to ISO 14120 or ANSI B11.19, but in no case shall exceed 0.375 in. (10 mm); guards of woven wire shall not be used;

d) require tools for guard removal.

7.3.2.1 Coupling guards shall be constructed of an agreed spark-resistant material, such as brass, aluminum, nickel-copper alloy (see Note on 6.10.2.9.3).

[•] **7.3.2.2** If specified for coupling guards with potentially explosive atmospheres, an ignition hazard assessment (risk analysis) in accordance with ISO 80079–36 shall be conducted and a report provided.

7.3.3 Exposed shaft areas including the area between pump bearing housing(s) and mechanical seal(s) shall have a shaft guard.

7.3.3.1 The shaft guard shall meet the following requirements:

a) prevent personnel from contacting moving parts in any accessible area during operation of the pump; allowable opening dimensions shall comply with ISO 14120 or ANSI B11.19;

b) adequately vented to prevent the accumulation of seal emissions, liquid, or vapor;

c) allow visual inspection of the seal without removal of guard;

d) be constructed of steel, stainless steel, brass, or aluminum materials, as suitable;

e) be fabricated from sheet (solid or perforated), plate, expanded metal, or woven wire and securely fastened to the pump.

7.3.3.2 For pipeline services, a solid shaft guard with a piping connection at the bottom to drain away any accumulated liquid and vapors is required. For this configuration, visual inspection of the seal is not required because the area is enclosed.

[•] **7.3.3.3** If specified, guards shall be designed to meet other purchaser requirements. These may include providing access for volatile organic compound (VOC) emissions testing (US EPA Method 21 for example); protection from environmental elements (e.g. rain, sand); provide protection from directional spray in event of significant seal leakage; and special venting/draining arrangements.

[•] **7.3.3.4** If specified, guards shall be constructed of an agreed spark-resistant material.

[•] **7.3.3.5** If specified for guards with potentially explosive atmospheres, an “ignition hazard assessment” (risk analysis) in accordance with ISO 80079–36 shall be conducted and a report provided.

7.4 Baseplates

7.4.1 Single-piece baseplates designed for grouting shall be furnished for horizontal pumps.

[•] **7.4.2** The purchaser shall specify the type and options as follows:

a) a flat deck plate with a sloped gutter drain that protrudes beyond the side rail flange and surrounds the entire baseplate (Figure 39);

b) a sloped full deck plate, mounted between the side rails, which extends under the pump and drives train components (Figure 40);

c) a sloped partial deck plate, mounted between the side rails, which extends only under the pump and coupling (Figure 41);

d) an open deck version of the above with no deck/top plate (this type may require increased structure to support the pump and nozzle loads);

NOTE The figures above (Figure 39, Figure 40, and Figure 41) are intended to clarify only the conceptual design of the deck plate and not the way it has to be supported by cross members.

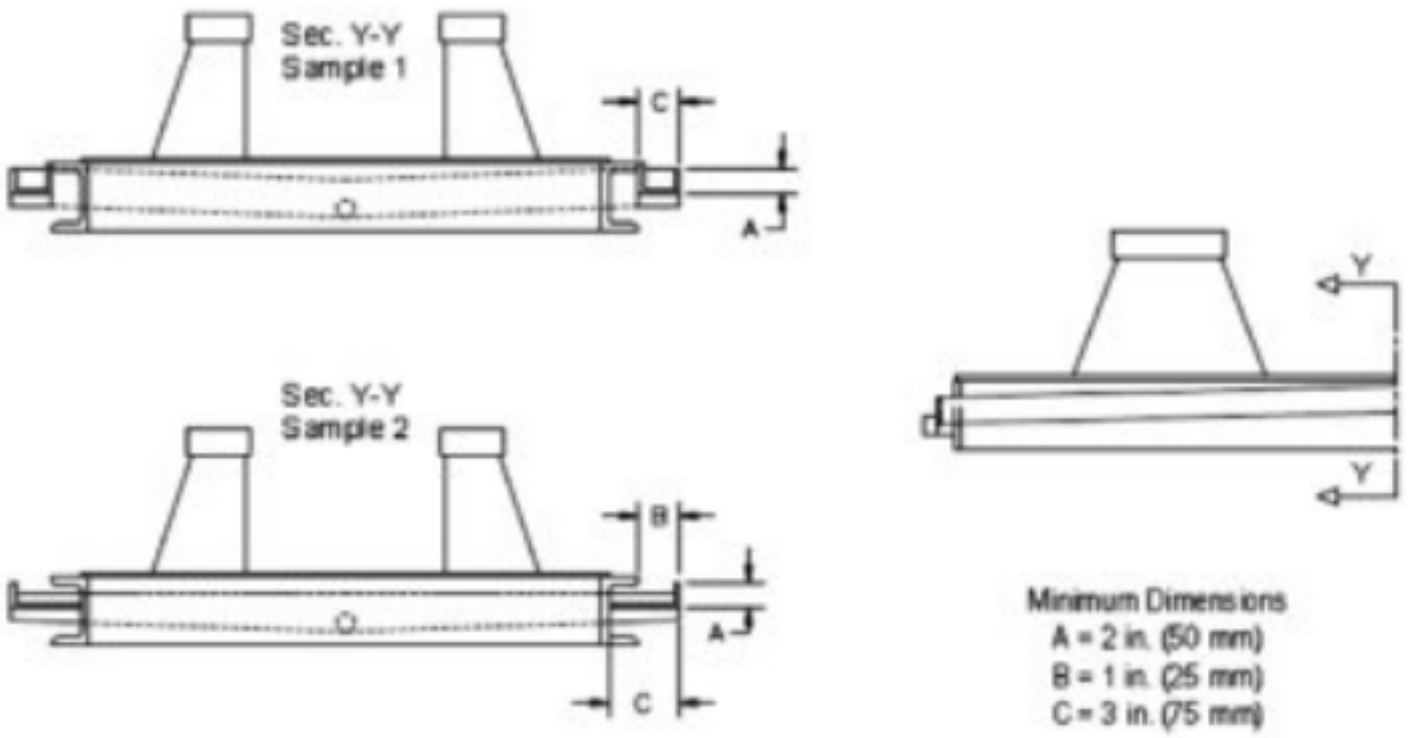


Figure 39—Flat Deck Plate with Sloping Gutter Drain



Figure 40—Sloped Full Deck Plate

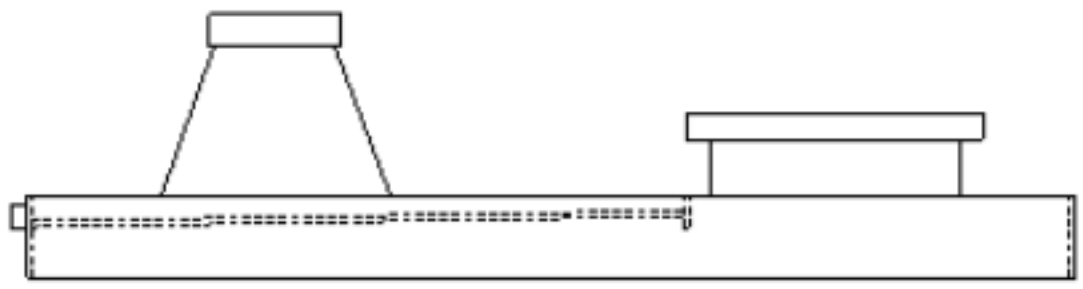


Figure 41—Sloped Partial Deck Plate

[•] **7.4.3** If specified, a non-grouted baseplate of one of the versions in 7.4.2 shall be supplied where the baseplate and pedestal support assembly shall be rigid enough to be mounted without a grout fill, but with uniform support along the side rails of the baseplate;

[•] **7.4.4** If specified, a non-grouted baseplate as in Item 7.4.3 with a gimbal mount, three-point mount, anti-vibration mount (AVM) spring mount, or other type of mount shall be supplied. Due to the distance between supports, such baseplate types require a significantly stiffer and heavier structure to minimize deflections from nozzle loads, driver torque, and other loads.

7.4.5 The slope of either the gutter or deck plate shall be at least 1 in 120 toward the pump end and shall terminate in a tapped drain connection of at least NPS 2 (DN 50). The bottom of the connection shall be located below the bottom of the rim or the deck to affect complete drainage.

7.4.6 The baseplate shall extend under the pump and drive-train components so that any leakage is contained within the baseplate. With purchaser approval, seal flushing system components may be mounted outside of the pump baseplate due to space limitations.

7.4.7 To minimize accidental damage to components, all pipe joints and pipe flange faces, including pump suction and discharge flanges, shall be within the drain collection area. All other projections of the equipment supplied shall fall within the maximum perimeter of the baseplate. Oversized junction boxes may overhang the perimeter of the baseplate with Purchaser's approval.

7.4.8 Baseplate anchor bolt sizing, locations and quantities should meet the following guidelines (see Figure 42):

a) The baseplate design shall have provisions for anchor bolting to withstand nozzle loads, driver torque, and any specified acceleration loads.

b) Quantity of not less than 3 holes, on each side of the baseplate, of minimum 1 in. (25 mm) diameter on the baseplate main longitudinal beams.

c) Holes should be located at the extreme ends of the baseplates at a distance (L1) between 5.875 in. and 6.875 in. (150 mm–175 mm).

d) Holes should be equally spaced, with a distance (L2) not exceeding 35.5 in. ± 0.12 (900 ± 3 mm).

NOTE Previous API 610 editions had a requirement for standard baseplates which have been removed and replaced with the above.

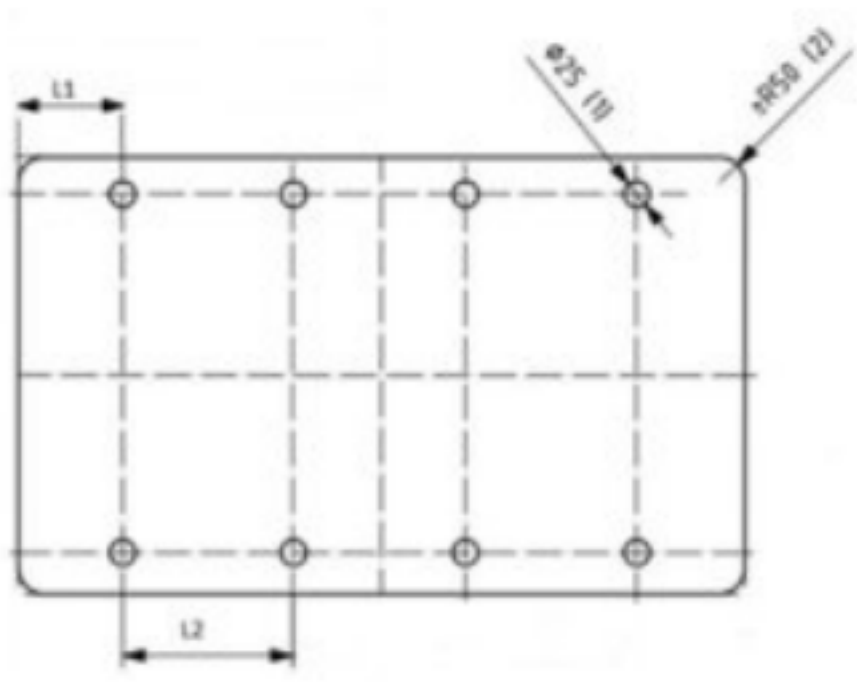


Figure 42—Typical Baseplate Anchor Bolt Hole Guidelines

7.4.9 Auxiliary systems shall not block access to maintain the pump, motor, and coupling. Clear access for maintenance shall be provided on at least one side of the baseplate.

7.4.10 Baseplates mounted to steel modules require special design and should meet the following criteria, as agreed with the purchaser.

7.4.10.1 Baseplates should be designed for point-support rather than perimeter support.

7.4.10.2 Baseplates should be provided with mounting pads on the underside of the baseplate. Mounting pads should be machined flat and parallel in accordance with 6.2.26.

7.4.10.3 Grouting between the baseplate and the steel supporting foundation should not be performed without agreement between purchaser and vendor.

7.4.10.4 The vendor shall provide shims for each baseplate underside mounting pad location. The shim stack shall be no more than 0.12 in (3 mm) thickness with no more than 5 shims per mounting pad location.

7.4.11 The height of the pump-shaft centerline above the baseplate shall be minimized.

7.4.11.1 Enough clearance shall be provided between the casing drain connection and the baseplate for connecting piping by the purchaser.

7.4.11.2 For threaded drains, clearance shall allow the use of a nipple and elbow of the same size as the connection. The use of a street (male–female) elbow is not allowed.

7.4.12 Mounting pads shall be provided for the pump and all drive-train components, such as motors and gears.

7.4.12.1 The pads shall be larger than the foot of the mounted equipment, including the extra width of shims under drive-train components, to allow levelling of the baseplate with a precision level, without removal of the equipment.

7.4.12.2 The minimum exposed area on the mounting pads for leveling purposes shall be 2 in. (50 mm) on two sides of each foot.

7.4.12.3 Mounting pads shall be fully machined flat and parallel to each other. Corresponding surfaces shall be in the same plane within 0.002 in./ft (150 µm/m) of distance between the pads.

7.4.12.4 The flatness requirement shall be demonstrated in the pump-vendor's shop prior to mounting of the equipment and with the baseplate supported at the foundation bolt holes only. This demonstration shall be performed with the baseplate in the unclamped condition, after completion of machining.

NOTE Installed baseplate flatness can be affected by transportation, handling, and installation procedures beyond the vendor's scope. Refer to API 686.

7.4.13 Pumps shall be mounted directly on mounting pads of the baseplate without shims.

7.4.13.1 Shims shall not be used under the pump.

7.4.13.2 Mounting pads for drive-train components shall be machined to allow for the installation of shims at least 0.12 in. (3 mm) thick under each component.

7.4.13.3 If the vendor mounts the drive train components, a set of stainless steel shims (shim packs) at least 0.12 in. (3 mm) thick shall be furnished.

7.4.13.4 Shim packs shall not be thicker than 0.5 in. (13 mm) or contain more than 5 shims.

7.4.13.5 All shim packs shall straddle the hold-down bolts and vertical jackscrews and be full-bearing (extending at least $^1/_4$ in. (5mm) beyond the outer edges of the equipment feet).

7.4.13.6 If the vendor does not mount the components, the pads shall not be drilled and shims shall not be provided.

[•] **7.4.14** If specified, in addition to shims under drive train components, pumps shall be mounted on solid stainless steel plates not less than 0.200 in. (5 mm) thick. Solid plates shall be machined parallel and flat on both sides and shall be the same length and width as the specific pump mounting feet.

[•] **7.4.15** If specified, mechanically adjustable chocks (MACS) shall be used under driver, gearbox, and driven equipment instead of shims.

NOTE MACS require additional gap between the mounted component and the mounting pedestal, this can be achieved by reducing the pedestal height. They are also not full-bearing, per 7.4.13.5.

7.4.16 All joints, including deck plate to structural members and brackets and supports that are welded to the skid, shall be continuously seal-welded on both sides, or full penetration welded, to prevent crevice corrosion. Stitch welding anywhere, top or bottom, is unacceptable.

7.4.17 The bottom of the baseplate between structural members shall be open if the baseplate is designed to be installed and grouted to a concrete foundation.

7.4.17.1 Accessibility shall be provided for grouting under all load-carrying members.

7.4.17.2 The bottom of the baseplate shall be in one plane to permit use of a single level foundation.

7.4.17.3 Cross members shall be provided under pump and driver supports and the members shall be shaped to lock positively into the grout.

7.4.18 All baseplates shall be provided with at least one grout hole in each bulkhead section.

7.4.18.1 Each grout hole shall have a clear area of at least 20 in.^{2} (125 cm^2), with no dimension less than 3 in. (75 mm).

7.4.18.2 Each hole shall be located to permit filling the entire cavity under the baseplate without creating air pockets.

7.4.18.3 The holes should be accessible for grouting with the pump and driver installed on the baseplate.

7.4.18.4 The vendor shall state on the pump arrangement drawing whether any equipment needs to be removed to accomplish grouting of the baseplate.

7.4.18.5 Grout holes in the deck plate area shall have 0.5 in. (13 mm) raised lip edges.

7.4.18.6 If the holes are located in an area where liquids can impinge on the exposed grout, metallic covers with a minimum thickness of 0.06 in. (1.5 mm, 16 gauge) shall be provided.

7.4.18.7 Vent holes at least 0.5 in. (13 mm) in diameter shall be provided on a maximum 18 in. (46cm) centers and at the high point in each bulkhead section of the baseplate.

7.4.19 The outside corners of the baseplate in contact with the grout shall have at least 2 in. (50 mm) radii in the plan view to minimize stress risers in the grout.

7.4.20 The vendor shall commercially sand-blast, in accordance with ISO 8501 Grade Sa2 or SSPC SP 6, all grout contact surfaces of the baseplate and coat those surfaces with a primer compatible with epoxy grout. The manufacturer shall advise the purchaser of the actual primer used.

NOTE Grouts other than epoxy can require alternative surface preparation.

7.4.21 The baseplate shall have provisions for at least four lifting points.

7.4.21.1 Lifting the baseplate, complete with all equipment mounted, shall not permanently distort or otherwise damage the baseplate or the machinery mounted on it.

7.4.21.2 Lifting lugs attached to the baseplate or equipment shall be designed using a maximum allowable stress of one-third of the specified minimum yield strength of the material.

7.4.21.3 Lugs or trunnions that are attached by welding shall have continuous welds. These welds shall be 100 % NDT tested in accordance with the applicable code (see Table 11).

[•] **7.4.22** Baseplates for installation in ambient temperatures below −20 °F (−29 °C), shall be constructed from standard structural steel materials and shapes since they are static, no impact, non-pressure boundary supports.

NOTE Low-temperature steels are not commonly available in structural shapes routinely used for baseplate construction.

7.4.22.1 The lifting lugs for low-temperature baseplates shall be constructed of low-temperature carbon steel or steel alloys rated for the low ambient temperature.

[•] **7.4.22.2** If specified, Charpy impact testing of materials shall be required on low-temperature lifting lugs.

7.4.22.3 Welds on the low-temperature lifting lugs shall be inspected as shown in Table 14.

7.4.22.4 Lifting lugs that are bolted to the baseplate shall have bolting rated for low ambient temperature.

7.4.23 Alignment positioning jackscrews shall be provided for drive-train components and the pump to facilitate horizontal transverse and longitudinal adjustments on the mounting pads.

7.4.23.1 Alignment jackscrews shall be sized for the weight of the drive train component and shall be not less than size $^{1}/_{2}$–13 UNC (M12). For motors where this size is too large, the vendor shall recommend suitable size for purchaser's approval.

7.4.23.2 Alignment jackscrews and any alignment fixtures shall not interfere with the installation or removal of shims under the drive component feet.

7.4.23.3 Any fixtures or any brackets used to hold removable fixtures that are welded to the baseplate shall be welded prior to the machining of the mounting pads to prevent distortion.

7.4.23.4 Any fixture or bracket shall be designed to not interfere with machining the pads in the field.

7.4.23.5 Welded brackets shall not be mounted higher than 0.25 in. (6 mm) below the final machined surface of the pad.

7.4.23.6 Jackscrews shall be corrosion resistant.

7.4.24 Vertical levelling screws shall be provided adjacent to each anchor bolt to level the baseplate during baseplate installation. The jackscrews shall be adequate for the weight of the baseplate, pump, and drive-train components.

7.4.25 To minimize misalignment of the pump and driver shafts due to nozzle loads, the pump and its baseplate shall be constructed with enough structural stiffness to limit displacement of the pump shaft at the drive end of the shaft or at the register fit of the coupling hub to the values shown in Table 13.

7.4.25.1 These values shall be the acceptance criteria for the nozzle load test in 7.4.26 b).

7.4.25.2 Grout shall not be used as a means of obtaining the required stiffness during this test.

[•] **7.4.26** If specified, the vendor shall test to demonstrate that the pump and its baseplate assembly, anchored at foundation bolt hole locations, are in compliance with the following:

NOTE It is recognized that grout can significantly increase the stiffness of the baseplate assembly. Adherence to the nozzle load values in Table 5 limits the total displacement at the pump and drive shaft ends to ~0.010 in. (250 µm) (see Annex F).

a) The pump casing shall be subjected to moments M_{Yc} and M_{Zc} applied to either the suction or discharge nozzle but not both simultaneously such that the corresponding shaft displacements can be measured and recorded. M_{Yc} and M_{Zc} shall not be applied simultaneously.

b) The shaft displacement measurements shall be absolute (not relative to the baseplate).

c) For record purposes, the vendor's test data shall include a schematic drawing of the test set-up, the calculated moment loads (M_{Yc} and M_{Zc}), and the applied moment loads and their corresponding displacements at the drive end of the pump shaft.

[•] d) If specified, the requirements of 7.4.26 can be demonstrated via FEA in lieu of the physical testing.

Table 13—Stiffness Test Acceptance Criteria

Baseplate Intended for Grouting		Baseplate Not Intended for Grouting	
Loading Condition	**Pump Shaft Displacement in. (µm)**	**Pump Shaft Displacement in. (µm)**	**Direction**
M_{Yc}	0.007 (175)	0.005 (125)	+Z
M_{Zc}	0.003 (75)	0.002 (50)	−Y
M_{Yc} and M_{Zc} equal the sum of the allowable suction and discharge nozzle moments from Table 5. $M_{Yc} = (M_Y)_{\text{suction}} + (M_Y)_{\text{discharge}}$ $M_{Zc} = (M_Z)_{\text{suction}} + (M_Z)_{\text{discharge}}$			

7.4.27 Baseplate Grounding

7.4.27.1 The baseplate shall have two grounding lugs/bosses located at diagonally opposite corners.

7.4.27.2 All baseplate mounted electrical equipment shall be grounded to the baseplate.

7.4.28 All accessible walking surfaces on the baseplate shall be made slip resistant.

7.5 Instrumentation

7.5.1 Gauges

Temperature indicators and pressure gauges shall be in accordance with API 614.

7.5.2 Vibration, Position, and Temperature Sensors

[•] **7.5.2.1** The purchaser shall specify whether vibration sensors shall be provided.

[•] **7.5.2.1.1** If specified, accelerometers shall be installed and tested in accordance with API 670.

[•] **7.5.2.1.2** If specified for equipment with hydrodynamic radial and thrust bearings, two radial-vibration probes in each bearing housing, two axial-position probes at the thrust end of each machine, and a one-event-per-revolution probe shall be provided in each machine.

[•] **7.5.2.1.3** If specified, provisions for mounting vibration sensors shall be provided.

7.5.2.1.4 If supplied, the vibration sensors or provisions for the sensors shall be in accordance with API 670.

7.5.2.1.5 If vibration proximity probes are provided, surface areas to be observed by the probes shall meet the requirements of API 670.

[•] **7.5.2.2** The purchaser shall specify whether temperature sensors shall be provided.

7.5.2.2.1 If provided, temperature sensors and their mounting and calibration shall be installed and tested in accordance with API 670.

[•] **7.5.2.2.2** If specified, hydrodynamic thrust and radial bearings shall be fitted with bearing metal temperature sensors.

7.5.2.2.3 If pressure-lubricated hydrodynamic thrust and radial bearings are fitted with temperature sensors, the sensors and their mounting and calibration shall be supplied, installed, and tested in accordance with API 670.

[•] **7.5.2.3** If specified, monitors with cables connecting to vibration, axial-position, or temperature sensors shall be provided. If provided, monitors and cables shall be supplied in accordance with API 670.

[•] **7.5.2.4** If specified, monitors and cables shall be installed on the pump-driver train within the confines of the baseplate in accordance with API 670.

7.6 Piping and Appurtenances

7.6.1 General

7.6.1.1 Piping shall be in accordance with API 614, API 682, API 686, and this standard.

7.6.1.2 Auxiliary systems are defined as piping systems that are in the following services:

— auxiliary process liquids;

— steam;

— cooling water;

— lubricating oil (see 9.2.6).

7.6.1.3 Auxiliary system materials shall be in accordance with Table H.4.

7.6.1.4 The piping systems shall be fully assembled and installed. If this requirement causes difficulty in shipping and handling, alternative arrangements are acceptable with purchaser approval.

[•] **7.6.1.5** If specified, seal liquid reservoirs shall be designed for mounting off the pump baseplate and shall be shipped separately. These reservoirs shall be fully assembled, except that the fluid-circulation tubing shall not be supplied.

7.6.1.6 The vendor shall furnish and locate all piping systems, including mounted appurtenances, within the confines of the baseplate.

[•] **7.6.1.7** If specified, each piping system shall be manifolded to a single purchaser's inlet or outlet connection near the edge and within the confines of the baseplate.

7.6.1.8 The bolting requirements of 6.1.44 shall apply to the connection of auxiliary piping to the equipment.

7.6.1.9 Flange fasteners on auxiliary piping systems shall be low-alloy steel (e.g. ASTM A193/A194M, Grade B7) minimum.

[•] **7.6.1.10** If specified, coating or plating shall be applied to auxiliary piping fasteners.

7.6.1.11 Cadmium plated bolting is not acceptable.

7.6.1.12 Plugs shall comply with 6.4.3.7.

7.6.1.13 Pipe bends shall have surfaces free of cracks and buckles. Flattening of the cross section of bends shall not exceed 8 % of the nominal pipe diameter.

7.6.1.14 Flanges, piping, and fittings shall be of the same class, rating, or wall thickness to match the rating of the system they are installed in.

7.6.2 Auxiliary Process Liquid Piping

7.6.2.1 Auxiliary process-liquid piping includes vent lines, drain lines, balance lines, product flushing lines, seal flush lines, and lines for injection of external fluid.

7.6.2.2 Auxiliary piping, except vent lines, shall be routed to the skid edge, terminating in a flanged connection.

7.6.2.3 The default configuration shall be butt-welded fittings. Socket-welded pipe fittings may be used with purchaser's approval.

NOTE The first pipe connection at the casing is defined by the purchaser per 6.4.3.1.

7.6.2.4 Piping components, including primary seal flush lines, shall have a pressure-temperature rating at least equal to MAWP of the pump casing but in no case less than the ASME flange rating of the discharge connection of the pump.

7.6.2.5 Piping and components subject to the process liquid shall have a corrosion/erosion resistance equal to or better than that of the casing. Otherwise, all components shall be steel.

7.6.2.6 Orifice Plates

7.6.2.6.1 Orifice openings shall not be less than 0.125 in. (3 mm) in diameter.

7.6.2.6.2 Restriction orifices shall have a tab extending from the orifice plate.

7.6.2.6.3 Restriction orifices shall have the size and orifice tag number stamped on the upstream side of the orifice tab.

7.6.2.6.4 The orifice plate shall be removable and flat.

7.6.2.6.5 Alternate flow restriction arrangements (e.g. unions, tubing orifice fittings, multi-stage orifices) require purchaser's approval.

7.6.2.7 Drain valves shall be installed on each drain prior to any manifolding for pumps that require more than one drain connection.

7.6.2.7.1 The drain manifold shall be inside the drain pan limits.

7.6.2.7.2 Atmospheric, low pressure, and high-pressure systems shall not be manifolded together.

NOTE For Between Bearing pumps, it is possible to have a recirculation between the drains or pressurization of an atmospheric connection if manifolded together.

7.6.2.8 For pumps with one drain connection, the drain shall be closed with a blind flange and gasket if a drain valve is not provided.

7.6.2.9 The vendor shall provide space on the baseplate for a purchaser-supplied drain valve inside the pump drain pan or drain rim.

7.6.2.10 If heating or cooling is provided, each exchanger component shall be suitable for the process liquid and cooling water to which it is exposed.

7.6.2.11 Flanges or counter-flanges on the first pipe nipples threaded into seal gland connections shall be lap-joint flanges with butt-weld attachments to the nipples. With purchaser approval, socket-welded unions may be used in place of lap-joint flanges at the first connection from the seal gland.

7.6.2.12 Threaded piping joints may be used only on seal glands, bearing housing instrumentation connections, and for pumps in pipeline services with maximum operating temperature not greater than 130 °F (55 °C).

7.6.2.13 Transmitters and pressure gauges shall have block-and-bleed valves.

7.6.3 Cooling-water Piping

7.6.3.1 The arrangement of cooling-water piping shall conform to Figure B.2, Figure B.3, Figure B.4, Figure B.5, Figure B.6, and Figure B.7, as applicable.

7.6.3.2 Individual coolers and cooler elements shall be piped in parallel such that cooling water does not cool multiple coolers in series.

7.6.3.3 The cooling-water piping shall be designed for the conditions in 6.1.31.

7.6.3.4 For cooling water piping, a closed sight flow indicator shall be furnished in each cooling water outlet.

7.7 Special Tools

7.7.1 If special tools and fixtures are required to disassemble, assemble, or maintain the unit, they shall be included in the quotation and furnished as part of the initial supply of the machine.

7.7.2 For multiple-unit installations, the requirements for quantities of special tools and fixtures shall be agreed upon by the purchaser and the vendor. These or similar special tools shall be used during shop assembly and post-test disassembly of the equipment.

7.7.3 If special tools are provided, they shall be packaged in separate, rugged metal boxes and marked “special tools for (tag/item number).”

7.7.4 Each tool shall be stamped or tagged to indicate its intended use.

7.7.5 Operating procedures for special tools, if any, shall be included in the installation, operating and maintenance manual.

8 Inspection, Testing, and Preparation for Shipment

8.1 General

8.1.1 The purchaser and the vendor shall agree on the points covered in 8.1.1.1 through 8.1.1.4.

[•] **8.1.1.1** If shop inspection and testing have been specified, the purchaser and the vendor shall coordinate hold points and inspector's visits.

8.1.1.2 The expected dates of testing shall be communicated at least 30 days in advance and the actual dates confirmed as agreed. The vendor shall give at least 5 working days advanced notification of a witnessed or observed inspection or test.

[•] **8.1.1.3** If specified, witnessed mechanical and performance tests shall require a written notification of a successful preliminary test. The vendor and purchaser shall agree whether or not to maintain the machine test set-up or whether the machine can be removed from the test stand between the preliminary and witnessed tests.

[•] **8.1.1.4** The purchaser shall specify if no unwitnessed preliminary test is allowed. Purchasers may prefer not to have preliminary tests prior to witnessed tests, to understand any difficulties encountered during testing.

8.1.2 The vendor shall notify subvendors of the purchaser's inspection and testing requirements.

8.1.3 After advance notification to the vendor by the purchaser, the purchaser's representative shall have reasonable access to all vendor and subvendor plants where manufacturing, testing, or inspection of the equipment is in progress. The level of access shall be agreed upon.

8.1.4 Equipment, materials, and utilities for the specified inspections and tests shall be provided by the vendor.

[•] **8.1.5** If specified, the purchaser's representative, the vendor's representative, or both shall indicate compliance in accordance with an inspector's checklist such as that provided in Annex E by initialing, dating, and submitting the completed checklist to the purchaser before shipment.

8.1.6 The purchaser's representative shall have access to the vendor's quality program for review.

8.2 Inspection

8.2.1 General

8.2.1.1 The vendor shall keep the following data available for at least 20 years:

a) necessary or specified certification of materials, such as mill test reports;

b) test data and results to verify that the requirements of the specification have been met;

c) details of all repairs and records of all heat treatment performed as part of a repair procedure;

d) results of quality control tests and inspections;

e) as-built running clearances;

f) other data specified by the purchaser or required by applicable codes and regulations (see L.3.1 and L.3.2).

8.2.1.2 Pressure-containing parts shall not be painted until the specified inspection and testing of the parts is complete.

[•] **8.2.1.3** In addition to the requirements of 8.2.1.6, the purchaser may specify the following:

a) parts that shall be subjected to surface and subsurface examinations;

b) type of examination required, such as magnetic particle (MT), liquid penetrant (PT), radiographic (RT), and ultrasonic (UT) examinations.

8.2.1.4 All preliminary running tests and mechanical checks shall be completed by the vendor before the purchaser's final inspection.

[•] **8.2.1.5** If specified, hydrodynamic radial and thrust bearings shall be inspected per the requirements of ISO 4386. Minimum bond contact between the babbitt and backing material shall be a minimum of 99 %.

8.2.1.6 Pressure Casing and Process Piping Materials Inspection

8.2.1.6.1 Pressure casing and process piping materials shall be inspected in accordance with the requirements of Table 14.

8.2.1.6.2 For double-casing pumps, the outer casing pressure/temperature shall be used to determine the inspection class of the outer casting. The inner casing shall be inspected to class I (Table 14).

8.2.1.7 The timing of the inspections required by Table 14 shall be as follows.

8.2.1.7.1 VI/MT/PT shall be performed after final heat treatment in the proof (rough) machined condition.

8.2.1.7.2 In the proof (rough) machined condition, an additional amount of material remains on areas where machining to critical dimensions and tolerances is required. The additional amount of material removed shall not exceed 0.040 in. (1 mm) material stock or 5 % of minimum allowable wall thickness, whichever is less.

8.2.1.7.3 RT/UT of castings shall be performed after final heat treatment.

8.2.1.7.4 RT of welds and UT of wrought material and welds shall be performed after final heat treatment.

8.2.1.7.5 UT of wrought material shall be performed prior to any machining operations that can interfere with the UT examination.

[•] **8.2.1.7.6** If specified, for machined surfaces, final MT or PT shall be performed after machining is completed.

8.2.1.8 Where the configuration of a casting makes radiography impossible, radiographic examination may be replaced by ultrasonic testing.

8.2.1.9 Inspection methods and acceptance criteria shall be in accordance with those in Table 15 as required by the material specification.

8.2.1.9.1 If additional RT, UT, MT, or PT examination of the welds or materials is specified by the purchaser, the methods and acceptance criteria shall also be in accordance with the standards shown in Table 15.

8.2.1.9.2 Alternative standards may be proposed by the vendor or specified by the purchaser.

[•] **8.2.1.10** If specified, the purchaser may inspect for cleanliness of the equipment and all piping and appurtenances furnished by or through the vendor before assembly.

Table 14—Pressure Casing and Process Piping Material Inspection Requirements

Type of Component	Requirements by Inspection Class [a, g]		
	I	II	III
—	Minimum	> 80 % MAWP and > 400 °F (200 °C) [h] or < −40 °F (−40 °C) [h]	< 0.5 SG or > 400 °F (200 °C)[h] and < 0.7 SG, or > 500 °F (260 °C)[h] and Extremely hazardous services [e] Highly corrosive services [e] Highly erosive services [e]
Casing [b]: cast	VI	VI, plus MT or PT of critical areas	VI, plus MT or PT of critical areas, plus RT or UT of critical areas
Casing [b]: wrought [c f]	VI	VI, plus MT or PT of critical areas	VI, plus MT or PT of critical areas, plus UT of critical areas
Nozzle weld: casing	VI, plus 100 % MT or PT	VI, plus 100 % MT or PT	VI, plus 100 % MT or PT plus RT (100 %)
Auxiliary connection welds [d]	VI	VI, plus 100% MT or PT	VI, plus MT or PT (100 %)
Internals and external gussets	VI	VI	VI
Auxiliary process piping: socket welded	VI	VI, plus 100 % MT or PT	VI, plus 100 % MT or PT
Auxiliary process piping: butt welded	VI, plus 5 % RT	VI, plus 100 % MT or PT and 5 % RT	VI, plus 100 % MT or PT and 10 % RT
Fabricated Casing Welds	VI, plus 100 % MT or PT	VI, plus 100 % MT or PT	VI, plus 100 % MT or PT, plus 100% RT or UT

[a] Definition of abbreviations: MT, magnetic particle inspection; PT, liquid penetrant inspection; RT, radiographic inspection; UT, ultrasonic examination; VI, visual inspection.

[b] "Casing" includes all items of the pressure boundary of the finished pump casing (e.g. the casing itself and other parts, such as nozzles, flanges, attached to the casing). "Critical areas" are inlet nozzle locations, outlet nozzle locations, casting wall thickness changes, casting windows, crossovers, and all through-wall or closure welds, including those made by the vendor or at the foundry level during the manufacturing and production process. The manufacturer shall submit details of the critical areas proposed to receive MT/PT/RT/UT inspection for purchaser's approval.

[c] "Wrought" materials include forgings, plate, and tubular products.

[d] Due to complex geometry and thickness variations, it is not practical to RT butt-welded auxiliary casing connections.

[e] Extremely hazardous, highly corrosive, or highly erosive services, as specified by the purchaser.

[f] Longitudinal welds of seam-welded pipe used to fabricate pressure casing walls shall be inspected by radiography, with 100 % coverage for all inspection classes. See 9.3.2.7.

[g] See 6.12.3.3 and 6.12.3.4 for further non-destructive testing requirement explanations.

[h] Temperatures listed are design temperatures

[•] **8.2.1.11** If specified, the hardness of parts, welds and heat-affected zones shall be verified as being within the allowable values by testing. The method, extent, documentation, and witnessing of the testing shall be agreed upon by the purchaser and the vendor.

[•] **8.2.1.12** If specified, pressure boundary parts of alloy materials shall be subject to positive material identification (PMI) using recognized testing methods, instrumentation, and standards.

8.2.1.12.1 The purchaser and vendor shall agree on the specific parts tested, procedures used, and acceptance criteria. Only techniques providing quantitative results shall be used.

8.2.1.12.2 Mill test reports, material composition certificates, visual stamps, or markings shall not be considered substitutes for PMI testing. PMI is not able to differentiate between grades of carbon steels.

Table 15—Materials Inspection Standards

Type of Inspection	Methods	Acceptance Criteria	
		For Fabrications	For Castings
Radiography (RT)	ASME *BPVC* Section V, Articles 2 and 22	ASME *BPVC* Section VIII, Division 1, UW-51 (for 100 % radiography) and UW-52 (for spot radiography)	ASME *BPVC* Section VIII, Division 1, Appendix 7
Ultrasonic inspection (UT)	ASME *BPVC* Section V, Articles 4, 5 and 23	ASME *BPVC* Section VIII, Division 1, Appendix 12 UW-53 (for Ultrasonic Inspection of welded joints)	ASME *BPVC* Section VIII, Division 1, Appendix 7
Liquid-penetrant inspection (PT)	ASME *BPVC* Section V, Articles 6 and 24	ASME *BPVC* Section VIII, Division 1, Appendix 8	ASME *BPVC* Section VIII, Division 1, Appendix 7
Magnetic-particle inspection (MT)	ASME *BPVC* Section V, Articles 7 and 25	ASME *BPVC* Section VIII, Division 1, Appendix 6	ASME *BPVC* Section VIII, Division 1, Appendix 7
Visual inspection (VI) (all surfaces)	ASME *BPVC* Section V, Article 9	In accordance with the material specification and the manufacturer's documented procedures	MSS SP-55

8.3 Testing

8.3.1 General

[•] **8.3.1.1** If specified, at least six weeks before the first scheduled running test, the vendor shall submit to the purchaser for review and comment, detailed procedures for all running tests and all specified optional tests (8.3.4). The test procedure shall include the actual measurement uncertainty of all data used in the calculation of flow, head, NPSH (if required), and power as well as all acceptance criteria.

8.3.1.2 Performance and NPSH tests shall be conducted using the methods and uncertainty requirements of HI 14.6 (ISO 9906), Grade 1B.

8.3.1.3 Performance tolerances shall be in accordance with Table 16.

8.3.1.4 Evaluation of results shall be in accordance with 8.3.3.4.4.

8.3.2 Hydrostatic Test

8.3.2.1 The intent of a hydrostatic test of a centrifugal pump casing is to ensure that the design and construction of the pump pressure-containing components and joints are leak-free from ambient conditions to the maximum operation conditions.

8.3.2.1.1 All connection welding shall be completed before the casing is hydrostatically tested.

8.3.2.1.2 All connections shall be suitable for the hydrostatic test pressure of the region of the casing to which they are attached.

8.3.2.2 All pressure casing components shall be hydrostatically tested as assemblies. Multistage VS pump bowls may be tested as sub-assemblies.

8.3.2.3 The test shall be conducted with liquid at a minimum of 1.5 times the MAWP.

8.3.2.4 The test set-up or apparatus shall not provide stiffening that improves the integrity of any joint.

8.3.2.5 Gaskets used during hydrostatic testing of an assembled pressure casing, less seal glands, shall be of the same design as those supplied with the pump and shall be installed without sealant.

8.3.2.6 The test liquid shall be at a temperature higher than the nil-ductility transition temperature of the material being tested. Refer to ASTM E1003 for additional information.

8.3.2.7 If the part tested will operate at a temperature at which the strength of a material is below the strength of that material at the testing temperature, the hydrostatic test pressure shall be multiplied by a factor obtained by dividing the allowable working stress for the material at the testing temperature by that at the rated operating temperature.

8.3.2.7.1 The stress values used shall be determined in accordance with 6.2.5.

8.3.2.7.2 For piping, the stress shall conform to ISO 15649 or ASME B31.3, as specified.

8.3.2.7.3 The pressure thus obtained shall then be the minimum pressure at which the hydrostatic test shall be performed.

8.3.2.7.4 The actual hydrostatic test pressures shall be recorded.

8.3.2.7.5 Applicability of this requirement to the material being tested shall be verified before hydrostatic test, as the properties of many grades of steel do not change appreciably at temperatures up to 400 °F (200 °C).

8.3.2.8 The hydrostatic test liquid shall include a wetting agent to reduce surface tension if one or more of the following conditions exist;

a) the liquid pumped has a relative density (specific gravity) of less than 0.7 at the pumping temperature;

b) the pumping temperature is higher than 500 °F (260 °C);

c) the casing is cast from a new or altered pattern;

d) the materials are known to have poor castability.

8.3.2.9 The chloride content of liquids used to test austenitic stainless steel materials shall not exceed 50 ppm (50 mg/kg). To prevent deposition of chlorides and to prevent stress-corrosion cracking as a result of evaporative drying, all residual liquid shall be removed from the tested parts at the conclusion of the test.

8.3.2.10 Hydrostatic testing is permitted without the seal-gland plate or removable seal chamber installed.

8.3.2.10.1 The mechanical seal shall not be included in the hydrostatic test of the pump case.

8.3.2.10.2 Gland plates and removable seal chambers shall be tested as specified in API 682.

8.3.2.11 Austenitic, duplex, and super duplex stainless steel pressure casing components may be hydrostatically tested in the proof (rough) machined condition (see 8.2.1.7). Any areas that are machined after hydrostatic testing shall be identified on the hydrostatic test report with a machining map.

NOTE By allowing a small amount of material to remain at these critical areas during hydrostatic testing, the necessity to add material by welding to restore close-tolerance dimensions after hydrostatic test is avoided.

8.3.2.12 Hydrostatic Testing Procedure

8.3.2.12.1 Test pressure shall be maintained to permit complete examination of parts under pressure for a minimum of 30 minutes.

8.3.2.12.2 Neither leaks nor seepage through the pressure-containing parts and joints are allowed.

[•] **8.3.2.12.3** If specified, pressure-containing parts shall be tested for a longer testing period, as agreed upon by the purchaser and the vendor.

8.3.2.12.4 Once pressurized, the pump shall be isolated from the pressure source for the duration of the hydrostatic test. Operation of a test pump to maintain pressure during hydrostatic testing requires purchaser's approval.

8.3.2.12.5 The hydrostatic test shall be considered satisfactory if the pump/part maintains the test pressure and no leaks or seepage occurs.

NOTE Ambient temperature changes (e.g. due to weather, direct sun) during longer hydrostatic tests can cause temporary increases or decreases in test pressure, which can be considered during test evaluation.

8.3.2.13 Special-design pumps may be segmentally hydrostatically tested with purchaser approval.

8.3.2.13.1 Seepage past internal closures required for testing of segmented cases and operation of a test pump to maintain pressure is acceptable.

8.3.2.13.2 If testing of sections of a casing at different pressures is approved, each section shall be tested independently at the required pressure. Alternatively, a combined test may be conducted with the required pressure simultaneously in each section.

NOTE A segmental hydrostatic test can be proposed if the physical construction of the pump (VS6, VS7, BB3, BB4, OH6, BB5) necessitates a different rating, if specified in 6.2.11.

8.3.2.14 Piping systems fabricated by welding shall be hydrostatically tested. Testing and stress levels shall be in accordance with ASME B31.3. It is unnecessary to hydrostatically test piping systems assembled with tubing or threaded connections after assembly.

8.3.2.15 Single-stage overhung-pump casing components with a radial joint (mean gasket diameter) 24 in. (610 mm) in diameter or less may be hydrostatically tested as components or subassemblies provided that joint design integrity has been proven by qualification testing for the specific size of pump and pressure rating in question.

8.3.2.16 Cooling passages and components, including jackets for bearings, seal chambers, oil coolers, and seal coolers, shall be tested at 1.5 times maximum operating gauge pressure or 150 psi (10.5 bar), whichever is greater.

8.3.2.17 Mechanical seal flushing liquid coolers shall be hydrostatically tested on the process side to at least the same pressure as the pump casing hydrostatic pressure.

8.3.2.18 Steam, cooling-water, and lubricating-oil piping, if fabricated by welding, shall be tested at 1.5 times maximum operating gauge pressure or 150 psi (10.5 bar), whichever is greater.

8.3.3 Performance Test

8.3.3.1 Each pump shall be given a performance test in accordance with HI 14.6 or ISO 9906, including the additional requirements of this standard.

8.3.3.2 The following requirements shall be met while the pump is operating on the test stand and before the performance test is performed.

8.3.3.2.1 The contract seals and bearings shall be used in the pump for the performance test.

8.3.3.2.2 The seal (or seals) shall not have a leakage rate during any phase of the pump performance test that is in excess of that specified in API 682, or as otherwise agreed.

8.3.3.2.2.1 Seal atmospheric vents and drains shall be left open or unplugged during the performance test.

8.3.3.2.2.2 Any unacceptable leakage during the pump performance test requires a disassembly and repair to the seal.

8.3.3.2.2.3 If the seal is disassembled or removed, the seal shall be retested with an air test of the pump using the criteria defined in 8.3.3.12.7.

8.3.3.2.2.4 When the pump is on the test stand and water is used as the test liquid, liquid seals suitable for testing on water shall exhibit no visible signs of leakage.

NOTE API 682 can be reviewed to confirm that a zero-visible-leakage criterion is appropriate for the seals being tested. Notably, pressurized dual seals with high barrier-fluid pressures greater than 600 psi (40 bar) are prone to visible signs of leakage.

8.3.3.2.3 All lubricating-oil pressures, viscosities, and temperatures shall be within the range of operating values recommended in the vendor's operating instructions for the specified unit being tested.

8.3.3.2.4 Bearings specified as normally lubricated from a pure oil-mist system shall be pre-lubricated prior to performance testing using a suitable hydrocarbon oil.

8.3.3.2.5 All joints and connections shall be checked for tightness and any leaks shall be corrected.

8.3.3.2.6 All warning, protective, and control devices used during the test shall be checked and adjusted as required.

8.3.3.2.7 Performance tests shall be performed using water at a temperature not exceeding 130 °F (55 °C).

NOTE Per HI 14.6–2022, Annex M, test measurement corrections are recommended once water temperature exceeds 86 °F (30 °C) and required above 104 °F (40 °C) because the change in water density is not negligible. Not correcting, as described, can lead to incorrect test results, due to underestimating the pressure and a “drooping” curve as you approach the MCSF and shutoff.

8.3.3.3 The following options for testing require approval of the purchaser.

8.3.3.3.1 If approved by the purchaser, substitute seals may be used during the performance test, if needed, to prevent damage to the contract seals or if the contract seals are not compatible with the test liquid. See API 682.

8.3.3.3.2 Seal leakage during the test beyond specified acceptance criteria established for the test shall require the assembled pump and seal to be rerun to demonstrate satisfactory seal performance.

8.3.3.3.3 Performance tests may be conducted at water temperatures exceeding 130 °F (55 °C) if approved by the purchaser. Proper personnel protection is required for elevated testing temperatures.

8.3.3.4 The performance test shall be conducted as follows.

8.3.3.4.1 The vendor shall take test data, including head, flowrate, power, and vibration at a suitable number of points, to characterize the performance curve.

8.3.3.4.2 Each test point within the allowable operating region of the pump shall be no more than 35 % (of rated flow) removed, by its flowrate, from any other test point within the allowable operating region. Figure M.4 provides a graph demonstrating the points. These points will normally include the following and may have a tolerance of ±3 % flow:

1) shutoff (no vibration data required),

2) flow point at minimum continuous stable flow (beginning of allowable operating region),

3) flow point at approximately halfway between minimum continuous stable flow and minimum flow of the preferred operating region,

4) flow point at minimum flow of the preferred operating region,

5) flow point at approximately halfway between minimum flow of the preferred operating region and rated flow,

6) flow point between 95 % and 99 % of rated flow,

7) flow point between rated flow and 105 % of rated flow,

8) flow point at the end of preferred operating region,

9) flow point at the end of allowable operating region if different from the end of the preferred operating region.

For units with BEP less than 50 gpm (11 m^3/h), Point 3) and Point 5) are not required.

8.3.3.4.3 Some higher energy, integrally geared, or multistage pumps exhibit a high rate of increase of the temperature at shutoff that makes it not feasible or unsafe to test them at shut off. The temperature increase is strictly related to the power density. The power density, PD, can be approximated as:

$$PD = \frac{P_{rated}}{D_{imp}{}^{2} \times D_{nozzle}} \tag{4}$$

where

P_{rated} is the rated power per stage in hp (or MW) with water;

D_{imp} is the rated impeller diameter in in. (or m);

D_{nozzle} is the outlet flange nominal (NPS diameter) in inches (or meters). For double-suction, single-stage pumps, D_{nozzle} is the inlet flange nominal diameter.

A typical critical value of PD, beyond which it is suggested that the pump not be operated at shutoff during performance test is 0.286 hp/$in.^3$ (13 MW/m^3).

8.3.3.4.4 For pumps that cannot be tested to shutoff, the rise to shutoff requirement (6.1.21) shall apply to minimum continuous thermal flow or to 10 % of BEP flowrate whichever is less.

8.3.3.4.5 The test data shall be fit to a spline or an appropriate polynomial (typically, not less than a third order) for head and for power using a least squares method.

8.3.3.4.5.1 The resulting polynomial equation shall be stated on the head and power calculated.

8.3.3.4.5.2 These values shall be corrected for speed, viscosity, and density (specific gravity).

8.3.3.4.5.3 The corrected values of head and power shall be within the tolerance bands allowed in Table 16.

Table 16—Performance Tolerances (See Footnotes c and d)

Condition	Rated Point %	Shutoff %
Rated differential head [c]:		
0 to 250 ft (0 to 75 m)	± 3	± 10 [a]
> 250 ft to 1000 ft (75 m to 300 m)	± 3	± 8 [a]
> 1000 ft (> 300 m)	± 3	± 5 [a]
Rated power	+4	—
Efficiency	[b]	
Rated NPSH	0	—

[a] The negative tolerance specified here shall be allowed only if the test curve still shows a rising characteristic, per 6.1.21.

[b] The uncertainty of test efficiency by the test code specified is ± 2.5 %; therefore, efficiency is not included in the pump's rated performance. In those applications where efficiency is of prime importance to the purchaser, a specific value and related tolerance shall be negotiated at the time of the order (see 8.3.3.10).

[c] The tolerances at the rated point in this table are identical to those for a class 1B test in HI 14.6 (ISO 9906). They are repeated here because HI does not list tolerances to shutoff. API 610 does not use the HI tolerance on efficiency. API 610 uses the head and the power tolerances at a specified rated flow.

[d] Attention is called to Section 14.6.3.4.1 of HI 14.6. The tolerances in this table may not be appropriate for pumps with rated powers less than 13 hp (10 kW).

[e] Pumps that will be operated in parallel or run on ASD shall comply with the head tolerance required in 6.1.18).

8.3.3.5 The test speed shall be within 3 % of the rated speed. Test results shall be corrected to rated speed.

8.3.3.6 For pumps with specific gravity less than 0.7, a reduced speed test may be performed with purchaser's approval.

NOTE HI 14.6 provides guidance for testing at reduced speed. Multistage pumps can also have an impeller stage removed, if necessary for testing.

8.3.3.7 The vendor shall maintain a complete, detailed log of all final tests and shall prepare the required number of copies, certified for correctness. Data shall include test curves and a summary of test performance data compared with guarantee points (see L.2.4, and example in Annex M).

8.3.3.8 In addition to formal submittal of final data in accordance with L.2.4, curves and test data (corrected for speed, specific gravity, and viscosity) shall be submitted within 24 hours after completion of performance testing for purchaser's engineering review and acceptance prior to shipment.

[•] **8.3.3.9** For higher-power pumps (> 1350 hp (> 1 MW)), performance tolerances other than those in Table 16 may be appropriate. If specified, pump efficiency at rated flow shall be quoted to the tolerance given by the purchaser and shall be included in the pump's rated performance.

8.3.3.10 If a tolerance is specified for rated efficiency, an additional test point as close to rated flow as practical shall be taken. The rated efficiency and tolerance shall be consistent with the test code being used, with particular attention to the uncertainty of efficiency determined by test to that code.

8.3.3.11 During the performance test, the following requirements shall be met.

8.3.3.11.1 Vibration values shall be recorded at the test points specified in 6.9.4.3 and in accordance with the requirements of 6.9.4.3 through 6.9.4.9. Vibration values shall not exceed those given in 6.9.4.10.

8.3.3.11.2 For ring and splash-oil systems, the temperatures of the bearing housing or bearings fitted with sensors shall be measured in real time and recorded throughout the test.

8.3.3.11.3 For pure-oil mist systems, the temperature of the bearing housing at the bearing location or bearings fitted with temperature sensors shall be measured in real time and recorded throughout the test.

8.3.3.11.4 For pressurized lubrication systems, bearing metal temperatures shall be measured in real time and recorded throughout the test. The duration of the test shall be indicated on the test report.

8.3.3.11.5 Pumps shall operate within bearing temperature limits as defined in 6.10.2.7 and shall not display signs of unusual operation, such as noise caused by cavitation.

8.3.3.11.6 If operated at rated speed, pumps shall perform within the tolerances given in Table 16.

8.3.3.11.7 The equipment shall operate without visible oil leaks.

[•] **8.3.3.11.8** If specified, the performance test shall be conducted with the test stand NPSHA controlled to no more than 110 % of the NPSHA specified by the purchaser.

8.3.3.12 The following requirements shall be met after the performance test is completed.

8.3.3.12.1 If it is necessary to dismantle a pump after the performance test for the sole purpose of machining impellers to meet the tolerances for differential head, no retest is required unless the reduction in diameter exceeds 5 % of the original diameter.

8.3.3.12.2 The diameter of the impeller at the time of shop test, as well as the final diameter of the impeller, shall be recorded on a certified shop test curve that shows the operating characteristics after the diameter of the impeller has been reduced.

8.3.3.12.3 Pump hydraulic modifications using impeller vane machining techniques of underfiling, overfiling, V-cutting or similar to change the curve shape after performance testing requires purchaser approval. If any such vane machining techniques are undertaken, the pump shall be fully re-tested and the modification shall be documented as part of the test report.

NOTE VS Pumps typically employ under-filing as part of the normal manufacturing process.

[•] **8.3.3.12.4** If specified, disassembly of multistage pumps for any head adjustment (including less than 5 % diameter change) after test shall be cause for retest.

8.3.3.12.5 If it is necessary to dismantle a pump for any other correction, such as hydraulic performance, NPSH, or mechanical operation, the initial test shall not be acceptable, and the final performance test shall be run after the correction is made.

8.3.3.12.6 When an impeller, inducer or rotor is post-test machined, the component or complete rotor shall be rebalanced prior to retest or shipment in accordance with 6.9.3 and 9.2.4.2 as applicable.

8.3.3.12.7 If it is necessary to disturb the mechanical seal assembly following the performance test, or if the test seal faces are replaced with the job seal faces, the final seal assembly shall be air-tested as follows:

a) pressurize each sealing section independently with clean air to a test gauge pressure of 25 psi (172 kPa, 1.72 bar);

b) isolate the test set-up from the pressurizing source and maintain the pressure for a minimum of 5 min, or 5 min per 1 ft^3 (30 l) of test volume, whichever is greater;

c) the maximum allowable pressure drop during the test shall be 2 psi (14 kPa, 0.14 bar).

8.3.3.13 Pumps shall not be disassembled after final performance testing.

8.3.3.13.1 The pump, including the seal chamber, shall be drained to the extent practical, filled with a water-displacing inhibitor within 4 hours of testing and redrained.

[•] **8.3.3.14** If specified, on some pump types such as BB2, BB3, and BB5, if the water cannot be eliminated by draining, the pumps shall be disassembled as much as necessary to ensure that water is removed from internal passageways and balance lines, prior to the inhibitor application, to prevent corrosion and possible freezing damage.

NOTE Disassembly of the pump after final performance testing can impact the mechanical integrity of the unit as proven on test.

8.3.4 Shaft Taper Check

8.3.4.1 A contact blue check and tape lifts shall be conducted for tapered fit couplings.

8.3.4.2 Surface contact shall be minimum 85 % for keyless tapers and 70 % for keyed tapered bores, per API 671.

8.3.4.3 A report of the contact check that includes pictures of the tape shall be provided to the purchaser.

8.3.5 Optional Tests

8.3.5.1 General

[•] If specified, the shop tests described in 8.3.5.2 through 8.3.5.8 shall be performed. Test details and required data (such as vibration and temperature data) shall be agreed upon by the purchaser and the vendor prior to conducting the tests.

8.3.5.2 Mechanical Run Test

[•] **8.3.5.2.1** If specified, the pump shall be run on the test stand at the rated flow and the bearing oil temperatures measured and recorded.

8.3.5.2.2 The test shall continue until oil temperature measurements stabilize; that is, when temperature rise relative to ambient temperature is not more than 2 °F (1 °C) over a 10-minute period.

NOTE Small pumps equipped with ring-oiled or splash lubrication systems typically do not reach temperature stabilization during performance tests of short duration.

8.3.5.2.3 After bearing temperature stabilizes, the test shall be run for a minimum of 1 hour. The acceptance criteria for mechanical run test shall be the same as the performance test.

8.3.5.2.4 All lubricating-oil pressures, viscosities, and temperatures shall be within the range of operating values recommended in the vendor's operating instructions for the specified unit being tested.

[•] **8.3.5.2.5** If specified, during the mechanical run test, the rated flow, pressure, power, vibration, speed, bearing oil or housing temperatures shall be measured and recorded every 5 minutes, at minimum, throughout the test.

[•] **8.3.5.2.6** If specified, the vibration trend shall be recorded continuously.

[•] **8.3.5.2.7** If specified, the pump shall be mechanically run at the rated flow for 4 hours.

8.3.5.2.8 If it is necessary to dismantle a pump for any correction such as due to noise, vibration or leakage, the mechanical run test of that unit shall be repeated after the correction is made.

8.3.5.3 NPSH Required Test

[•] **8.3.5.3.1** If specified, the pump shall be given an NPSH required test in accordance with HI 14.6 or ISO 9906 except with the additional requirements of this standard.

8.3.5.3.2 NPSH required tests shall determine NPSH3 values at each of the test points 2, 5, 6, 7 and 8 (as identified in 8.3.3.4.2) based on 3 % head drop in single-stage pumps or 3 % first-stage head drop in pumps with two or more stages.

8.3.5.3.3 The first-stage head of pumps with two or more stages shall be measured using a separate connection to the first-stage discharge, if possible, or alternatively, testing of the first stage only shall be conducted. Only with purchaser approval, the first-stage head may be determined by dividing total head by the number of stages.

8.3.5.3.4 NPSH required tests shall be conducted for each test point by maintaining constant flow and measuring head vs NPSHA data as the NPSHA to the pump is reduced progressively until the drop in head is more than 3 % of the first head value (NPSH Type II test per HI 14.6). The test shall not proceed beyond a 20 % head breakdown (20 % of first-stage head for multistage pumps).

8.3.5.3.5 For each test point of the NPSH required test, the first head value shall be measured with a starting NPSHA value equal to at least twice the NPSH3 value shown on the proposal curve at the corresponding flow, or if greater, with a starting NPSHA value equal to the NPSHA provided to the pump during the performance test at the corresponding flow.

NOTE If the performance test requirement of 8.3.3.11.8 is specified, it is possible that the head has already been affected by insufficient NPSHA, so starting at a higher NPSHA is desirable.

8.3.5.3.6 For each test point of the NPSH required test, the second head value measured after the NPSHA to the pump is reduced by at least 3 ft (1 m) shall differ from the first head value by no more than the combined permissible uncertainty of the differential head measurement devices used to conduct the test. If the second head value differs from the first by more than this uncertainty, the starting NPSHA to the pump shall be increased and the measurements for head vs NPSHA data shall be repeated, until two consecutive head measurements achieve this tolerance.

8.3.5.3.7 Sets of head vs NPSHA data, one set for each test point of the NPSH required test, shall be recorded and plotted as separate head vs NPSHA curves. The NPSH3 values achieved at each test point shall be marked on these curves respectively.

8.3.5.3.8 The NPSH required test results shall be recorded by producing an NPSH3 vs flow curve in accordance with HI 14.6 or ISO 9906.

8.3.5.3.9 NPSH3 at the rated point shall not exceed the quoted value (see Table 16).

8.3.5.3.10 Dismantling to correct NPSH3 performance requires retesting of the performance and NPSH required tests.

8.3.5.4 Complete Unit Test

[•] **8.3.5.4.1** If specified, the pump and driver train, complete with all auxiliaries that make up the unit, shall be tested together.

[•] **8.3.5.4.2** If specified, torsional vibration measurements shall be made to verify the vendor's analysis.

[•] **8.3.5.4.3** The complete-unit test shall be performed in place of or in addition to separate tests of individual components specified by the purchaser.

8.3.5.4.4 The acceptable vibration limits of each component of the train shall be as per its applicable standards and specifications, except for reciprocating engines (in this case, limits shall be mutually agreed upon by purchaser, pump vendor, and engine subvendor).

8.3.5.5 Sound Level Test

[•] If specified, sound level tests shall be performed as agreed between the purchaser and the vendor.

NOTE Noise levels are typically recorded during the performance or mechanical run test for information only. During the FAT, noise levels cannot be guaranteed or meet the specified noise level values because of uncontrolled workshop background noise and the generated noise from the throttle valve.

8.3.5.6 Auxiliary Equipment Test

[•] If specified, auxiliary equipment, such as oil systems, gears, and control systems, shall be tested in the vendor's shop. Details of the auxiliary equipment test(s) shall be developed jointly by the purchaser and the vendor.

8.3.5.7 Bearing-housing Resonance Test

[•] If a resonance test is specified, the bearing housing(s) shall be excited by impact with the pump unpiped, and the modes of natural frequency determined by FFT spectrum measurements and the requirements of 8.3.5.7.1 through 8.3.5.7.6 shall be followed.

8.3.5.7.1 Data shall be recorded and stored.

8.3.5.7.2 An agreed separation margin shall exist between determined modes of natural frequency and the following excitation frequencies:

a) multiples of running speed, expressed in revolutions per minute equal to 1.0, 2.0, and 3.0 times;

b) multiples of vane passing frequency equal to 1.0 and 2.0 times.

8.3.5.7.3 Test acceptance criteria shall be agreed upon between the purchaser and the vendor.

8.3.5.7.4 In case identified bearing housing resonance conditions cannot be detuned, acceptability of the dynamic behavior of the bearing housing(s) and its attachment shall be proven by means of bearing housing vibration measurements during regular performance test.

8.3.5.7.5 Conditions where bearing housing vibration levels exceed the maximum allowable values stated in 6.9.4.10 shall then be corrected by changing the design, with purchaser's approval.

[•] **8.3.5.7.6** In case of an ASD application, the purchaser shall specify the applicable continuous operating speed range.

8.3.5.8 Spare Parts Test

[•] **8.3.5.8.1** If specified, spare parts such as rotors, couplings, gears, diaphragms, bearings, and seals shall be tested.

8.3.5.8.2 The vendor shall ensure that the spare parts are available for test at the same time as the original equipment and that all of the necessary components to test them are supplied.

8.3.5.8.3 The test procedure shall be agreed between purchaser and vendor.

8.4 Preparation for Shipment

[•] **8.4.1** If specified, a physical alignment between the pump shaft and the driver shaft shall be performed prior to shipment and reported to the purchaser.

8.4.1.1 The alignment report shall document the angular and offset alignment requirements with allowances for thermal growth for the particular equipment.

8.4.1.2 The alignment report shall record the actual angular and offset alignment results achieved, the measured DBSE, and the smallest measured radial clearance between each hold-down bolt and its bolt hole in the equipment.

8.4.1.3 The minimum radial clearances between hold-down bolts and bolt holes shall be no less than 0.03 in. (0.8 mm) after acceptable alignment is achieved.

[•] **8.4.1.4** If specified, the alignment between the pump shaft and the driver shaft shall be witnessed by the purchaser.

8.4.2 Equipment shall be prepared for domestic shipment, which shall make the equipment suitable for outdoor storage for a period of at least 6 months with no disassembly required before operation except possibly inspection of bearings and seals.

[•] **8.4.3** If Specified, preparation for longer storage or export shipment shall be provided by the vendor following agreed procedures.

8.4.4 The equipment shall be prepared for shipment after all testing and inspection has been completed and the equipment has been released by the purchaser. The preparation shall include that specified in 8.4.4.1 through 8.4.4.9.

8.4.4.1 Rotors shall be blocked if necessary. Blocked rotors shall be identified by means of corrosion-resistant tags attached with stainless steel wire.

8.4.4.2 Internal surfaces of bearing housings and carbon-steel oil-systems components shall be coated with an oil-soluble rust preventive that is compatible with the lubricating oil.

8.4.4.3 Bearing assemblies shall be fully protected from the entry of moisture and dirt.

8.4.4.3.1 If vapor-phase inhibitor crystals in bags are installed in large cavities, the bags shall be attached in an accessible area for ease of removal.

8.4.4.3.2 If applicable, bags shall be installed in wire cages attached to flanged covers and bag locations shall be indicated by corrosion-resistant tags attached with stainless steel wire.

8.4.4.4 Exterior surfaces, except for machined surfaces, shall be given at least one coat of the manufacturer's standard paint.

8.4.4.4.1 The paint shall not contain lead or chromates.

8.4.4.4.2 It is not necessary to paint stainless steel parts.

8.4.4.4.3 The undersides of baseplates shall be prepared for grout in accordance with 7.4.20.

8.4.4.5 Exterior machined surfaces, except for corrosion-resistant material, shall be coated with a rust preventive.

8.4.4.6 Flanged openings shall be provided with metal closures at least 0.19 in. (5 mm) thick, with elastomeric gaskets and at least four full-diameter bolts. For studded openings, all nuts required for the intended service shall be used to secure closures.

8.4.4.7 Threaded openings shall be provided with plugs in accordance with 6.4.3.7.

8.4.4.8 Openings that have been beveled for welding shall be provided with closures designed to prevent entrance of foreign materials and damage to the bevel.

8.4.4.9 Exposed shafts and shaft couplings shall be wrapped with waterproof, moldable waxed cloth or volatile-corrosion inhibitor paper. The seams shall be sealed with oil-proof adhesive tape.

8.4.5 Auxiliary piping connections furnished on the purchased equipment shall be impression-stamped or permanently tagged to agree with the vendor's connection table or general arrangement drawing.

8.4.5.1 Service and connection designations shall be indicated.

8.4.5.2 Symbols for all pump connections, including plugged connections, shall be in accordance with Annex B.

8.4.6 Lifting points and lifting lugs shall be clearly identified.

8.4.7 The equipment shall be identified with item and serial numbers.

8.4.8 Material shipped separately shall be identified with securely affixed, corrosion-resistant metal tags indicating the item and serial number of the equipment for which it is intended.

8.4.9 Crated equipment shall be shipped with duplicate packing lists, one inside and one on the outside of the shipping container.

8.4.10 One copy of the manufacturer's standard installation manual shall be packed and shipped with the equipment.

8.4.11 The vendor shall provide the purchaser with API 686-compliant instructions for the preservation of the integrity of the storage preparation at the job site and before start-up.

8.4.12 Horizontal pumps, and all furnished drivers and auxiliaries, shall be shipped fully assembled on their baseplates, except as noted below.

8.4.13 Coupling spacers with bolts and other items, such as minimum flow orifices that are not part of the assembled pumping unit, shall be separately boxed, tagged, and securely attached to the baseplate.

8.4.14 Drivers for vertical pumps and horizontal drivers with a mass over 450 lb (200 kg) may be removed after shop mounting and alignment and shipped separately but alongside pump.

8.4.15 Vertical pumps with suction cans shall be shipped with the suction cans (barrels) removed.

8.4.16 If it is necessary to ship other major components separately, prior purchaser approval is required.

8.4.17 Metal filter elements and screens shall be cleaned and reinstalled prior to shipment. Nonmetallic filter elements shall be shipped and installed in an unused condition.

8.4.18 Suitable rust preventatives shall be oil-soluble and compatible with all pumped liquids.

8.4.19 Equipment or materials that contain or are coated with chemical substances (for example, oils, corrosion inhibitors, antifreeze solutions, desiccants, hydrocarbon substances, and unused paint) shall be prominently tagged at openings to indicate the nature of contents and precautions for shipping, storage, and handling.

8.4.19.1 Substances that are supplied with the shipment shall have a safety data sheet (SDS).

8.4.19.2 If a substance is exempt from regulation, a statement to that effect shall be included.

8.4.19.3 At least 2 weeks before shipment, SDSs shall be forwarded to the receiving facility, to allow planning for handling of any regulated substances.

8.4.19.4 SDSs in protective envelopes shall be affixed to the outside of the shipping package.

9 Specific Pump Types

9.1 Single-stage Overhung Pumps

9.1.1 Horizontal (Type OH2) Pumps

9.1.1.1 Rear pump bearing housing supports are not permitted.

9.1.1.2 The distance between the pump and driver shaft ends (DBSE) shall permit removal of the coupling spacer and back pull-out assembly without disturbing the driver, coupling hubs, or casing.

[•] **9.1.1.3** If specified, the shaft flexibility index (SFI) shall be calculated by the vendor in accordance with Equation (K.1) and reported.

9.1.1.4 Rotor balancing for OH2 pumps is not required. Component balancing is sufficient.

9.1.2 Vertical In-line (Type OH3 / OH4 / OH5) Pumps

9.1.2.1 A flat contact surface shall be provided on the bottom of the casing to make the pump stable if freestanding on a pad or foundation.

9.1.2.1.1 The ratio of the unit center of gravity height to the contact surface width shall be no greater than 3:1.

9.1.2.1.2 This stability shall be achieved through the design of the casing or by a permanent external stand.

9.1.2.2 Pumps shall be designed so that they can either float with the suction and discharge piping or be bolted to a pad or foundation. Flange loading on the pump can increase if the purchaser elects to bolt down the unit and should be addressed in the piping requirements.

9.1.2.3 A minimum NPS $^{1}/_{2}$ (DN 15) tapped drain connection shall be provided so that no liquid collects on the cover or driver support.

9.1.2.4 The pump and seal chamber shall be continuously vented with a high point connection in either the seal chamber or seal flush piping. Purchaser approval is necessary for systems that require manual venting.

9.1.2.5 For OH3 pumps, the distance between the pump and driver shaft ends (DBSE) shall permit removal of the coupling spacer and back pull-out assembly without disturbing the driver, coupling hubs, or casing.

[•] **9.1.2.6** If specified, for OH3 pumps, a device that allows direct rigging or lifting of the back pull-out assembly from outside the motor support with the driver in place shall be provided.

9.1.2.7 With the purchaser's approval, for OH3 pumps, bearing housings may be arranged for grease lubrication (6.11.5). The stabilized bearing-housing temperature shall not exceed a 70 °F (39 K) rise above ambient temperature.

9.1.2.8 Drivers shall be aligned and matchmarked in the vendor's shop prior to shipment.

9.1.2.9 For vertical inline pumps, driver supports shall be steel.

9.1.2.10 Rotor balancing for vertical inline pumps is not required. Component balancing is sufficient.

9.1.3 Integral Gear-driven (Type OH6) Pumps

9.1.3.1 The impeller shall be keyed or splined to the gearbox output shaft.

9.1.3.2 Integral-gear pumps may require removal of the driver to allow disassembly of the rotor and the seal assembly.

9.1.3.3 The need for a rotor lateral analysis shall be determined as described in Annex I.

9.1.3.4 Single-piece hydrodynamic radial bearings may be used.

[•] **9.1.3.5** If specified, separable threaded solid-bar thermowells shall be supplied for temperature gauges.

9.1.3.6 Inducers, impellers, and similar major rotating components shall be dynamically balanced to ISO 21940–11, Grade G2.5, or to a residual unbalance of 0.01 oz in. (7 g mm), whichever is greater.

9.1.3.6.1 If possible, the mass of the arbor used for balancing shall not exceed the mass of the component being balanced.

9.1.3.6.2 The resulting vibration measured during the performance test shall not exceed the levels in Table 8.

9.1.3.7 Pumps shall be furnished by the vendor on a rigid mounting plate or baseplate, in accordance with 7.4 requirements.

9.1.3.8 Vendor's standard gearbox is acceptable and shall meet the following requirements.

9.1.3.8.1 The gearbox maximum power rating shall not be less than 125 % of the pump power at rated point.

9.1.3.8.2 In addition to the vibration requirements in 6.9.4 of this specification, the gearbox casing vibration shall not exceed 7G's RMS.

9.1.3.9 Couplings meeting the requirement of ANSI/AGMA 9000 class 8 is acceptable and coupling component balancing is not required.

9.1.3.10 Rotor balancing for OH6 pumps is not required. Component balancing is sufficient.

9.2 Between-bearings Pumps (Types BB1, BB2, BB3, and BB5)

9.2.1 Pressure Casings

9.2.1.1 Axially split casings may have a composition sheet gasket or a metal-to-metal joint; the vendor's bid shall state which is being offered.

9.2.1.2 Pumps for service temperatures below 300 °F (150 °C) may be foot-mounted.

9.2.1.3 For pumps with axially split casings, lifting lugs or tapped holes for eyebolts shall be provided for lifting only the top half of the casing and shall be so tagged. Methods for lifting the assembled machine shall be specified by the vendor [see Annex L].

[•] **9.2.1.4** If specified, proposed connection designs shall be submitted to the purchaser for approval before fabrication. The drawing shall show weld designs, size, materials, and pre-weld and postweld heat treatments.

[•] **9.2.1.5** If specified, an acoustic analysis of the crossover passages of BB3 and BB5 pumps shall be performed and reported to the purchaser.

9.2.1.5.1 The acoustic report shall document the lengths of crossover passages within the pump casing, the sound velocity in the pumped fluid at operating temperature and pressure, the calculated half-wave frequencies corresponding to crossover passage lengths and sound velocity, and the vane-pass frequencies produced by the impeller-casing design of the pump.

9.2.1.5.2 Acoustical resonance responses shall be evaluated whenever one or more half-wave frequency in any crossover passage is sympathetic to a vane-pass frequency of the pump.

[•] **9.2.1.6** If specified, for BB3 and BB5 pumps, the casing shall have a provision for balance chamber pressure measurement or provision on the balance line for flow measurement.

9.2.2 Rotor

9.2.2.1 Impellers of multistage pumps shall be individually located along the shaft by a shoulder or captive split ring in the direction of normal hydraulic thrust.

9.2.2.2 Rotors with clearance-fit impellers shall have mechanical means to limit impeller movement in the direction opposite to normal hydraulic thrust to 0.030 in. (0.75 mm) or less.

[•] **9.2.2.3** If specified, rotors with shrink-fit impellers shall have mechanical means to limit movement in the direction opposite to normal hydraulic thrust to 0.030 in. (0.75 mm) or less.

9.2.2.4 The runout of shafts and assembled rotors measured with the shaft or rotor supported on V-blocks or bench rollers adjacent to its bearings shall be within the limits given in Table 17.

9.2.3 Running Clearances

9.2.3.1 Renewable casing bushings and interstage sleeves or the equivalent shall be provided at all interstage points.

9.2.3.2 Running clearances associated with components used to balance axial thrust or to serve as product-lubricated internal bearings may be the manufacturer's standard, provided these clearances are stated as exceptions to this standard (see 6.7.7) in the proposal and are approved by the purchaser. If the manufacturer's standard clearances are based on material combinations exhibiting superior wear characteristics, supporting data shall be included in the proposal.

Table 17—Shaft and Rotor Runout Requirements

Flexibility Factor F_f [a, b] in.^{2} (mm^2)	Allowable Shaft Runout TIR in. (μm)	Component Fit on Shaft	Allowable Rotor Radial Runout TIR [c] in. (μm)
> 3 × 10^6 (1.9 × 10^9)	0.0015 (40)	Clearance	0.0035 (90)
		Interference	0.0025 (60)
≤ 3 × 10^6 (1.9 × 10^9)	0.0010 (25)	Clearance	0.0030 (75)
		Interference	0.0020 (50)

[a] $F_f = L^4/D^2$ where L is the bearing span; D is the shaft diameter (largest) at impeller.

[b] The shaft flexibility factor, F_f, is directly related to the static deflection of a simply supported shaft and is, therefore, a good indicator of the runout attainable during manufacture and the quality of balance that can be achieved and maintained.

[c] Runout of impeller hubs, balancing drum, and sleeves.

9.2.4 Dynamics

9.2.4.1 Lateral Analysis

9.2.4.1.1 The need for a lateral analysis of a pump's rotor shall be determined using the decision logic set out in Annex I, Table I.1.

[•] **9.2.4.1.2** If a lateral analysis is required, or if specified by the purchaser, it shall be performed and assessed in accordance with Annex I.

9.2.4.2 Rotor Balancing

9.2.4.2.1 Rotors of the categories listed below shall be two-plane dynamically balanced at low speed to the balance grade in Table 18:

a) multistage pumps (three or more stages);

b) one- and two-stage pumps whose maximum continuous speed is greater than 3800 r/min.

9.2.4.2.2 The sequence of rotor assembly and balance correction shall follow ISO 21940-12. For balancing, the rotor does not include the pump half-coupling hub or the rotary units of the mechanical seals. Refer to Table 18.

NOTE 1 Table 18 shows ISO 21940–12, Grade G2.5 for all interference fit rotors to speeds of 3800 r/min. This is based on two factors:

— at 3800 r/min, the upper limit of balance Grade G2.5 produces a force due to unbalance of 10 % of rotor weight, which means that unbalance does not have any material effect on the rotor's operating shape;

— for rotors whose flexibility is high (see Table 17), it is not practical to achieve and maintain the rotor straightness necessary for balance Grade G1.

NOTE 2 The mass eccentricity associated with balance Grade G1 is very small, e.g. 0.00010 in. (2.5 µm) maximum for operation at 3800 r/min. This has two consequences:

— it is not practical to balance the components to better than G2.5 (see 6.9.3.1.3) because the arbor effectively changes when the component is mounted;

— the balance quality might not be verifiable if the rotor is disturbed from its position on the balancing stand or disassembled and reassembled. It is normally possible, however, to perform a residual unbalance check to verify the accuracy of the balancing stand.

Table 18—Rotor Balance Requirements

Component Fit on Shaft	Maximum Continuous Speed r/min	Flexibility Factor L^4/D^2 in.^{2} (mm^2)	Rotor Balance Procedure(s) [b]	Rotor Balance Grade
Clearance	≤ 3800 [a]	No limit	C	[c]
Interference	≤ 3800	No limit	B+C; or C+D; or C+E	G2.5/(16W/N) [d]
	> 3800	≤ 3 × 10^6 (1.9 ×10^9)	B+C; or C+D; or C+E	G0.67/(4W/N) [d], [e]

NOTE See Table 17 for shaft and rotor runout requirements.

[a] To allow for 5 % speed increase.
[b] See ISO 21940-12
[c] Balance correction during assembly is not feasible because clearance fit does not maintain corrected balance.
[d] Approximately equal to the midpoint of the corresponding ISO balance quality grade.
[e] If rotors of higher flexibility are used at speeds above 3800 r/min, achieving and maintaining this balance level requires special attention to design, manufacture, and maintenance.

9.2.4.2.3 For rotor balancing, any vacant single keyways shall be filled completely with crowned half keys.

9.2.4.2.4 If a rotor is balanced as an assembly, a residual unbalance test shall be performed.

9.2.4.2.4.1 The check shall be performed after final balancing of the rotor, following the procedure given in Annex J.

9.2.4.2.4.2 The mass and location of all half keys used during final balancing of the assembled rotor shall be recorded on the residual unbalance worksheet as part of the "rotor sketch" or separately sketched and recorded on an attachment to the Annex J worksheet.

9.2.5 Bearings and Bearing Housings

9.2.5.1 If supplied, hydrodynamic radial bearings shall be in accordance with 9.2.5.1.1 through 9.2.5.1.4.

9.2.5.1.1 Bearings shall be split for ease of assembly, precision-bored, and of the sleeve or pad type, with steel-backed, babbitted replaceable liners, pads, or shells.

9.2.5.1.2 The bearings shall be equipped with anti-rotation pins and shall be positively secured in the axial direction.

9.2.5.1.3 The liners, pads, or shells shall be in axially split housings and shall be replaceable without having to dismantle any portion of the casing or remove the coupling hub.

9.2.5.1.4 Bearings shall be designed to prevent installation backward or upside down or both.

9.2.5.1.5 If the shaft contains more than 1.0 % chromium and the journal surface speed is above 65 ft/s (20 m/s), the shaft's journal shall be hard-chromium-plated, hard-coated, or sleeved with carbon steel, to avoid damage to the bearing journal from wire-wooling.

9.2.5.2 Hydrodynamic thrust bearings shall be in accordance with 9.2.5.2.1 through 9.2.5.2.6 below.

9.2.5.2.1 Thrust bearings shall be of the steel-backed, babbitted multiple-segment type, designed for equal thrust capacity in both axial directions and arranged for continuous, pressurized lubrication to each side.

9.2.5.2.2 Both sides shall be of the tilting-pad type, incorporating a self-levelling feature that assures that each pad carries an equal share of the thrust load with minor variation in pad thickness.

9.2.5.2.3 Thrust collars shall be replaceable and shall be mounted to the shaft with an interference fit to prevent fretting and positively locked to prevent axial movement.

9.2.5.2.4 Both faces of thrust collars shall have a surface roughness of not more than 16 µin. (0.4 µm) Ra, and, after mounting, the axial TIR of either face shall not exceed 0.0005 in. (13 µm).

9.2.5.2.5 Thrust bearings shall be sized for the maximum applied load (see 6.10.1.5). At this load, and the corresponding rotational speed, the following parameters shall be met:

a) minimum oil-film thickness of 0.0003 in. (8 µm);

b) maximum unit pressure (load divided by area) of 500 psi (35 bar, 3500 kPa);

c) maximum calculated babbitt surface temperature of 265 °F (130 °C).

NOTE The limits given above correspond to a design factor of two or more, based on the bearing's ultimate capacity. The calculated Babbitt surface temperature is a design value and is not representative of actual babbitt temperatures under these conditions.

9.2.5.2.6 Thrust bearings shall be arranged to allow axial positioning of each rotor relative to the casing and the setting of the bearing's clearance or preload.

[•] **9.2.5.2.7** If specified, thrust-bearing design and sizing shall be reviewed and approved by the purchaser.

9.2.5.3 If the inlet oil temperature exceeds 120 °F (50 °C), special consideration shall be given to bearing design, oil flow, and allowable temperature rise.

9.2.5.4 Oil outlets from thrust bearings shall be as recommended by the bearing manufacturer for the collar speed and lubrication method involved.

9.2.5.5 Oil connections on bearing housings shall be in accordance with 7.6.

9.2.5.6 Axially split bearing housings shall have a metal-to-metal split joint whose halves are located by means of cylindrical dowels.

9.2.5.7 Bearing housings for multistage pumps shall be doweled after verification of seal chamber runouts.

9.2.6 Lubrication

[•] **9.2.6.1** If specified, or if recommended by the vendor and approved by the purchaser, a pressure-lubrication system shall be furnished to supply oil at a suitable pressure to the pump bearings, the driver, and any other driven equipment, including gears.

9.2.6.2 External pressure-lubrication systems shall comply with the requirements of API 614. The minimum acceptable system for equipment furnished to this standard shall meet Oil System Codes LO-PRAA0-R1-H0-BP0-CS1-F2-A0-PV1-TV1-OT0 for oil systems mounted on pump baseplates and LO-PRAA0-R1-H0-BP1-CS1-F2-A0-PV1-TV1-OT0 for oil systems mounted as stand-alone consoles, as defined in Table 1 of API 614. For such a lubrication system, data sheets shall be supplied.

9.2.6.3 If oil is supplied from a common system to two or more machines (such as a pump, a gear, and a motor), the oil characteristics shall be suitable for all equipment supplied. The vendor having unit responsibility shall obtain approval from the purchaser and the other equipment vendors for the oil selection.

9.2.7 Testing

9.2.7.1 For pressure-lubricated bearings, test stand oil and oil system components downstream of the filters shall meet the cleanliness requirements specified in API 614.

9.2.7.2 All purchased vibration probes, transducers, and oscillator-demodulators shall be in use during the test. If vibration probes are not furnished by the vendor or if the purchased probes are not compatible with shop readout facilities, shop probes and readouts that meet the accuracy requirements of API 670 shall be used. The vibration measured with this instrumentation shall be the basis for acceptance or rejection of the pump.

9.2.7.3 With the purchaser's approval, single-stage, double-suction pumps may be assembled for testing by driving from the opposite end of the pump when compared with the general arrangement for the contract pump and driver. No retest is required after final assembly. If such an arrangement is required, it shall be stated in the proposal.

[•] **9.2.7.4** If specified, hydrodynamic bearings shall be removed, inspected by the purchaser or their representative, and reassembled after the performance test is completed. No retest is required after final assembly.

9.2.8 Preparation for Shipment

9.2.8.1 If a spare rotor or bundle is purchased, it shall be prepared for unheated indoor storage for three years.

9.2.8.1.1 Storage preparation shall include treatment with a rust preventive and enclosure in a vapor-barrier envelope with slow-release vapor-phase inhibitor.

9.2.8.1.2 The rotor or bundle shall be boxed for the same type of shipment specified for the pump.

9.2.8.1.3 A rotor shall have a resilient material [but not lead, tetrafluoroethylene (TFE), or PTFE], at least 0.12 in. (3 mm) thick, between the rotor and its support cradle; support shall not be at the rotor's journals.

9.2.8.1.4 A bundle shall have its rotor secured to prevent movement within the stator.

[•] **9.2.8.2** If specified, spare rotors and cartridge-type bundles shall be prepared for vertical storage.

9.2.8.2.1 A rotor shall be supported from its coupling end with a fixture designed to support 1.5 times the rotor's mass without damaging the shaft.

9.2.8.2.2 A cartridge-type bundle shall be supported from the casing cover (with the rotor hanging from its thrust bearing).

[•] **9.2.8.3** If specified, a shipping and storage container designed to store the rotor or cartridge vertically shall be provided.

[•] **9.2.8.4** If specified, the shipping and storage container shall be designed to allow inert-gas inhibition during storage.

9.3 Vertically Suspended Pumps (Types VS1 Through VS7)

9.3.1 General

9.3.1.1 Specified discharge pressure shall be at the purchaser discharge connection.

9.3.1.1.1 Hydraulic performance shall be corrected for column static and friction head losses.

9.3.1.1.2 Bowl or pump casing performance curves shall be furnished with the correction indicated.

9.3.1.2 It is not necessary that bearing housings for vertically suspended pumps be arranged so that bearings can be replaced without disturbing pump drives or mountings.

9.3.1.3 A minimum of four alignment-positioning screws shall be provided for each driver to facilitate horizontal adjustment for shaft alignment.

9.3.1.4 Driver supports of vertically suspended pumps that use thrust bearings in the driver shall be steel.

9.3.2 Pressure Casings

9.3.2.1 Jackscrews and casing alignment dowels are not required for rabbeted bowl assemblies.

9.3.2.2 Pumps shall be provided with vent connections for suction barrels and seal chambers.

9.3.2.3 Assemblies designed for O-ring seals only do not require flanges and bolting designed to seat a spiral-wound gasket (see 6.2.16).

9.3.2.4 Suction barrels or cans shall be designed with elliptical (of either ellipsoidal or torispherical shapes) heads of required wall thickness for MAWP to be within tensile stress limits (see 6.2.5) to facilitate weld inspection by radiography.

9.3.2.5 All welds of suction barrels shall be full-penetration welds (see 6.12.3.3).

9.3.2.6 All welds used to fabricate pressure casing walls of pump heads and suction barrels shall be inspected by radiography, with 100 % coverage in accordance with 8.2.1.6, prior to component fabrication.

[•] **9.3.2.7** If specified, pressure casing walls of pump heads and suction barrels or cans shall be fabricated of seamless pipe in accordance with ASTM A106, ASTM A53, ISO 3183, or API 5L material specifications.

9.3.2.8 For VS pumps with bowl OD of 8 in. (200 mm) or smaller, 0.31 in. (8 mm) fasteners are acceptable.

9.3.2.9 For VS1, VS2, and VS3 type pumps designed to be installed in pressure vessels or tanks, the component that constitutes the pressure casing is the discharge head.

9.3.2.10 For VS6 and VS7 type pumps, the components that constitute the pressure casing are the discharge head and the suction can.

9.3.3 Rotors

9.3.3.1 All pump shafts shall be machined or ground and finished throughout their entire length.

9.3.3.2 The shaft TIR shall not exceed 0.0005 in./ft (40 µm/m) of length.

9.3.3.3 The shaft TIR runout shall be inspected at lengths of 4 ft (1.2 m) or less.

9.3.3.4 The pump shaft shall be in one piece unless otherwise approved by the purchaser (because of total shaft length or shipping restrictions).

9.3.3.5 For VS pumps, assembly balancing of the rotor is not required. Individual part balancing will meet the balancing requirement.

9.3.4 Wear Parts and Running Clearances

9.3.4.1 Renewable casing bushings shall be provided at all interstage and other bushing locations.

9.3.4.2 The running clearances specified in 6.7.7 do not apply to the clearances of bushings. The clearances used shall be stated in the proposal and approved by the purchaser.

9.3.4.3 Pumps with semi-open impellers in an erosive service shall have a replaceable casing liner.

9.3.5 Dynamics

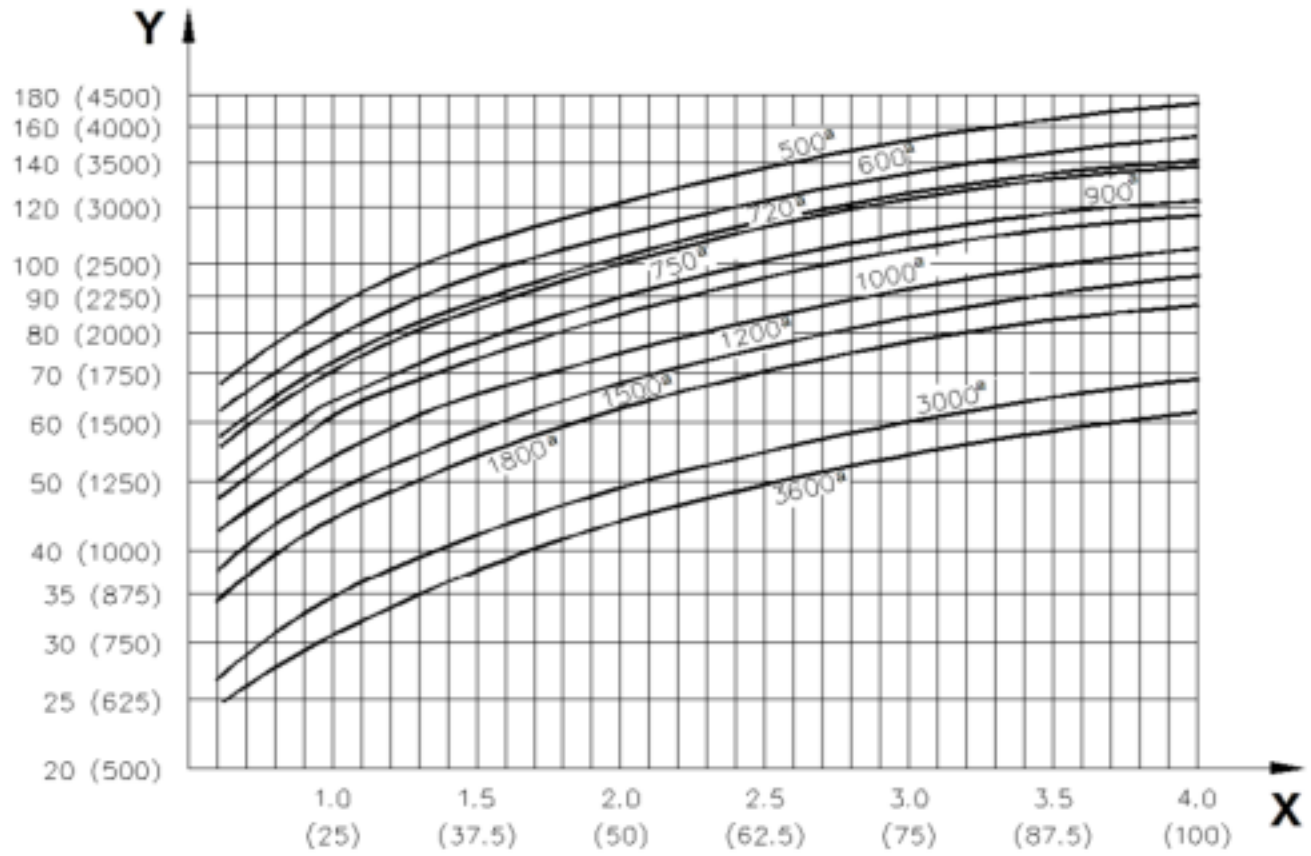
[•] If specified, the vendor shall furnish a dynamic analysis of the complete pump (which includes belowground components) and driver structure (including up to the 3rd column bending modes) on its foundation or support structure to confirm acceptability of the design. The purchaser and the vendor shall agree on the extent, method, and acceptance criteria for this analysis.

NOTE 1 Generally, an acceptance criteria of 20 % above and 15 % below running speed margin of separation is maintained between all the natural frequencies of the complete pump and driver structure and the rotational speed. If natural frequencies cannot be shifted to meet separation margins, then a forced response analysis can be conducted to assess vibration levels and determine the suitability of operating on or near a structural natural frequency. Levels of damping and excitation forces will then be agreed upon by both the vendor and the purchaser.

NOTE 2 Shaft lateral analysis is not necessary for vertically suspended pumps because shaft bearing spacing is covered by Figure 43. Detailed guidelines for the application and methodology of structural dynamic analyses to vertically suspended pumps are available from HI 9.6.8. API 684 can also be referenced for further guidance.

9.3.6 Bushings and Bearings

9.3.6.1 Bushings shall be suitably corrosion-resistant and abrasion-resistant for the specified product and temperature. The maximum spacing between shaft bushings shall be in accordance with Figure 43 to maintain the first critical speed above the maximum allowable continuous speed.



Key

X shaft diameter, expressed in inches (millimeters)

Y maximum bushing spacing, expressed in inches (millimeters)

[a] curves for various rotational speeds, expressed in revolutions per minute

Figure 43—Maximum Spacing between Shaft Guide Bushings

9.3.6.2 Thrust bearings that are integral with the driver shall meet the requirements of 7.1.13.

9.3.6.3 Thrust bearings and housings integral with the pump shall meet the applicable requirements of 6.10.

9.3.6.4 To allow axial rotor adjustment and oil lubrication, the integral pump thrust bearing shall be mounted with an interference fit on a slide-fit, key-driven sleeve.

9.3.6.5 Pump Manufacturer shall calculate the maximum upthrust or downthrust to be imposed on the thrust bearing, and shall be responsible for proper sizing of the thrust bearing to accommodate these loads. These values shall be supplied to the purchaser.

9.3.6.6 Except for sump pumps of type VS4 or VS5, the first-stage impeller shall be located between bushings.

NOTE Although between-bushing first-stage impellers can result in superior rotor support, certain applications, such as for sumps, require superior suction performance and can benefit from an overhung first-stage impeller arrangement.

9.3.7 Lubrication

Bushings in vertical pumps are normally lubricated by the liquid pumped. Alternative methods of lubrication shall be proposed if the pumped liquid is not suitable.

9.3.8 Accessories

9.3.8.1 Drivers

9.3.8.1.1 Pumps and motor assemblies that can be damaged by reverse rotation shall be provided with a non-reverse ratchet or another purchaser-approved device to prevent reverse rotation.

9.3.8.1.2 Motors for vertical pumps shall have solid shafts. Motors shall meet the shaft and base tolerances shown in Figure 38.

9.3.8.1.3 Drivers shall be aligned and matchmarked or doweled in the vendor's shop prior to shipment.

9.3.8.2 Couplings and Guards

9.3.8.2.1 Coupling faces shall be perpendicular to the axis of the coupling within 0.0001 in./in. (0.1 µm/mm) of face diameter or 0.0005 in. (13 µm) TIR, whichever is greater.

9.3.8.2.2 For vertical pumps equipped with mechanical seals and without integral thrust bearings, the coupling shall be a rigid adjustable-type type with spacer length to permit replacement of the seal assembly, including the sleeve, without removal of the driver.

9.3.8.2.3 For vertical pumps equipped with mechanical seals and with integral thrust bearings, the coupling arrangement shall have spacer length to permit replacement of the seal assembly, including the sleeve, without removal of the driver. One of the following arrangements may be provided:

a) an all-metal flexible element, spacer-type coupling located between the thrust bearing and driver shafts, and no coupling between the mechanical seal and the thrust bearing housing;

b) a rigid adjustable, spacer-type coupling between the pump and thrust bearing housing shafts, and an all-metal flexible element, non-spacer-type coupling between the thrust bearing and driver shafts (see Figure 44).

NOTE The second arrangement with two couplings enables seal replacement without disturbing and lifting the thrust bearing housing. The size and weight of the thrust bearing housing assembly is a consideration in selecting one of these arrangements.

9.3.8.2.4 For vertical pumps where the driver stand encloses the access to remove the bearing arrangement and seal, removal points within the driver stand and special tooling, as required, shall be provided to allow removal without disturbing the driver.

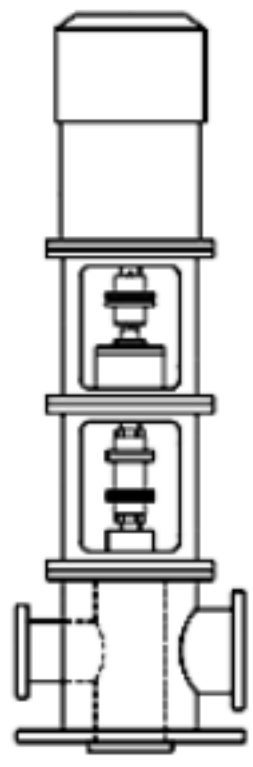


Figure 44—Thrust Bearing Arrangement with Two Couplings

9.3.8.3 Mounting Plates

[•] **9.3.8.3.1** If specified, the mounting plate for double-casing pumps (VS6 and VS7) shall be separate from the main body flange and located below it far enough to permit the use of through-bolting on the body flange (see Figure 45).

9.3.8.3.1.1 When separate from the main body flange, the mounting plate shall be continuously welded to the outer casing on both sides.

9.3.8.3.1.2 Contact areas between mounting plate and sole plate(s) shall be flat and perpendicular to the centerline of the shaft.

9.3.8.3.1.3 Mounting tolerances shall be in accordance with 7.4.13.

[•] **9.3.8.3.2** If specified, pumps shall be provided with a separate sole plate for bolting and grouting to the foundation (see Figure 45). This plate shall be machined on its top surface for mounting of the discharge head, can or motor support.

NOTE The design shown in Figure 45 results in a higher joint integrity and is often used for critical and cryogenic services.

9.3.8.3.3 The outside corners of the sole plate or mounting plate imbedded in the grout shall have at least 2 in. (50 mm) radii in the plan view (see Figure 42).

9.3.8.3.4 The sole plate shall be provided with a total of four removable jacking screws, each located adjacent to a holding down bolt hole.

9.3.8.3.5 Grout contact surfaces of the sole plate shall be sand-blasted and coated in accordance with the requirements of 7.4.20.

9.3.8.4 Piping and Appurtenances

If mechanical seals and drivers are not installed prior to shipment, the seal piping system shall not be fully assembled.

9.3.9 Testing

9.3.9.1 Pumps shall be tested as complete assemblies. Tests using only bowls and impellers are not recommended.

9.3.9.1.1 In cases where assembled-unit testing is impractical, the vendor shall submit alternative testing procedures with the proposal.

9.3.9.1.2 Suction cans, if supplied, are not required for performance testing.

[•] **9.3.9.2** If specified, a resonance test with the pump unpiped shall be conducted on the pump structure/driver frame assembly. The test shall be performed as follows:

a) excite the assembly by making an impact on the driver frame in the direction of the discharge flange;

b) determine the natural frequencies by the response;

c) excite the assembly by making an impact on the driver frame at 90° to the direction of the discharge flange;

d) determine the natural frequencies by the response.

e) The natural frequencies determined with this method shall be at least 10 % below the minimum continuous operating speed or shall be at least 10 % above the maximum continuous operating speed.

[•] **9.3.9.3** If specified, for vertical submerged wet pit pumps (VS1, VS2, VS3, VS4, VS5), a minimum submergence test shall be performed at 120 % flow, in addition to the NPSH required test, if the minimum submergence is more restrictive. The minimum submergence test at 120 % flow can be performed in lieu of NSPH required test, with purchasers' approval.

NOTE A minimum submergence test is more restrictive than an NPSH required test when the required submergence of a pump inlet bell, to avoid the occurrence of free-surface air vortices, is greater than the submergence needed to provide the required NPSH for the pump. Hydraulic Institute's Section 9.8 discusses vortex formation and model testing methods.

9.3.10 Single-case Diffuser (VS1) and Volute (VS2) Pumps

9.3.10.1 Line shafts may be open or enclosed. For enclosed line shafts, where the pumped liquid is not suitable as a lubricant, the type of lubrication shall be approved by the purchaser.

9.3.10.2 The discharge-head mounting surface shall be suitable for both grouting and mounting on a machined sole plate.

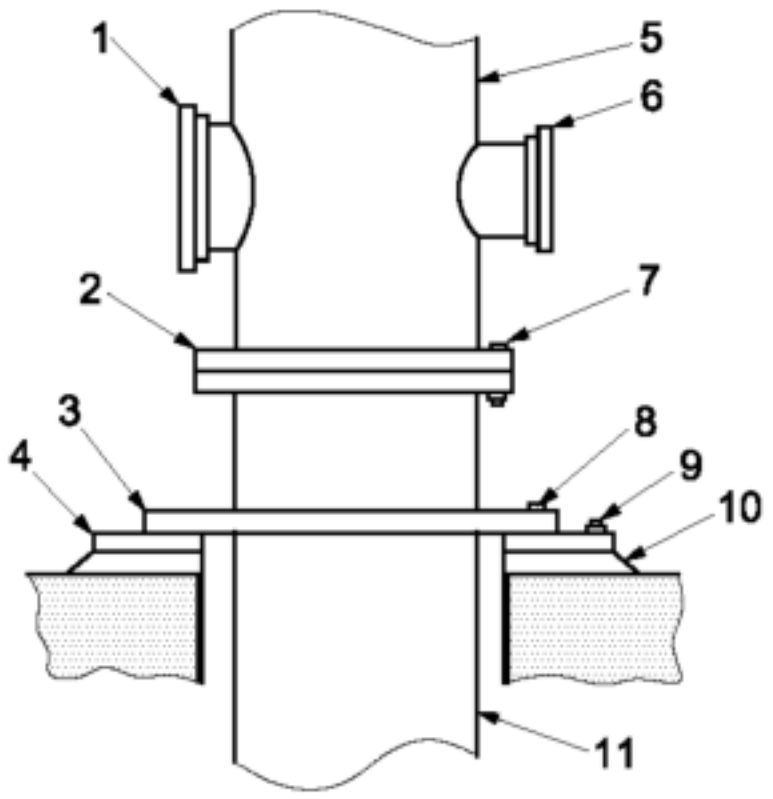
9.3.10.3 Thrust restraints are required at the pump if an expansion joint is installed on the discharge nozzle. Design review of the proposed installation and piping by the vendor is recommended.

[•] **9.3.10.4** If specified, bowl and line shaft bearings shall be furnished with hardened bearing journals under each bearing.

9.3.10.5 Integral bushing spiders and rabbeted fits shall be used for column sizes greater than or equal to 16 in. (400 mm).

9.3.10.6 Bowls shall be flanged and shall have metal-to-metal rabbeted fits.

9.3.10.7 Single-case diffuser (VS1) and volute (VS2) pump bowls shall have O-ring or gasketed seals.



Key

1	suction flange	7	main body flange through-bolting (typical)
2	main body flange	8	hold-down bolts (typical)
3	mounting flange	9	anchor bolts (typical)
4	sole plate	10	grout
5	pump head	11	can (outer casing)
6	discharge flange		

Figure 45—Optional Mounting for Vertically Suspended, Double-case Pumps (VS6 and VS7) with Sole Plate

9.3.11 Single-casing Axial Flow (VS3) Pumps

9.3.11.1 Integral bushing spiders and rabbeted fits shall be used for all column sizes.

9.3.11.2 Bowls shall have metal-to-metal rabbeted fits.

9.3.11.3 Single-casing axial flow (VS3) pump bowls shall have O-ring or gasketed seals.

9.3.12 Single-casing Line Shaft (VS4) and Cantilever (VS5) Pumps

9.3.12.1 For VS4 pumps, bushings shall be provided to support the shaft and impeller.

9.3.12.2 VS5 pumps shall comply with Items a) through d) as follows.

a) The rotor shall be cantilevered from its bearing assembly. Submerged bottom bushings are not used to guide the shaft.

b) The shaft stiffness shall limit total deflection, without the use of a casing bushing, such that the impeller does not contact the pump casing under the most severe dynamic conditions over the complete head-flow curve with a maximum diameter impeller and at the maximum speed and liquid density.

c) Cantilever type pumps shall have their first dry critical speed, for their rotors, 30 % above their maximum allowable continuous speed.

d) For cantilever-type VS5 pumps, the shaft TIR shall not exceed 0.002 in. (50 µm) as measured on the shaft directly above the mechanical seal or stuffing box.

9.3.12.3 The components that constitute the pressure casing of VS4 and VS5 pumps are the casing, suction cover, and discharge line.

[•] **9.3.12.4** The purchaser shall specify whether sump-pump services for VS4 and VS5 pumps are open-system or closed-system arrangements.

[•] **9.3.12.5** For closed-system arrangements with pressure-containing vessels or tanks, the purchaser shall specify the maximum pressure in the vessel or tank as the maximum suction pressure for the pumps.

9.3.12.6 For VS4 and VS5 pumps in closed-system arrangements, cover-plate joints shall be vapor-tight as a minimum.

9.3.12.7 For VS4 and VS5 pumps in closed-system arrangements with pressure-containing vessels or tanks, the seal chamber, pump cover plate and tank cover shall be designed to contain the maximum suction pressure specified. The cover-plate design and its mounting interface with the pump-mounting nozzle of the vessel or tank shall be agreed by the purchaser and vendor.

NOTE For closed-system arrangements with single small sump pumps, the purchaser and vendor typically agree to pump-mounting nozzles which conform to the pressure ratings and dimensional requirements of ASME B16.5 and ASME B16.47. Larger sump pumps may require special pump-mounting nozzles to accommodate the weight and size of the equipment.

9.3.12.8 For VS4 pumps, the thrust bearing shall be designed for either grease or oil lubrication. Bushings may be lubricated with water, grease or product, or be self-lubricated.

9.3.12.9 Bearings for VS5 pumps shall be grease-lubricated. The stabilized bearing-housing temperature shall not exceed a 70 °F (39 K) rise above ambient temperature.

9.3.12.10 Packing shall be supplied on VS4 and VS5 pumps except for closed-system services.

9.3.12.11 Mechanical seals, if supplied, shall be located at the cover plate, to seal the vapor in the supply tank or vessel.

9.3.12.11.1 Mechanical seals normally seal vapor; however, they shall be designed to operate in liquid in the event of tank or vessel overfilling.

9.3.12.11.2 The seal chamber shall have provisions for a high-point vent.

9.3.12.12 Lifting lugs shall be provided in the cover plate for lifting the pump assembly, including the driver.

9.3.12.13 The discharge nozzle and cover plate shall be designed as required in 6.2.5.

9.3.12.14 For pumps that are mounted in pressure vessels or tanks, the pump-mounting nozzle of the vessel or tank shall be required to withstand the allowable nozzle loads. Refer to 6.5 for allowable nozzle loads.

9.3.12.15 Pump-out vanes may be used in lieu of wear rings to reduce leakage back to the sump.

9.3.12.16 Typically, spacer couplings are not used on VS4 and VS5 pumps.

9.3.12.16.1 Coupling hubs shall be supplied with slip fits to the shaft.

9.3.12.16.2 The coupling hubs and keys shall be secured to the shaft with set-screws to facilitate final coupling adjustment.

9.3.13 Double-casing Diffuser (VS6) and Volute (VS7) Pumps

[•] **9.3.13.1** If specified, bowls and column pipe shall be hydrostatically tested with liquid at a minimum of 1.5 times the maximum differential pressure developed by the bowl assembly. Hydrostatic testing shall be conducted in accordance with the requirements of 8.3.2.

9.3.13.2 Complete outer-case venting shall be ensured by means of a high-point vent connection.

9.3.13.3 Provision shall be made to ensure complete venting of the inner assembly within the seal chamber or associated auxiliary process piping.

[•] **9.3.13.4** If specified, the suction shall be supplied with a drain piped to the surface.

9.3.13.5 Column sections shall incorporate integral bushing spiders and rabbeted fits for column sizes greater than or equal to 16 in. (400 mm).

[•] **9.3.13.6** If specified, bowl and line shaft bearings shall be furnished with hardened bearing journals under each bearing.

9.3.13.7 Double-casing diffuser (VS6) and volute (VS7) pump bowls shall have metal-to-metal rabbeted fits.

9.3.13.8 Double-casing diffuser (VS6) and volute (VS7) pump bowls shall have O-ring or gasketed seals.

10 Vendor's Data

[•] **10.1** The purchaser may specify the content of proposals, meeting frequency and vendor data content/format as described in Annex L. Annex L provides a general outline of information that potentially may be requested by the purchaser.

[•] **10.2** If specified, the information specified in Annex L shall be provided.

Annex A
(normative)

Specific Speed and Suction-specific Speed

Specific speed, n_s, is an index number relating to a pump's performance at BEP flowrate with the maximum diameter impeller and at a given rotational speed. Specific speed is defined by Equation (A.1):

$$n_s = n(q)^{0.5}/H^{0.75} \quad \text{(A.1)}$$

where

n the rotational speed, expressed in revolutions per minute;

q is the total pump flowrate, expressed in U.S. gallons per minute (cubic meters per second);

H is the head per stage, expressed in feet (meters).

NOTE 1 Specific speed derived using USC units divided by a factor of 51.64 is equal to specific speed in SI units.

NOTE 2 For simplicity, industry omits the gravitational constant from the dimensionless equations for specific speed and suction-specific speed.

An alternative definition of specific speed is sometimes used (flowrate per impeller eye rather than total flowrate). The purchaser is cautioned to understand which definition is being used when comparing data.

Suction-specific speed, *S*, an index number relating to a pump's suction performance, is calculated at BEP flowrate with the maximum diameter impeller at a given rotational speed and is defined by Equation (A.2):

$$S = n(q)^{0.5}/(NPSH3)^{0.75} \quad \text{(A.2)}$$

where

n is the rotational speed, expressed in revolutions per minute;

q is the flowrate per impeller eye, expressed in U.S. gallons per minute (cubic meters per second) equal to one of the following:

— total flowrate for single-suction impellers,

— one-half the total flowrate for double-suction impellers;

(NPSH3) is the net positive suction head required, expressed in feet (meters).

NOTE 3 Suction-specific speed derived using USC units divided by a factor of 51.64 is equal to suction-specific speed in SI units.

NOTE 4 The USC symbol N_{ss} is sometimes used to designate suction-specific speed.

Annex B
(normative)

Cooling Water Schematics

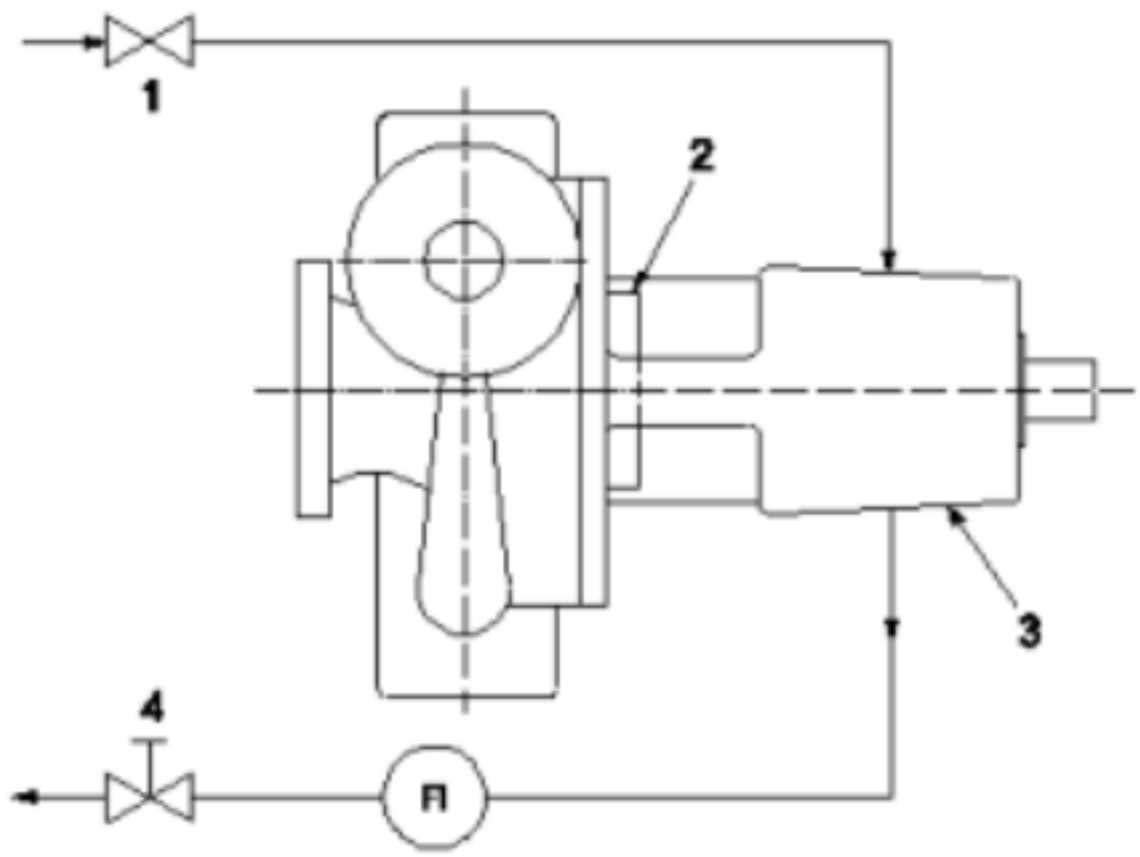
This annex contains representative schematic diagrams for cooling water systems. The symbols used in Figure B.2, Figure B.3, Figure B.4, Figure B.5, Figure B.6, Figure B.7, Figure B.8, and Figure B.9 are shown and identified in Figure B.1. These symbols represent commonly used systems. Other configurations and systems are available and may be used if specified or if agreed upon by the purchaser and the vendor.



Key

1	heat exchanger	11	flow-regulating valve
6	flowrate indicator	12	block valve (gate valve)

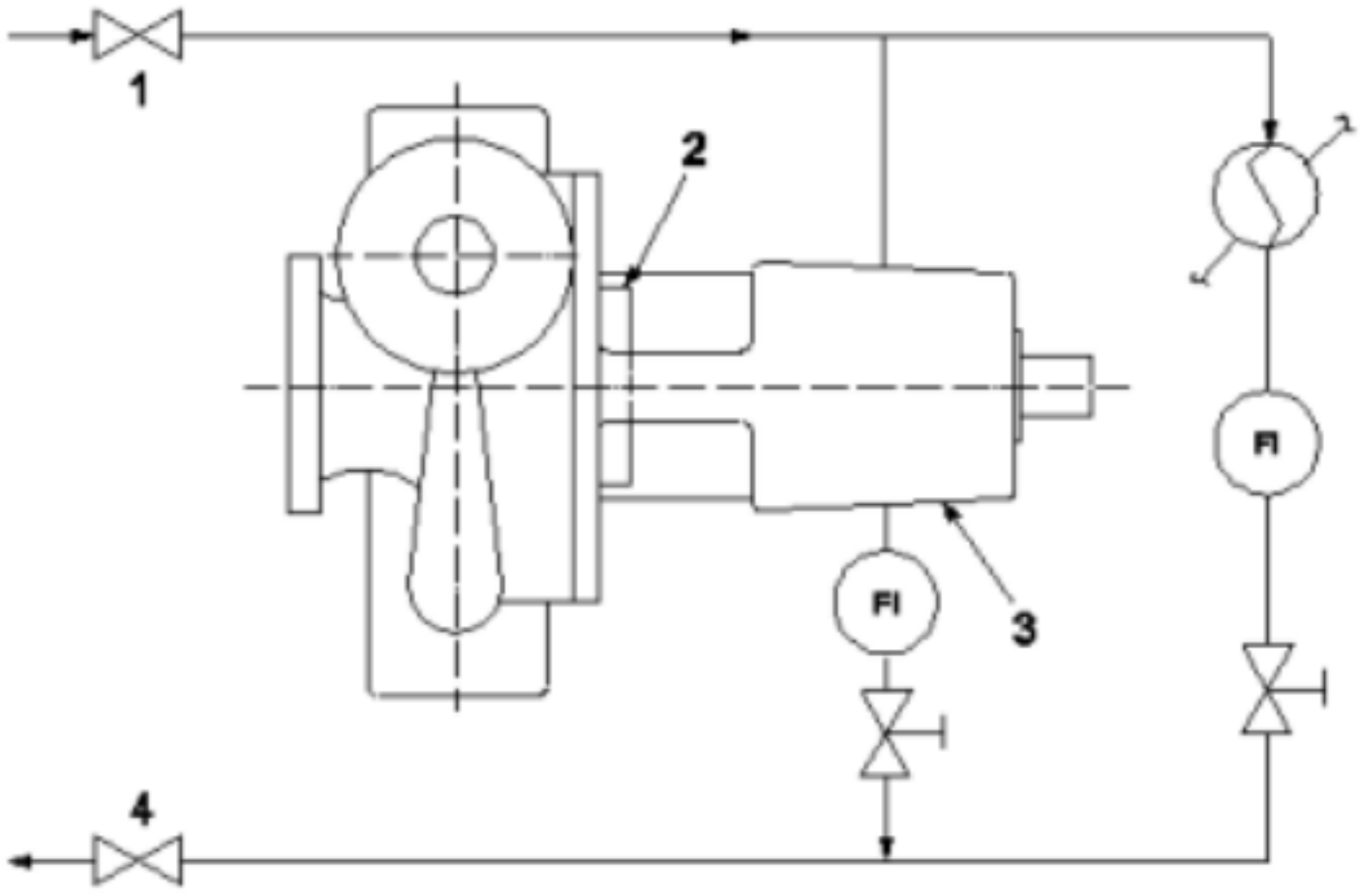
Figure B.1—Symbols Used in Figures B.2 through B.7



Key

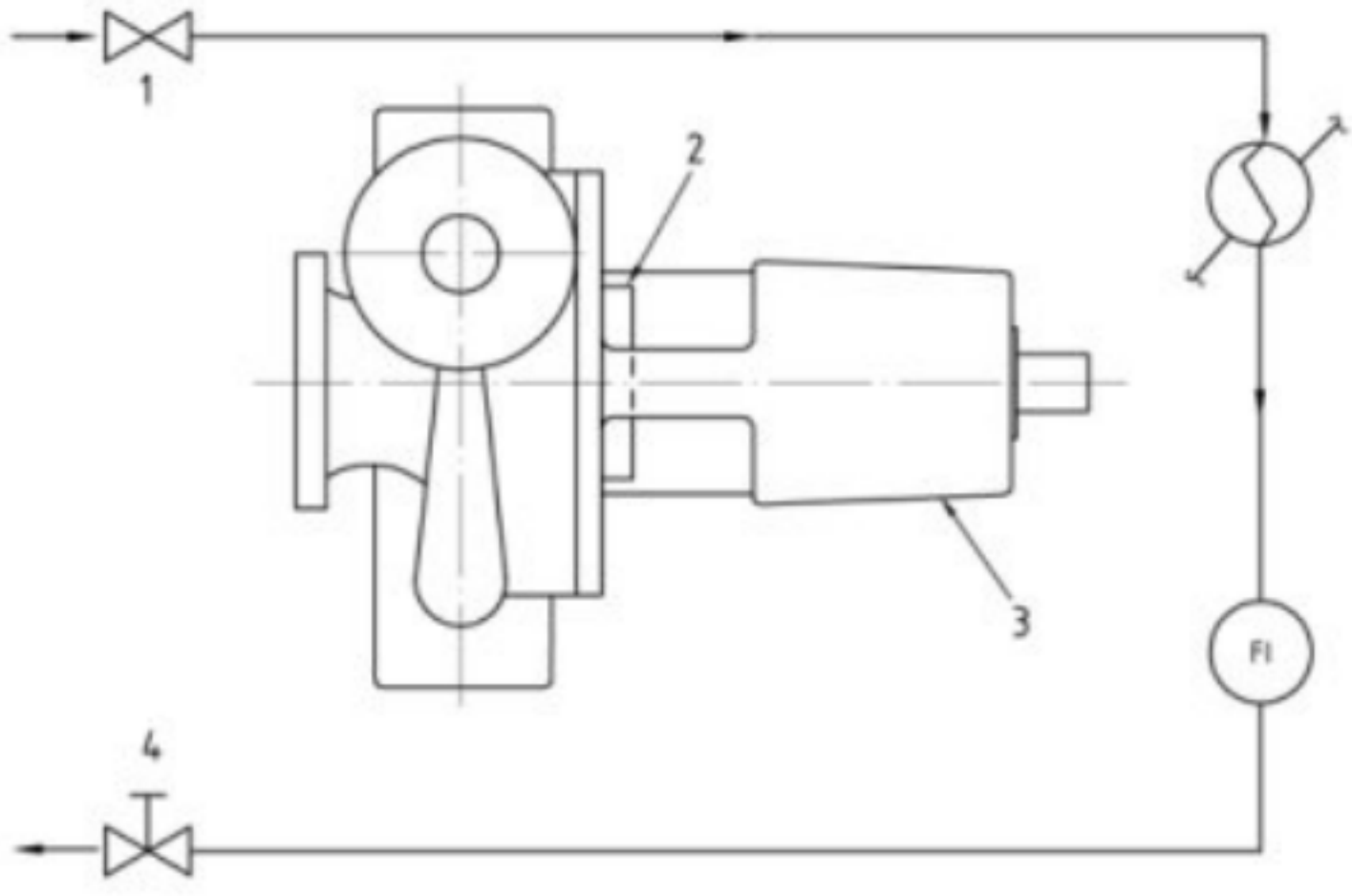
1 inlet valve
2 gland
3 bearing housing
4 exit valve

Figure B.2—Piping for Overhung Pumps—Plan A, Cooling to Bearing Housing



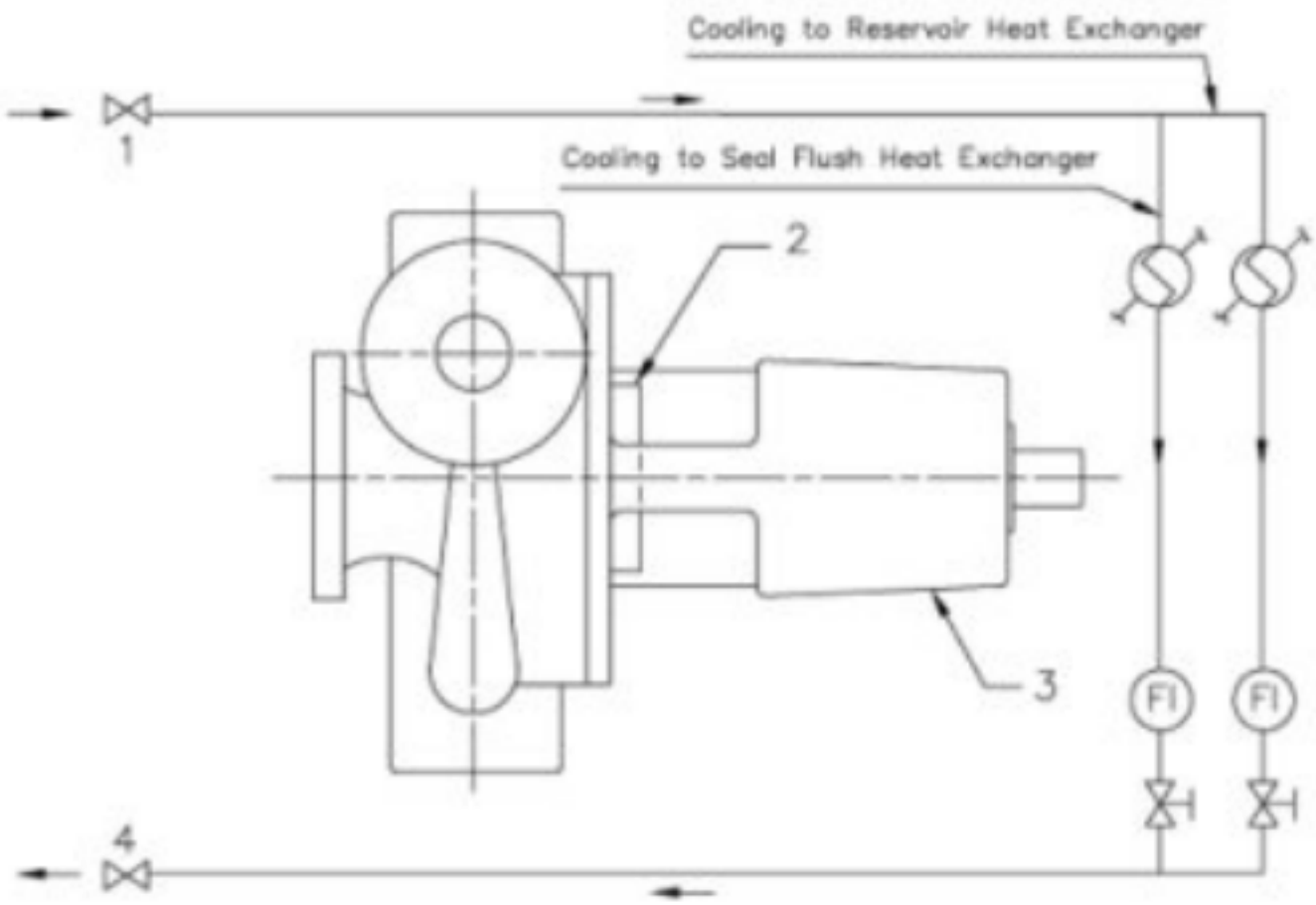
Key
1 inlet valve
2 gland
3 bearing housing
4 exit valve

Figure B.3—Piping for Overhung Pumps—Plan K, Cooling to Bearing Housing with Parallel Flow to Seal Heat Exchanger



Key
1 inlet valve
2 gland
3 bearing housing
4 exit valve

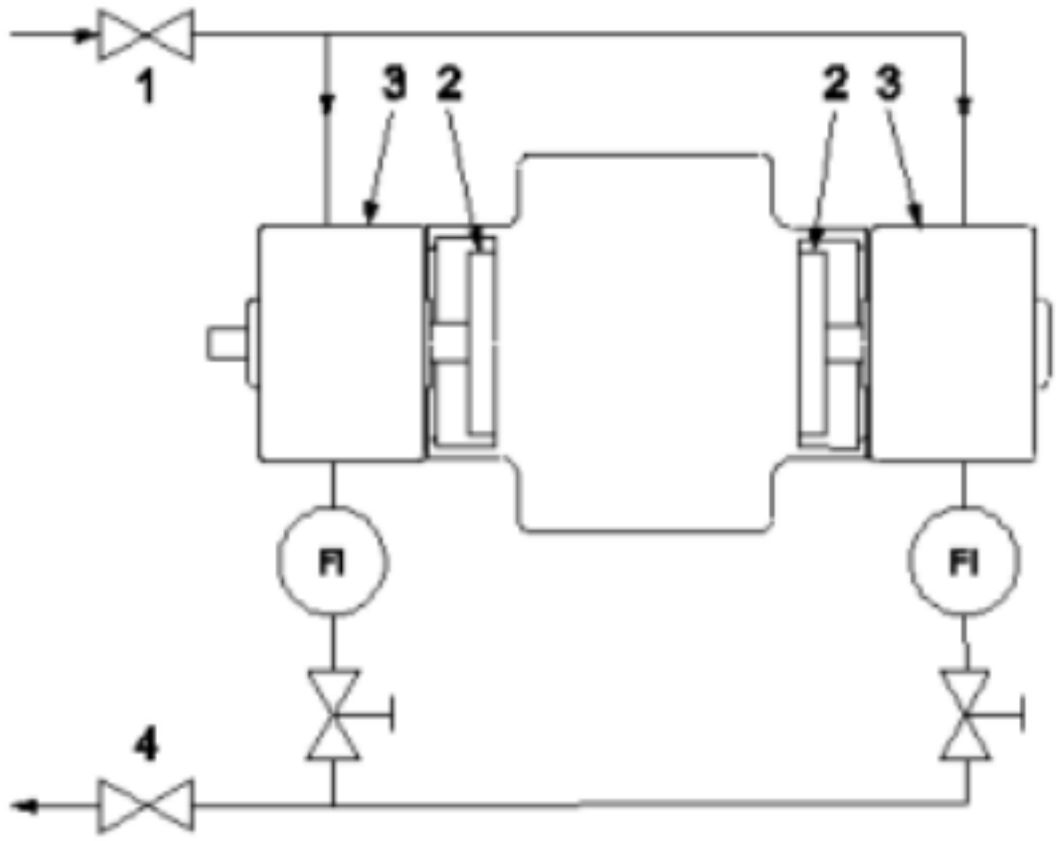
Figure B.4—Piping for Overhung Pumps—Plan M, Cooling to Seal Heat Exchanger



Key

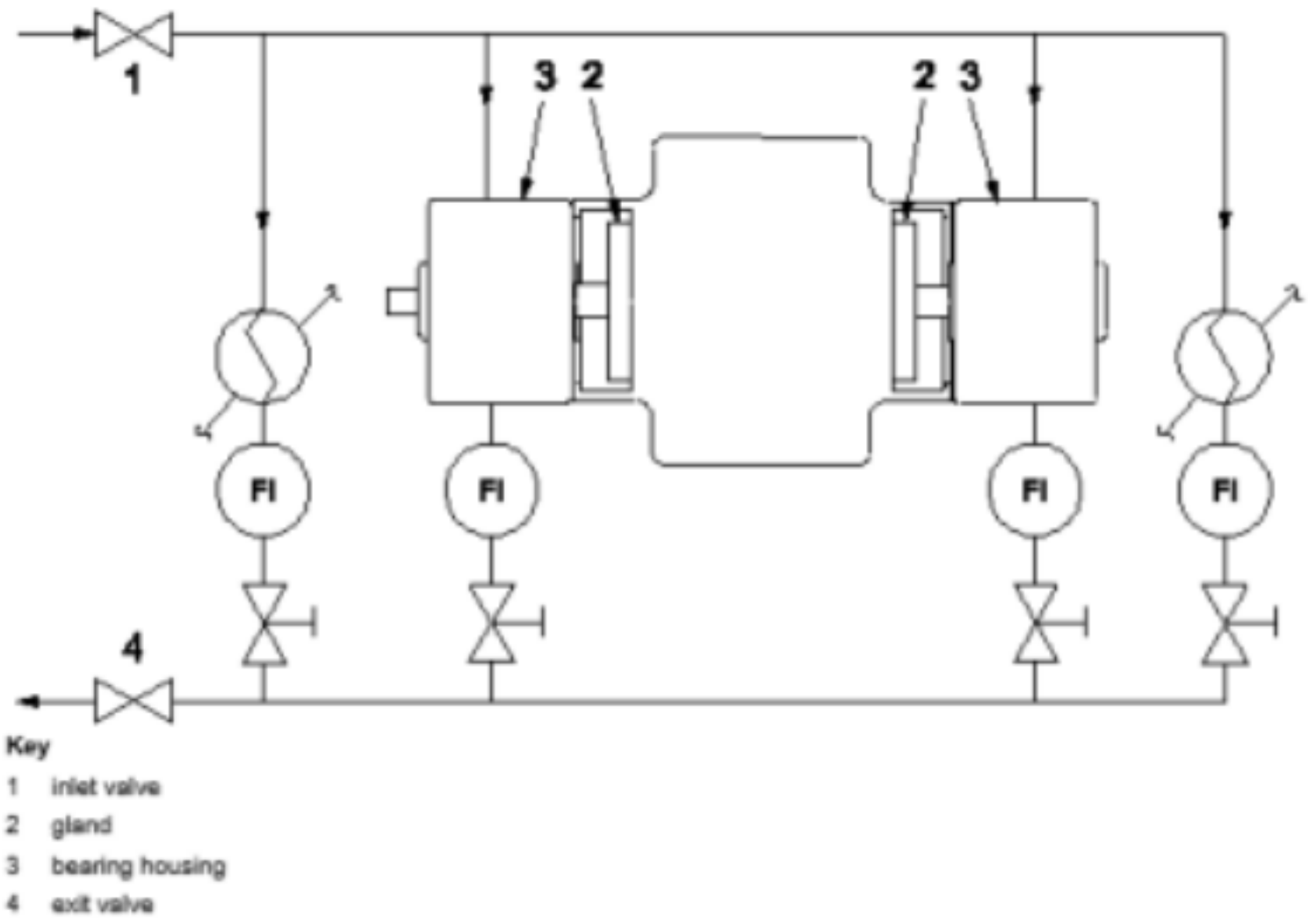
1 inlet valve
2 gland
3 bearing housing
4 exit valve

Figure B.5—Piping for Overhung Pumps—Plan M, Cooling to Seal Heat Exchanger and Reservoir



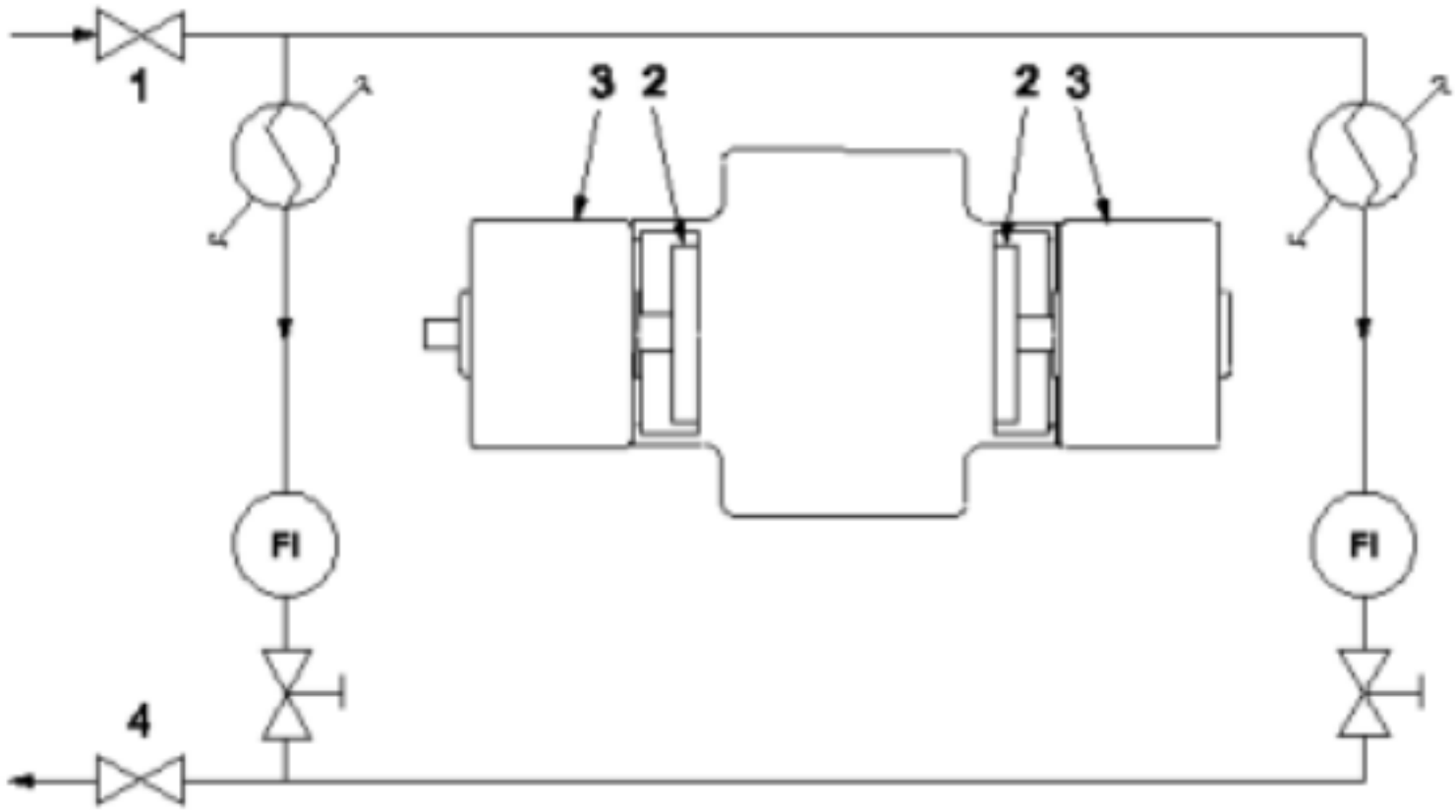
Key
1 inlet valve
2 gland
3 bearing housing
4 exit valve

Figure B.6—Piping for Between-bearing Pumps—Plan A, Cooling to Bearing Housings



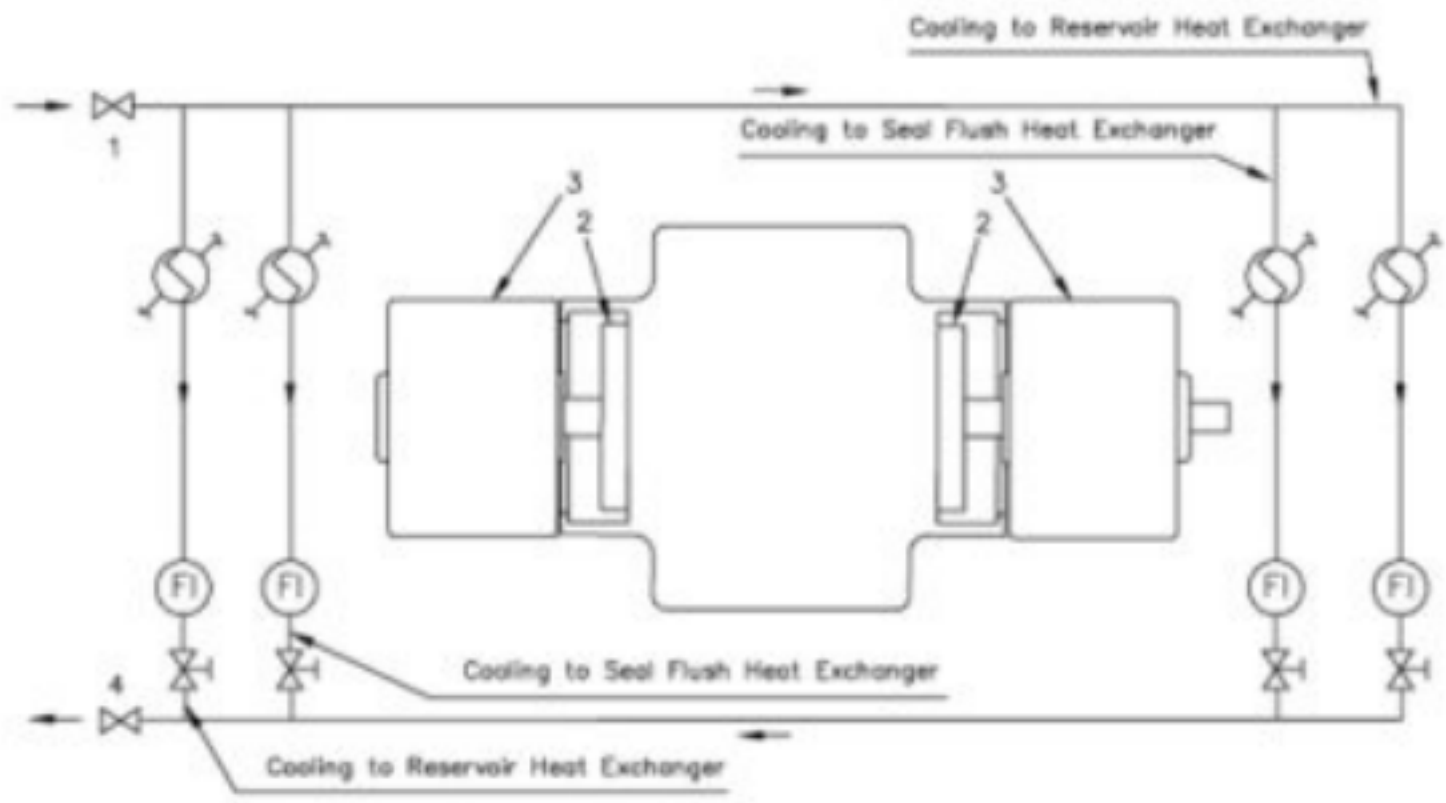
Key
1 inlet valve
2 gland
3 bearing housing
4 exit valve

Figure B.7—Piping for Between-bearing Pumps—Plan K, Cooling to Bearing Housings with Parallel Flow to Seal Heat Exchangers



Key
1 inlet valve
2 gland
3 bearing housing
4 exit valve

Figure B.8—Piping for Between-bearing Pumps—Plan M, Cooling to Seal Heat Exchangers



Key
1 inlet valve
2 gland
3 bearing housing
4 exit valve

Figure B.9—Piping for Between-bearing Pumps—Plan M, Cooling to Seal Heat Exchangers and Reservoirs

Annex C
(normative)

Hydraulic Power Recovery Turbines

C.1 General

This annex applies to hydraulic power recovery turbines (HPRTs).

Power recovery is generally achieved by the reduction of liquid pressure, sometimes with a contribution from vapor or gas evolution (i.e. 2-phase flow) during the pressure reduction. A HPRT may be a pump operated with reverse flow. Refer to Figure C.1 a) and Figure C.1b).

C.2 Terminology

This standard uses terms that need to be changed or ignored if the standard is applied to HPRTs. The direction of flow through the HPRT is the reverse of that through the pump. In such a context, the word “pump” may be interpreted as meaning “HPRT,” the term “pump suction” should be interpreted as meaning the “HPRT outlet,” and the term “pump discharge” should be interpreted as meaning the “HPRT inlet.”

C.3 Design

C.3.1 Liquid Characteristics

[•] **C.3.1.1** The purchaser shall advise the HPRT manufacturer whether any portion of the process stream entering the HPRT can flash to vapor and whether absorbed gas in the stream can evolve at any pressure less than the inlet pressure.

[•] **C.3.1.2** The purchaser shall specify the volume percentage of vapor or gas, or both, at the turbine outlet and the pressure and temperature at which the vapor can flash off.

[•] **C.3.1.3** If known, the liquid composition, and the liquid and vapor (or gas) density vs pressure, should also be specified. It can be necessary to control the HPRT outlet pressure to limit the amount of liquid that flashes to vapor or the amount of gas coming out of solution.

C.3.2 Seal-flushing System

To avoid shortening seal life, consideration shall be given to the evolution of gas and vaporization in seal-flushing streams. If this potential exists, a seal flush from other than the HPRT inlet is generally recommended.

C.3.3 Overspeed Trip

C.3.3.1 An overspeed trip shall be provided if the HPRT and other equipment in the train cannot tolerate the calculated runaway speed (the maximum speed reached by the HPRT if unloaded and subjected to the worst combination of specified inlet and outlet conditions). Typically, overspeed trips are set in the range of 115 % to 120 % of rated speed.

NOTE 1 It is important to realize that runaway speed with inlet liquids rich in absorbed gas or with liquids that partially flash as they flow through the HPRT can be several times higher than the runaway speed with water. With such liquids, the runaway speed cannot be accurately determined.

NOTE 2 The risk of overspeed is reduced if the driven equipment, such as a pump or fan, cannot realistically be expected to lose load. The risk is increased if the driven equipment is a generator, since a sudden disconnection from electric power circuits unloads the HPRT.

[•] **C.3.3.2** If specified, a quick-acting brake system to prevent overspeed due to accidental unloading shall be provided.

C.3.4 Dual Drivers

C.3.4.1 If an HPRT is used to assist another driver, the considerations in C.3.4.2 through C.3.4.5 apply.

C.3.4.2 The main driver shall be rated to drive the train without assistance from the HPRT.

C.3.4.3 An overrunning clutch (that is a clutch that transmits torque in one direction and freewheels in the other) shall be used between the HPRT and the train to allow the driven equipment to operate during HPRT maintenance and to permit start-up of the train before the HPRT process stream is lined up.

C.3.4.4 Flow to the HPRT can vary widely and frequently. If the flow drops to ~40 % of the rated flow, the HPRT stops producing power and a drag can be imposed on the main driver. An overrunning clutch prevents this drag.

C.3.4.5 The HPRT shall not be positioned between the main driver and the driven equipment.

C.3.5 Generators

If a generator is driven by a HPRT, the generator shall be sized to handle all predicted power levels output by the HPRT when operating on the actual process stream. Output power of HPRTs in gas-rich process streams can be as much as 20 % to 30 % or more above that predicted by water tests, as a result of the effects of evolved gas or flashed liquid.

NOTE See Figure C.1c.

C.3.6 Throttle Valves

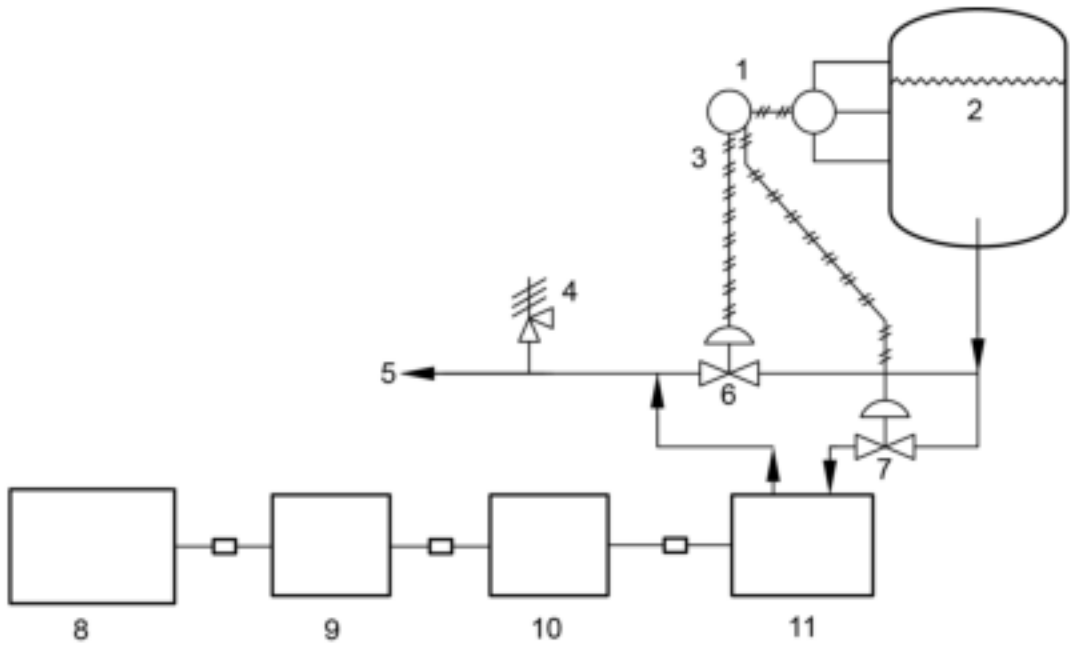
For most applications, valves used to control flow to the HPRT should be placed upstream and near the inlet of the HPRT (see Figure C.1). Placement upstream minimizes the pressures and severity of transients experienced by the turbine during throttling and shutdown actions and allows the mechanical seals to operate at the outlet pressure of the HPRT.

C.3.7 Bypass Valves

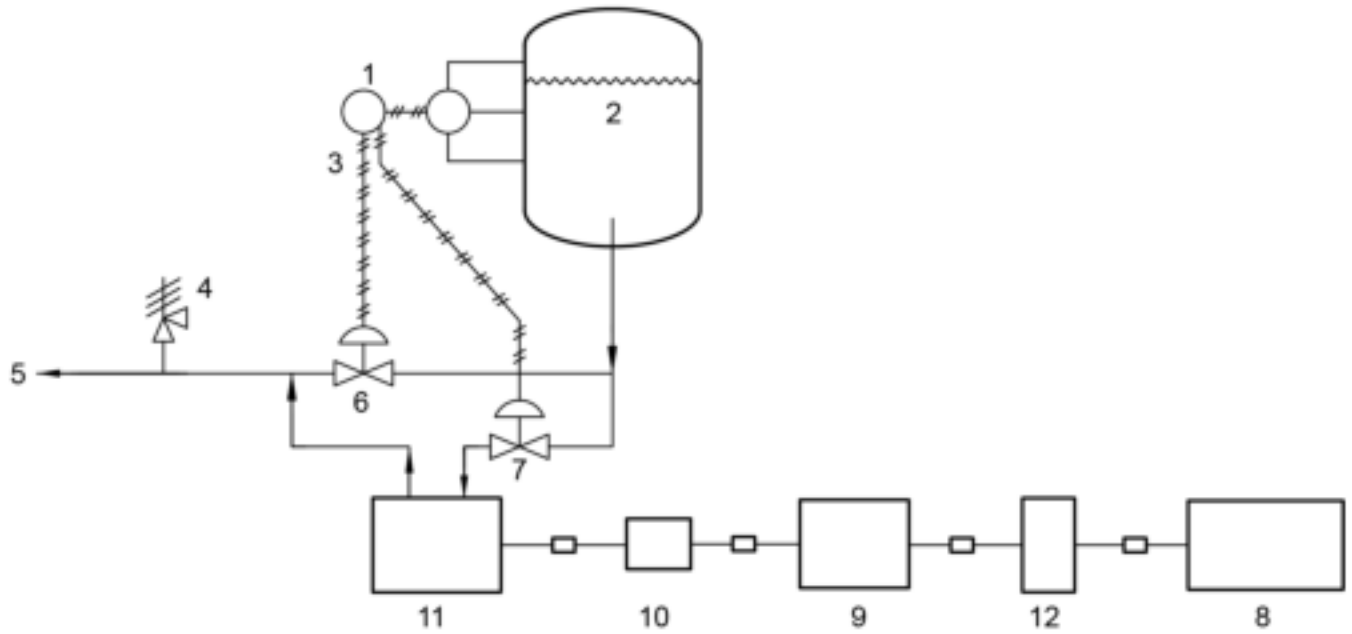
Regardless of the arrangement of the HPRT train, a full-flow bypass valve with modulation capability should be installed. Common control of the modulating bypass valve and the HPRT inlet control valve is normally achieved by means of a split-level arrangement (see Figure C.1).

C.3.8 Relief Valves

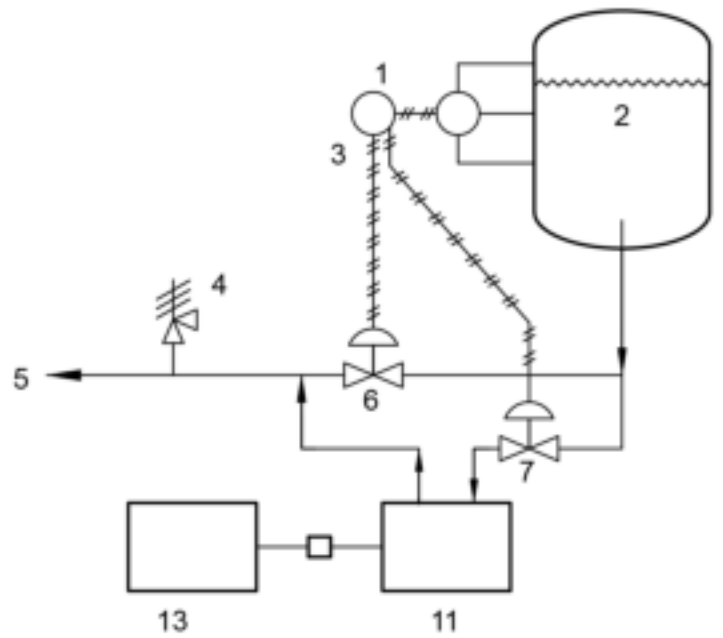
To protect the HPRT outlet casing integrity and mechanical seals from possible downstream back-pressure transients, a relief valve installed in the HPRT outlet piping circuit should be considered (see Figure C.1).



a) Pump Drive at Motor Speed



b) Pump Drive at Greater than Motor Speed



c) Generator Drive

Key

1 level indicator, controller
2 high-pressure source
3 split range
4 relief valve
5 low-pressure destination
6 bypass
7 inlet throttle valve
8 pump
9 motor
10 overrunning clutch
11 HPRT
12 gear
13 generator

Figure C.1—Typical HPRT Arrangements

C.4 Testing

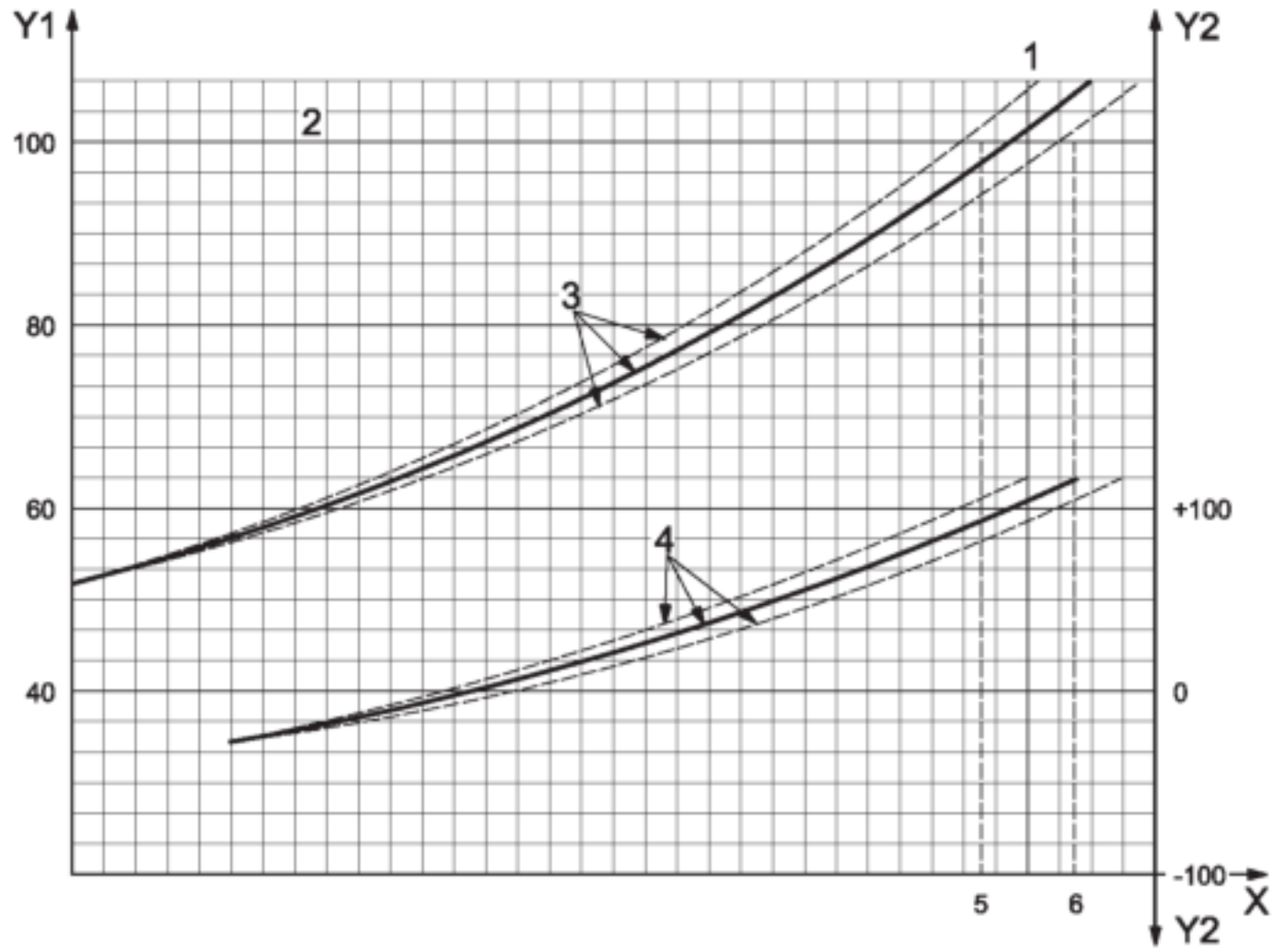
C.4.1 The HPRT shall receive a performance test at the manufacturer's test facility. Hydraulic and mechanical performance guarantees shall be based on water testing.

C.4.2 Figure C.2 shows recommended test performance tolerances for HPRTs. The pump criteria given in the main body of this standard are not applicable.

C.4.3 Vibration levels for HPRTs shall meet the criteria for pumps given in the main body of this standard.

[•] **C.4.4** If specified, the overspeed trip setting for the HPRT shall be verified at the manufacturer's test facility. If required, overspeed trip devices shall be checked and adjusted until values within 1 % of the nominal trip setting are attained. Mechanical overspeed devices shall attain three consecutive non-trending trip values that meet this criterion.

NOTE Determining the runaway speed during a water test can be considered, but this speed can be accurately calculated once performance with water is known. Runaway speed for gas-rich steams cannot be determined by water tests.



Key

X	flowrate
Y1	differential head, expressed as a percentage
Y2	rated power, expressed as a percentage
1	rated flow
2	rated head
3	typical head vs. flowrate curve
4	typical power vs. flowrate curve
5	low-side tolerance (95 %)
6	high-side tolerance (105 %)

Figure C.2—HPRT Test Performance Tolerances

Annex D
(informative)

Special-purpose Centrifugal Pumps

D.1 Definition

A special-purpose centrifugal pump is one that meets one or more of the following criteria:

a) operating in a severe, critical application;

b) a prototype—a design that is different enough from existing designs to warrant, in either the manufacturer's or the purchaser's opinion, the extent of engineering analysis and manufacturing process specified in this Annex D;

c) high energy (see definition 3.1.12 and 6.1.27).

NOTE "High energy" is a term that has been used by some in the oil and gas industry to define if a design requires more attention to engineering and manufacturing than is required by API 610. "High energy" encompasses a myriad of parameters (shaft power, power per stage, head per stage, pressure rise per stage, value of first stage impeller inlet peripheral velocity).

D.2 Selection

D.2.1 Pressure Boundary Type Selection

Determine pressure boundary configuration (axially or radially split) following rules of API 610 unless purchaser's specification requires otherwise.

D.2.2 Inner Casing (Type BB5 Multistage Pumps) Selection

For type BB5 barrel pumps, specify the inner case requirements:

a) radially split (generically "diffuser"),

b) axially split (generically "volute").

D.2.3 Rotor Selection

Determine/specify rotor requirements using the following criteria and guidelines.

D.2.3.1 For one and two stage pumps, follow rules of API 610 for rotor stiffness.

D.2.3.2 For multistage pumps (type BB3 and BB5 pumps), apply D.2.3.2.1 or D.2.3.2.2 as applicable.

D.2.3.2.1 Follow rules of API 610. Step the shaft diameter at each impeller for ease of assembly and disassembly.

D.2.3.2.2 If agreed by the purchaser and the vendor, the pump rotor can be designed for stiffness based on static deflection with the rotor centered in its end clearances which provides static radial clearance at mid-span ≥ 0.30 of the minimum new diametral clearance at rotor mid-span.

D.3 Design

D.3.1 General

The design of such pumps often requires more purchaser and vendor interface than normal pumps described in the body of API 610. The following are recommended discussion points and reviews that should be agreed between the parties and reflected in the final purchase specification.

D.3.2 Overall Design Review

A more detailed overall review of the pump and its desired operating points and characteristics is often required. A thorough understanding of the pump and other components that make up the pump-driver package is essential. The parties will define documentation and inspection requirements that go beyond API 610 normal requirements. Below is a partial list:

a) conceptual design and predicted hydraulic performance;

b) detailed mechanical layout;

c) review of all engineering reports, major detail drawings, purchase specifications, relevant inspections, and test procedures;

d) a design failure mode and effect analysis (DFMEA) as described in 6.1.5 and API 691.

D.3.3 Pressure Boundary Design Review

The parties will determine if the pressure boundary design will follow the “design by rule” method of API 610 or a “design by analysis” method.

D.3.3.1 The “design by rule” method is in the body of API 610.

D.3.3.2 The “design by analysis” method establishes the design by finite element analysis (FEA) analysis conducted at hydrostatic test pressure in 8.3.2 (ASME *BPVC* Section VIII, Division 2) based upon the following:

a) allowable local stress for 20 year fatigue life considering pressure and temperature variations,

   NOTE Pressure variations can occur during cycling of speeds on AFD drives or pumps in series, such as pipeline applications.

b) required distribution of residual stress in metal-to-metal sealing surfaces,

c) allowable deflection at critical fits,

d) allowable changes of running clearances caused by casing or component deflections.

NOTE A different hydrostatic test multiplier is sometimes agreed.

D.3.4 Impeller(s) Design Review

At high energy levels, hydraulic forces can become substantial. The parties will define and agree upon the impeller design requirements. A partial list of suggested topics follows below.

D.3.4.1 First stage: Consider the use of NPSH3 or incipient NPSH (NPSHi) and effect on impeller life.

D.3.4.2 All stages: If head/stage ≥ 1640 ft (≥ 500 m), consider the following:

a) structural analysis to determine shroud natural frequencies, wet mode shapes, and potential for resonance-separation on interference diagram ≥ 10 %;

b) structural analysis with hydraulic loading, including the effect of the operating flow range and impeller to collector vane tip clearance, to verify impeller fatigue life at rated speed accounting for the effect of the number of cycles on the endurance strength.

D.3.5 Diffuser(s) or Volute(s) Review

Like the impellers above, the stationary hydraulic components also see substantial hydraulic forces in high-energy pumps. If head/stage ≥ 1640 ft (≥ 500 m), structural analysis of the diffusers or volutes similar to but simpler than impellers should be performed.

D.3.6 Shaft Seals Review

Special-purpose pumps can have large shafts at high speed or can have other duties that are beyond normal mechanical seal operating parameters. For such pumps, the parties are to consider the following steps in seal selection and especially prototype testing:

a) abide by requirements of API 610 and API 682 where applicable,

b) consider gas seals for light hydrocarbons below 0.70 specific gravity and CO_2

c) consider API 682 qualification test of prototype seals to establish seal performance before releasing for manufacture.

D.3.7 Bearing Type and Design Review

Consider operating history of planned bearing types and lubrication system. Follow API 610 and API 614 for established bearing and lubrication system designs and select subvendors with applicable operating history. If new designs or new subvendors are selected, qualification test of prototypes to establish performance should be considered before releasing for manufacture.

D.3.8 Bearing Housing Review

Bearing housing resonance and other issues can be troublesome during performance test and cause other issues. For established design with applicable operating history, follow API 610. If new design, determine by FEA the natural frequencies of housing(s) bolted to pump casing.

D.3.9 Condition Monitoring and Other Instrumentation Review

Consider the level of instrumentation for condition monitoring, and uniformity of pump warm-up or cooldown before operation. Plan for or include provisions for measurements that might be useful in troubleshooting.

D.4 Manufacture

D.4.1 General

Due to the often-critical nature of such special-purpose pumps, greater attention to manufacturing processes can be deemed advisable. Below is a partial list of such processes and discussion points that can be considered by purchaser and vendor.

D.4.2 Pressure Boundary Manufacture

D.4.2.1 Castings should consider the following.

a) Verify castability as designed and rigged with solidification modeling.

b) Verify heat treatment of all steel castings.

c) Nondestructive testing (NDT)—Refer to 8.2.1.6, Table 14. Verify with foundry to ensure molding design reflects required quality level and needed test coupons.

d) Verify all welding procedures used for manufacturing, repair, and fabrications applicable to the materials provided.

D.4.2.2 Forgings should consider the following.

a) Verify heat treatment of all steel forgings.

b) NDT—Refer to 8.2.1.6, Table 14. Verify with foundry to ensure molding design reflects required quality level and needed test coupons.

c) Verify all welding procedures used for manufacturing, repair, and fabrications applicable to the materials provided

D.4.3 Impeller(s) Manufacture

Special-purpose pumps often utilize precision cast impellers to control hydraulic forces and flow, as well as provide improved interior surface finish.

a) Verify heat treatment of all stainless steel castings.

b) NDT—consider MT or PT of accessible areas. Consider dimensional and template checks of highly stressed critical areas, flow areas, vane spacing, and vane angles.

D.4.4 Shaft Manufacture

Both parties should agree upon barstock or forging production by appropriate method.

D.4.4.1 Refer to 8.2.1.9, Table 15. Consider NDT, heat treating methods, and test coupons required.

D.4.4.2 Machine to produce finished shaft with runout of critical fits within allowable limits—can be lower than allowed by body of API 610.

D.4.5 Diffuser(s) Manufacture

If a diffuser design pump was selected in D.2.2 above, there are two usual diffuser designs as listed below. In either case, verify the hydraulic and mechanical dimensions.

D.4.5.1 Diffuser integral—machined into cast stage piece—treat as pressure boundary casting in D.4.2.1 above.

D.4.5.2 Diffuser separate—precision cast, machined then attached to semi-finished stage piece—treat diffuser as impeller; treat balance of stage piece as pressure boundary casting in D.4.2.1 above.

D.4.6 Volute(s) Manufacture

If an axially split inner casing type of BB5 pump was selected in D.2.2 above, treat inner case castings as pressure boundary casting in D.4.2.1 above. Verify hydraulic and mechanical dimensions.

D.4.7 Shaft Seal Manufacture

See D.3.6 above.

D.4.8 Bearing Manufacture

See D.3.7 above.

D.4.9 Bearing Housing Manufacture

If bearing housings are castings, treat as pressure boundary casting in D.4.2.1 above. If new design, perform resonance test when assembled to confirm FEA in D.3.8 above.

D.4.10 Testing

Due to the often-critical nature of such special-purpose pumps, greater attention to testing and possibly more extensive testing can be deemed advisable. A few such testing topic discussion points that can be considered by purchaser and vendor are given in D.4.10.1 and D.4.10.2.

D.4.10.1 Engineering/development testing points for discussion are as follows:

a) prototype test of one or two stages if deemed necessary to check hydraulic design before releasing for manufacture;

b) verify NPSHi if required by manufacturer or purchaser's specification;

c) test complete pump with maximum diameter impellers to verify performance (head and power vs flow; rotor residual axial thrust and mechanical operation);

d) perform additional testing required by manufacturer or purchaser's specification;

e) test complete pump with impellers cut to diameter for rated performance, then perform a 4-hour run test per 8.3.5.2.7.

D.4.10.2 Contract/witness test of pump with rated impellers is often specified.

Annex E
(informative)

Inspector’s Checklist

The levels indicated in Table E.1 may be characterized as follows:

— Level 1 is typically used for pumps in general services;

— Level 2 comprises performance and material requirements and is more stringent than Level 1;

— Level 3 items are considered for pumps in critical services.

The required inspection shall be indicated in the first column as follows.

— C: Certification only.

— O: Observed inspection.

— W: Witnessed inspection.

Table E.1—Inspector's Checklist

Inspection Required C, O, or W	Item	API 610 13th Edition Subsection	Date Inspected	Inspected by	Status
Level 1—Basic					
	Casing marking (Serial No.)	6.13.3			
	Motors and electrical components area classification	6.1.36			
	Casing jackscrews	6.2.25			
	Nozzle size, rating, and finish [a]	Outline drawing, 6.4.1.1, 6.4.2			
	Baseplate requirements	7.4			
	Certified hydrostatic test	8.3.2			
	Performance within tolerance (certified)	8.3.3.4.5			
	NPSH3 within tolerance (certified)	8.3.4.3.4			
	Vibration within tolerance (certified)	8.3.3.11.1			
	Rotation arrow	6.13.4			
	Overall dimensions and connection locations [a]	Outline drawing			
	Anchor bolt layout and size [a]	Outline drawing			
	Auxiliary piping flow diagram	Annex B			
	Piping fabrication and installation	7.6			
	Equipment nameplate data	6.13.2			
	Oil and bearing temperature (certified)	6.10.2.7			
	Restrained rotor	8.4.4.1			
	Storage preservation instructions	8.4.4.9			
	Rust prevention	8.4.4.2, 8.4.4.3, 8.4.4.5, 8.4.4.7, 8.4.4.9, 8.4.8			
	Painting	8.4.4.4			
	Preparation for shipment	8.4.2, 8.4.4.6, 8.4.4.8			
	Shipping documents and tags	8.4.8, 8.4.9, 8.4.11			
Level 2—Intermediate (add to Level 1)					
	Material certification	6.12.1.9			
	Nondestructive examination (components)	6.12.1.4, 8.2.1.6			
	Hydrostatic test (witnessed)	8.3.2			
	Building records (runouts, clearances)	6.6.7, 6.6.9, 6.6.10, 6.6.13, 6.7.4, 9.2.2.4, 9.3.3.1, 9.3.4.2, 9.3.8.2.1, 9.3.12.2 d)			

Table E.1—Inspector's Checklist (Continued)

Inspection Required C, O, or W	Item	API 610 13th Edition Subsection	Date Inspected	Inspected by	Status
	Alignment	8.4.1.1			
	Hold down bolt clearance	8.4.1.3			
	Performance and NPSH tests (witnessed)	8.3.3, 8.3.4.3			
Level 3—Special (add to Levels 1 and 2)					
	Welding procedures approved	6.12.3.1			
	Welding repairs approved	6.12.3.2			
	Welding repair maps	6.12.2.7			
	Impeller/rotor balancing	6.9.3, 9.2.4.2			
	Bearing inspection after testing	9.2.7.4			
	Nozzle forces and moments test	7.4.26			
	Mechanical run test	8.3.4.2			
	Complete unit test	8.3.5.4			
	Sound level test	8.3.5.5			
	Auxiliary equipment test	8.3.5.6			
	Resonance test (bearing housing)	8.3.5.7, 9.3.9.2			

[a] Check against certified dimensional outline drawing.

Annex F
(normative)

Criteria for Piping Design

F.1 Horizontal Pumps

F.1.1 Acceptable piping configurations should not cause excessive misalignment between the pump and driver. Piping configurations that produce component nozzle loads lying within the ranges specified in Table 5 limit casing distortion to one-half the pump vendor's design criterion (see 6.2.4) and ensure pump shaft displacement of less than 0.010 in. (250 µm).

F.1.2 Piping configurations that produce loads outside the ranges specified in Table 5 are also acceptable without consultation with the pump vendor if the conditions specified in F.1.2 a) through F.1.2 c) as follows are satisfied. Satisfying these conditions ensures that any pump casing distortion is within the vendor's design criteria (see 6.2.4) and that the displacement of the pump shaft is less than 0.015 in. (380 µm).

a) The individual component forces and moments acting on each pump nozzle flange shall not exceed the range specified in Table 5 (T4) by a factor of more than 2.

b) The resultant applied force (F_{RSA}, F_{RDA}) and the resultant applied moment (M_{RSA}, M_{RDA}) acting on each pump-nozzle flange shall satisfy the appropriate interaction equations as given in Equations (F.1) and (F.2):

$$\left[F_{RSA}/(1.5 \times F_{RST4})\right] + \left[M_{RSA}/(1.5 \times M_{RST4})\right] < 2 \quad \text{(F.1)}$$

$$\left[F_{RDA}/(1.5 \times F_{RDT4})\right] + \left[M_{RDA}/(1.5 \times M_{RDT4})\right] < 2 \quad \text{(F.2)}$$

c) The applied component forces and moments acting on each pump nozzle flange shall be translated to the center of the pump. The magnitude of the resultant applied force, F_{RCA}, the resultant applied moment, M_{RCA}, and the applied moment shall be limited by Equations (F.3) to (F.5). (The sign convention shown in Figure 23, Figure 24, Figure 25, Figure 26, and Figure 27 and the right-hand rule should be used in evaluating these equations.)

$$F_{RCA} < 1.5(F_{RST4} + F_{RDT4}) \quad \text{(F.3)}$$

$$|M_{YCA}| < 2.0(M_{YST4} + M_{YDT4}) \quad \text{(F.4)}$$

$$M_{RCA} < 1.5(M_{RST4} + M_{RDT4}) \quad \text{(F.5)}$$

where

$$F_{RCA} = \left[(F_{XCA})2 + (F_{YCA})2 + (F_{ZCA})2\right]^{0.5} \quad \text{(F.3.1)}$$

where

$$F_{XCA} = F_{XSA} + F_{XDA} \quad \text{(F.3.1.1)}$$

$$F_{YCA} = F_{YSA} + F_{YDA} \quad \text{(F.3.1.2)}$$

$$F_{ZCA} = F_{ZSA} + F_{ZDA} \quad \text{(F.3.1.3)}$$

$$M_{RCA} = \left[(M_{XCA})2 + (M_{YCA})2 + (M_{ZCA})2\right]^{0.5} \quad \text{(F.5.1)}$$

where

$$M_{XCA} = M_{XSA} + M_{XDA} - \left[(F_{YSA})(zS) + (F_{YDA})(zD) - (F_{ZSA})(yS) - (FZDA)(yD)\right] / 12 \quad \text{(F.5.1.1)}$$

$$@\mathrm{m}M_{YCA} = M_{YSA} + M_{YDA} + \left[(F_{XSA})(zS) + (F_{XDA})(zD) - (F_{ZSA})(xS) - (F_{ZDA})(xD)\right] / 12 \quad \text{(F.5.1.2)}$$

$$M_{ZCA} = M_{ZSA} + M_{ZDA} - \left[(F_{XSA})(yS) + (F_{XDA})(yD) - (F_{YSA})(xS) - (F_{YDA})(xD)\right] / 12 \quad \text{(F.5.1.3)}$$

In SI units, the constant 12 shall be changed to 1000. This constant is the conversion factor to change inches to feet or millimeters to meters.

F.1.3 Piping configurations that produce loads greater than those allowed in F.1.2 shall be approved by the purchaser and the vendor. To evaluate the actual machine distortion (at ambient conditions), the piping alignment checks required in API 686:2009, Chapter 6, should be performed.

NOTE API 686 allows only a small fraction of the permitted distortion resulting from use of the values from this annex.

F.2 Vertical In-line Pumps

Vertical in-line pumps (OH3 and OH6) that are supported only by the attached piping can be subjected to component piping loads that are more than double the values shown in Table 5 if these loads do not cause a principal stress greater than 5950 psi (41 N/mm²) in either nozzle. For calculation purposes, the section properties of the pump nozzles shall be based on Schedule 40 pipe whose nominal size is equal to that of the appropriate pump nozzle. Equations (F.6), (F.7), and (F.8) can be used to evaluate the principal stress, the longitudinal stress, and the shear stress, respectively, in the nozzles.

For SI units, Equations (F.6) to (F.8) apply:

$$\sigma_p = (\sigma/2) + (\sigma^2/4 + \tau^2)^{0.5} < 41 \quad \text{(F.6)}$$

$$\sigma_l = \left[1.27 F_Y/(D_o^{\,2} - D_i^{\,2})\right] + \left[10{,}200\, D_o (M_X^{\,2} + M_Z^{\,2})^{0.5}\right] / (D_o^{\,4} - D_i^{\,4}) \quad \text{(F.7)}$$

$$\tau = \left[1.27 (F_X^{\,2} + F_Z^{\,2})^{0.5}\right] / (D_o^{\,2} - D_i^{\,2}) + \left[5100\, D_o |(M_Y)|\right] / (D_o^{\,4} - D_i^{\,4}) \quad \text{(F.8)}$$

For USC units, Equations (F.9) through (F.11) apply:

$$\sigma_p = (\sigma/2) + (\sigma^2/4 + \tau^{\,2})^{\,0.5} < 5950 \quad \text{(F.9)}$$

$$\sigma_l = \left[1.27 F_Y/(D_o^{\,2} - D_i^{\,2})\right] + \left[122\, D_o (M_X^{\,2} + M_Z^{\,2})^{0.5}\right] / (D_o^{\,4} - D_i^{\,4}) \quad \text{(F.10)}$$

$$\tau = \left[1.27 (F_X^{\,2} + F_Z^{\,2})^{0.5}\right] / (D_o^{\,2} - D_i^{\,2}) + \left[61\, D_o (|M_Y|)\right] / (D_o^{\,4} - D_i^{\,4}) \quad \text{(F.11)}$$

where

σ_p is the principal stress, expressed in pounds-force per square inch (megapascals);

σ_l is the longitudinal stress, expressed in pounds-force per square inch (megapascals);

τ is the shear stress, expressed in pounds-force per square inch (megapascals);

F_X is the applied force on the X axis;

F_Y is the applied force on the Y axis;

F_Z is the applied force on the Z axis;

M_X is the applied moment on the X axis;

M_Y is the applied moment on the Y axis;

M_Z is the applied moment on the Z axis;

D_i, D_o are the inner and outer diameters of the nozzles, expressed in inches (millimeters).

F_X, F_Y, F_Z, M_X, M_Y, and M_Z represent the applied loads acting on the suction or discharge nozzles, thus subscripts S_A and D_A have been omitted to simplify the equations. The sign of F_Y is positive if the load puts the nozzle in tension; the sign is negative if the load puts the nozzle in compression. Reference can be made to Figure 23 and the applied nozzle loads to determine whether the nozzle is in tension or compression. The absolute value of M_Y should be used in Equations (F.8) and (F.11).

F.3 Nomenclature

The following definitions apply to the sample problems in F.4.

C is the center of the pump. For pump types OH2, BB2, and BB5 with two support pedestals, the center is defined by the intersection of the pump shaft centerline and a vertical plane passing through the center of the two pedestals (see Figure 26 and Figure 27). For pump types BB1, BB3, and BB5 with four support pedestals, the center is defined by the intersection of the pump shaft centerline and a vertical plane passing midway between the four pedestals (see Figure 25);

D is the discharge nozzle;

D_i is the inside diameter of Schedule 40 pipe whose nominal size is equal to that of the pump nozzle in question, expressed in inches (millimeters);

D_o is the OD of Schedule 40 pipe whose nominal size is equal to that of the pump nozzle in question, expressed in inches (millimeters);

F is the force, expressed in pounds force (newtons);

F_R is the resultant force; F_{RSA} and F_{RDA} are calculated by the square root of the sum of the squares method using the applied component forces acting on the nozzle flange; F_{RST4} and F_{RDT4} are extracted from Table 5, using the appropriate nozzle size;

M is the moment, expressed in foot-pounds force (newton meters);

M_R is the resultant moment; M_{RSA} and M_{RDA} are calculated by the square root of the squares method using the applied component moments acting on the nozzle flange; M_{RST4} and M_{RDT4} are extracted from Table 5 using the appropriate nozzle size;

σ_l is the longitudinal stress, expressed in pounds per square inch (newtons per square millimeter);

σ_p is the principal stress, expressed in pounds force per square inch (megapascals);

τ is the shear stress, expressed in pounds per square inch (newtons per square millimeter);

S is the suction nozzle;

x, y, z are the location coordinates of the nozzle flanges with respect to the center of the pump, expressed in inches (millimeters);

X, Y, Z are the directions of the load (see Figure 23, Figure 24, Figure 25, Figure 26, and Figure 27);

Subscript A is an applied load;

Subscript T4 is a load extracted from Table 5.

F.4 Sample Problems

These sample problems are merely examples for illustration purposes only. [Each company shall develop its own approach.] They are not to be considered exclusive or exhaustive in nature. API makes no warranties, express or implied for reliance on or any omissions from the information contained in this document.

F.4.1 Example 1A—SI Units

F.4.1.1 Problem

For an overhung-end suction process pump (OH2), the nozzle sizes and location coordinates are as given in Table F.1. The applied nozzle loadings are as shown in Table F.2. The problem is to determine whether the conditions specified in F.1.2 a), F.1.2 b), and F.1.2 c) are satisfied.

F.4.1.2 Solution

F.4.1.2.1 A check of condition F.1.2 a) is as follows.

For the DN 250 end suction nozzle:

$$|F_{XSA}/F_{XST4}| = |12,900/6670|1.93 < 2.00$$

$$|F_{YSA}/F_{YST4}| = |0/5340| = 0 < 2.00$$

$$|F_{ZSA}/F_{ZST4}| = |-8852/4450| = 1.99 < 2.00$$

$$|M_{XSA}/M_{XST4}| = |-1356/5020| = 0.27 < 2.00$$

$$|M_{YSA}/M_{YST4}| = |-5017/2440| = 2.06 = 2.00$$

$$|M_{ZSA}/M_{ZST4}| = |-7458/3800| = 1.96 < 2.00$$

Since M_{YSA} exceeds the value specified in Table 5 (SI units) by more than a factor of 2, it is not satisfactory. Assume that M_{YSA} can be reduced to −4879. Then,

êM_{YSA}/M_{YST4}ú = ê−4879/2440ú = 1.999 < 2.00

For the DN 200 top discharge nozzle:

$$|F_{XDA}/F_{XDT4}| = |+7117/3780| = 1.88 < 2.00$$

$$|F_{YDA}/F_{YDT4}| = |-445/3110| = 0.14 < 2.00$$

$$|F_{ZDA}/F_{ZDT4}| = |+8674/4890| = 1.77 < 2.00$$

$$|M_{XDA}/M_{XDT4}| = |+678/3530| = 0.19 < 2.00$$

$$|M_{YDA}/M_{YDT4}| = |-3390/1760| = 1.93 < 2.00$$

$$|M_{ZDA}/M_{ZDT4}| = |-4882/2580| = 1.89 < 2.00$$

Provided that M_{YSA} can be reduced to −4879, the applied piping loads acting on each nozzle satisfy the condition specified in F.1.2 a).

Table F.1—Nozzle Sizes and Location Coordinates for Example 1A

Nozzle	Size DN	*x* mm	*y* mm	*z* mm
Suction	250	+267	0	0
Discharge	200	0	−311	+381

Table F.2—Applied Nozzle Loadings for Example 1A

Force	Value N	Moment	Value N × m
—	—	Suction	—
F_{XSA}	+12,900	M_{XSA}	−1356
F_{YSA}	0	M_{YSA}	−5017 [a]
F_{ZSA}	−8852	M_{ZSA}	−7458
—	—	Discharge	—
F_{XDA}	+7117	M_{XDA}	+678
F_{YDA}	−445	M_{YDA}	−3390
F_{ZDA}	+8674	M_{ZDA}	−4882

[a] See F.4.1.2.1.

F.4.1.2.3 A check of condition F.1.2 c) is as follows.

To check this condition, the applied component forces and moments are translated and resolved to the center of the pump. F_{RCA} is determined as follows [see F.1.2 c)]:

$$F_{RCA} = \left[(F_{XCA})^2 + (F_{YCA})^2 + (F_{ZCA})^2\right]^{0.5}$$

$$F_{XCA} = (+12,900) + (+7117) = +20,017$$

$$F_{YCA} = (0) + (-445) = -445$$

$$F_{ZCA} = (-8852) + (+8674) = -178$$

$$F_{RCA} = \left[(+20,017)^2 + (-445)^2 + (-178)^2\right]^{0.5} = 20,023@\text{m}$$

Referring to Equation (F.3):

$F_{RCA} < 1.5 \times (F_{RST4} + F_{RDT4})$

20,023 < 1.5 × (9630 + 6920)

20,023 < 24,825

M_{YCA} is determined as follows [see F.1.2 c)]:

$$M_{YCA} = M_{YSA} + M_{YDA} + \left[(F_{XSA})(zS) + (F_{XDA})(zD) - (F_{ZSA})(xS) - (F_{ZDA})(xD)/1000\right]$$
$$= (-4879) + (-3390) + [(+12,900)(0.00) + \ldots$$
$$\ldots + (+7117)(+381) + (-8852)(+267) - (+8674)(0.00)]/1000$$
$$= -3194$$

Referring to Equation (F.4):

$|M_{YCA}| < 2.0(M_{YST4} + M_{YDT4})$

$|-3194| < 2.0(2440 + 1760)$

$3194 < 8400$

M_{RCA} is determined as follows [see F.1.2 c)].

$$M_{XCA} = M_{XSA} + M_{XDA} - [(F_{YSA})(zS) + (F_{YDA})(zD) - (F_{ZSA})(yS) - (F_{ZDA})(yD)]/1000$$

$$M_{YCA} = M_{YSA} + M_{YDA} + [(F_{XSA})(zS) + (F_{XDA})(zD) - (F_{ZSA})(xS) - (F_{ZDA})(xD)]/1000$$

$$M_{ZCA} = M_{ZSA} + M_{ZDA} - [(F_{XSA})(yS) + (F_{XDA})(yD) - (F_{YSA})(xS) - (F_{YDA})(xD)]/1000$$

$$M_{RCA} = [(M_{XCA})^2 + (M_{YCA})^2 + (M^{ZCA})^2]^{0.5}$$

$$M_{XCA} = (-1356) + (+678) - [(0)(0.00) + (-445)(+381) - (-8852)(0.00) - (+8674)(-311)]/1000$$
$$= -3206$$

$M_{YCA} = -3194$ (see previous calculation)

$$M_{ZCA} = (-7458) + (-4882) - [(+12,900)(0.00) + (+7117)(-311) - (0)(+267) - (-445)(0.00)]/1000$$
$$= -10,127$$

$$M_{RCA} = [(-3206)^2 + (-3194)^2 + (-10,127)^2]^{0.5}$$
$$= 11,092$$

Referring to Equation (F.5):

$M_{RCA} < 1.5 \times (M_{RST4} + M_{RDT4})$

$11,092 < 1.5 \times (6750 + 4710)$

$11,092 < 17,190$

Thus, all the requirements of F.1.2 c) have been satisfied.

F.4.2 Example 2A—SI Units

F.4.2.1 Problem

For a DN 80 × DN 100 × 178 mm vertical in-line pump (OH3 or OH6), the proposed applied nozzle loadings are as given in Table F.3. By inspection, F_{ZSA}, M_{ZSA}, and M_{XDA} are greater than two times the values shown in Table 5 (SI units). As stated in F.2, these component loads are acceptable provided that the calculated principal stress is less than 41 MPa. The problem is to determine the principal stress for the suction nozzle and the discharge nozzle.

Table F.3—Proposed Applied Nozzle Loadings for Example 2A

Force	Value N	Moment	Value N × m
—	—	DN 100 suction	—
F_{XSA}	−2224	M_{XSA}	+136
F_{YSA}	−5338	M_{YSA}	−2034
F_{ZSA}	+1334	M_{ZSA}	+1356
—	—	DN 80 discharge	—
F_{XDA}	+1334	M_{XDA}	+2712
F_{YDA}	−2224	M_{YDA}	+271
F_{ZDA}	+445	M_{ZDA}	+136

F.4.2.2 Solution

F.4.2.2.1 Suction nozzle calculations are as follows.

For Schedule 40 pipe with a nominal size of DN 100, D_o = 114 mm and D_i = 102 mm. Therefore,

$$D_o^{\,2} - D_i^{\,2} = (114)^2 - (102)^2 = 2592$$

$$D_o^{\,4} - D_i^{\,4} = (114)^4 - (102)^4 = 6.065 \times 10^7$$

$$\left[(F_{XSA})^2 + (F_{ZSA})^2\right]^{0.5} = \left[(-2224)^2 + (+1334)^2\right]^{0.5} = 2593$$

$$\left[(M_{XSA})^2 + (M_{ZSA})^2\right]^{0.5} = \left[(+136)^2 + (+1356)^2\right]^{0.5} = 1363$$

Equation (F.7) is used to determine the longitudinal stress for the suction nozzle, σ_s.

The applied F_{YSA} load acting on the suction nozzle is in the negative Y direction and produces a compressive stress; therefore, the negative sign on F_{YSA} is used.

$$\sigma_s = \left[1.27 F_{YSA}/(D_o^{\,2} - D_i^{\,2})\right] + \left[10{,}200\, D_o (M_{XSA}^{\,2} + M_{ZSA}^{\,2})^{0.5}/(D_o^{\,4} - D_i^{\,4})\right]$$

$$= \left[1.27(-5338)/2592\right] + \left[10{,}200 \times 114 \times 1363/(6.065 \times 10^7)\right] = 23.52$$

Equation (F.8) is used to determine the shear stress for the suction nozzle, τ_s.

$$\tau_s = \left[1.27 (F_{XSA})^2 + (F_{ZSA})^2\right]^{0.5}/(D_o^{\,2} - D_i^{\,2}) + \left[0.51 \times 104\, D_o (\,|M_{YSA}|\,)\right]/(D_o^{\,4} - D_i^{\,4})$$

$$= (1.27 \times 2593/2592) + [5100 \times 114 \times (\,|-2034|\,)]/(6.065 \times 10^7) = 20.77$$

The principal stress for the suction nozzle, $\sigma_{p,s}$, is calculated using Equation (F.6):

$$\sigma_{p,s} = (\sigma_s/2) + (\sigma_s^{\,2} + \tau_s^{\,2})^{0.5} < 41$$

$$= (+23.52/2) + [(+23.52)^2 + (+20.77)^2]^{0.5} < 41$$

$$= +35.63 < 41$$

Thus, the suction nozzle loads are satisfactory.

F.4.2.2.2 Discharge nozzle calculations are as follows:

For Schedule 40 pipe with a nominal size of 80 mm, D_o = 89 mm and D_i = 78 mm. Therefore,

$$D_o^{\,2} - D_i^{\,2} = (89)^2 - (78)^2 = 1837$$

$$D_o^{\,4} - D_i^{\,4} = (89)^4 - (78)^4 = 2.573 \times 10^7$$

$$\left[(F_{XDA})^2 + (F_{ZDA})^2\right]^{0.5} = [(+1334)^2 + (+445)^2]^{0.5} = 1406$$

$$\left[(M_{XDA})^2 + (M_{ZDA})^2\right] = [(+2712)^2 + (+136)^2]^{0.5} = 2715$$

Equation (F.7) is used to determine the longitudinal stress for the discharge nozzle, σ_D.

The applied F_{YDA} load acting on the discharge nozzle is in the negative Y direction and produces a tensile stress; therefore, a positive sign on F_{XDA} is used.

$$\sigma_D = \left[1.27\,F_{YDA}/\left(D_o^{\,2} - D_i^{\,2}\right)\right] + [10,200\,D_o(M_{XDA}^{\,2} + M_{ZDA}^{\,2})^{0.5}]/\left(D_o^{\,4} - D_i^{\,4}\right)$$

$$= [1.27(+2224)/1837] + [10,200(89)(2715)]/2.573 \times 10^7 = 97.33$$

Equation (F.8) is used to determine the shear stress for the discharge nozzle, τ_D.

$$\tau_D = [1.27\,(F_{XDA})^2 + (F_{ZDA})^2]^{0.5}/\left(D_o^{\,2} - D_i^{\,2}\right) + \left[5100\,D_o\left(|M_{YDA}|\right)\right]/\left(D_o^{\,4} - D_i^{\,4}\right)$$

$$= [1.27 \times 1406/1837] + [5100 \times 89 \times (|+271|)]/(2.573 \times 107) = 5.75$$

The principal stress for the discharge nozzle, $\sigma_{p,D}$, is calculated using Equation (F.6):

$$\sigma_{p,D} = \left(\sigma_D/^2\right) + (\sigma_D^{\,2}/4 + \tau_D^{\,2})^{0.5} < 41$$

$$= (+97.33/^2) + [(+97.33)^2/4 + (+5.75)^2]^{0.5}$$

$$= +97.67 > 41$$

Thus, the discharge nozzle loads are too large. By inspection, if M_{XDA} is reduced by 50 % to 1356 N×m, the resulting principal stress still exceeds 41 MPa. Therefore, the maximum value for M_{XDA} is twice M_{XDT4}, or 1900 N×m.

F.4.3 Example 1B—USC Units

F.4.3.1 Problem

For an overhung end-suction process pump (OH2), the nozzle sizes and location coordinates are as given in Table F.4. The applied nozzle loadings are as given in Table F.5. The problem is to determine whether the conditions specified in F.1.2 a), F.1.2 b), and F.1.2 c) are satisfied.

Table F.4—Nozzle Sizes and Location Coordinates for Example 1B

Nozzle	Size	*x*	*y*	*z*
Suction	10	+10.50	0	0
Discharge	8	0	−12.25	+15
NOTE Dimensions in inches.				

Table F.5—Applied Nozzle Loadings for Example 1B

Force	Value lbf	Moment	Value ft × lbf
—	—	Suction	—
F_{XSA}	+2900	M_{XSA}	−1000
F_{YSA}	0	M_{YSA}	−3700 [a]
F_{ZSA}	−1990	M_{ZSA}	−5500
—	—	Discharge	—
F_{XDA}	+1600	M_{XDA}	+500
F_{YDA}	−100	M_{YDA}	−2500
F_{ZDA}	+1950	M_{ZDA}	−3600
[a] See F.4.1.2.1.			

F.4.3.2.1 A check of condition of F.1.2 a) is as follows:

For the 10 in. end suction nozzle:

$$|F_{XSA}/F_{XST4}| = |+2900/1500| = 1.93 < 2.00$$

$$|F_{YSA}/F_{YST4}| = |\ 0/1200\ | = 0 < 2.00$$

$$|F_{ZSA}/F_{ZST4}| = |-1990/1000| = 1.99 < 2.00$$

$$|M_{XSA}/M_{XST4}| = |-1000/3700\ | = 0.27 < 2.00$$

$$|M_{YSA}/M_{YST4}\ | = |-3700/1800\ | = 2.06 > 2.00$$

$$|M_{ZSA}/M_{ZST4}| = |-5500/2800| = 1.96 < 2.00$$

Since M_{YSA} exceeds the value specified in Table 5 (USC units) by more than a factor of 2, it is not satisfactory. Assume that M_{YSA} can be reduced to −3599. Then,

$$|M_{YSA}/M_{YST4}| = |-3599/1800| = 1.999 < 2.00$$

For the 8 in. top discharge nozzle:

$$|F_{XDA}/F_{XDT}| = |+1600/850| = 1.88 < 2.00$$

$$L_{10h,sys} = [(1/L_{10hA})^{3/2} + (1/L10hB)^{3/2} + ... + (1/L10hN)3/2]-2/3$$

$$|F_{YDA}/F_{YDT}| = |-100/700| = 0.14 < 2.00$$

$$|F_{ZDA}/F_{ZDT4}| = |+1950/1100| = 1.77 < 2.00$$

$$|M_{XDA}/M_{XDT4}| = |+500/2600| = 0.19 < 2.00$$

$$|M_{YDA}/M_{YDT4}| = |-2500/1300| = 1.93 < 2.00$$

$$|M_{ZDA}/M_{ZDT4}| = |-3600/1900| = 1.89 < 2.00$$

Provided that M_{YSA} can be reduced to −3599, the applied piping loads acting on each nozzle satisfy the condition specified in F.1.2 a).

F.4.3.2.2 A check of condition F.1.2 b) is as follows:

For the suction nozzle, F_{RSA} and M_{RSA} are determined using the square root of the sum of the squares method:

$$F_{RSA} = [(F_{XSA})^2 + (F_{YSA})^2 + (F_{ZSA})^2]^{0.5} = [(+2900)^2 + (0)^2 + (-1990)^2]^{0.5} = 3517$$

$$M_{RSA} = [(M_{XSA})^2 + (M_{YSA})^2 + (M_{ZSA})^2]^{0.5} = [(-1000)^2 + (-3599)^2 + (-5500)^2]^{0.5} = 6649$$

Referring to Equation (F.1),

$$F_{RSA}/(1.5 \times F_{RST4}) + M_{RSA}/(1.5 \times M_{RST4}) < 2$$

$$3517/(1.5 \times 2200) + 6649/(1.5 \times 5000) < 2$$
$$1.95 < 2$$

For the discharge nozzle, F_{RDA} and M_{RDA} are determined by the same method used to find F_{RSA} and M_{RSA}:

$$F_{RDA} = [(F_{XDA})^2 + (F_{YDA})^2 + (F_{ZDA})^2]^{0.5} = [(+1600)^2 + (-100)^2 + (+1950)^2]^{0.5} = 2524$$

$$M_{RDA} = [M_{XDA})^2 + (M_{YDA})^2 + (M_{ZDA})^2]^{0.5} = [(+500)^2 + (-2500)^2 + (-3600)^2]^{0.5} = 4411$$

Referring to Equation (F.2),

$F_{RDA}/(1.5 \times F_{RDT4}) + M_{RDA}/(1.5 \times M_{RDT4}) < 2$

$2524/(1.5 \times 1560) + 4411/(1.5 \times 3500) < 2$

$1.92 < 2$

The loads acting on each nozzle satisfy the appropriate interaction equation, so the condition specified in F.1.2 b) is satisfied.

F.4.3.2.3 A check of condition F.1.2 c) is as follows.

To check this condition, the applied component forces and moments are translated and resolved to the center of the pump. F_{RCA} is determined as follows [see F.1.2 c)]:

$$F_{XCA} = F_{XSA} + F_{XDA}$$

$$F_{YCA} = F_{YSA} + F_{YDA}$$

$$F_{ZCA} = F_{ZSA} + F_{ZDA}$$

$$F_{RCA} = [(F_{XCA})^2 + (F_{YCA})^2 + (F_{ZCA})^2]^{0.5}$$

$$F_{XCA} = (+2900) + (+1600) = +4500$$

$$F_{YCA} = (0) + (-100) = -100$$

$$F_{ZCA} = (-1990) + (+1950) = -40$$

$$F_{RCA} = [(+4500)^2 + (-100)^2 + (-40)^2]^{0.5} = 4501$$

Referring to Equation (F.3),

$$F_{RCA} < 1.5 \times (F_{RST4} + F_{RDT4})$$

$$4501 < 1.5 \times (2200 + 1560)$$

$$4501 < 5640$$

M_{YCA} is determined as follows [see F.1.2 c)]:

$$M_{YCA} = M_{YSA} + M_{YDA} + [(F_{XSA})(zS) + (F_{XDA})(zD) - (F_{ZSA})(xS) - (F_{ZDA})(xD)]/12$$

$$= (-3599) + (-2500) + [(+2900)(0.00) + (+1600)(+15) - (-1990)(+10.5) - (+1950)(0.00)]/12$$
$$= -2358$$

Referring to Equation (F.4),

$$|M_{YCA}| < 2.0 \times (M_{YST4} + M_{YDT4})$$

$$|-2358| < 2.0 \times (1800 + 1300)$$

$$2358 < 6200$$

M_{RCA} is determined as follows [see F.1.2 c)]:

$$M_{XCA} = M_{XSA} + M_{XDA} - [(F_{YSA})(zS) + (F_{YDA})(zD) - (F_{ZSA})(yS) - (F_{ZDA})(yD)]/12$$

$$M_{YCA} = M_{YSA} + M_{YDA} + [(F_{XSA})(zS) + (F_{XDA})(zD) - (F_{ZSA})(xS) - (F_{ZDA})(xD)]/12$$

$$M_{ZCA} = M_{ZSA} + M_{ZDA} - [(F_{XSA})(yS) + (F_{XDA})(yD) - (F_{YSA})(xS) - (F_{YDA})(xD)]/12$$

$$M_{RCA} = \left[(M_{XCA})^2 + (M_{YCA})^2 + (M_{ZCA})^2\right]^{0.5}$$

$$M_{XCA} = (-1000) + (+500) - [(0)(0.00) + (-100)(+15.00) - (-1990)(0.00) - (+1950)(-12.25)]/12$$
$$= -2366$$

$M_{YCA} = -2358$ (see previous calculation)

$$M_{ZCA} = (-5500) + (-3600) - [(+2900)(0.00) + (+1600)(-12.25) - (0)(+10.50) - (-100)(0.00)]/12$$
$$= -7467$$

$$M_{RCA} = [(-2366)^2 + (-2358)^2 + (-7467)^2]^{0.5} = 8180$$

Referring to Equation (F.5),

$$M_{RCA} < 1.5 \times (M_{RST4} + M_{RDT4})$$

$$8180 < 1.5 \times (5000 + 3500)$$

$$8180 < 12,750$$

Thus, all the requirements of F.1.2 c) have been satisfied.

F.4.4 Example 2B—USC Units

F.4.4.1 Problem

For a NPS 3 × NPS 4 × 7 in. vertical in-line pump (OH3 or OH6), the proposed applied nozzle loadings are as given in Table F.6. By inspection, F_{ZSA}, M_{ZSA}, and M_{XDA} are greater than two times the values shown in Table 5 (USC units). As stated in F.2, these component loads are acceptable provided that the calculated principal stress is less than 5950 psi. The problem is to determine the principal stress for the suction nozzle and the discharge nozzle.

Table F.6—Proposed Applied Nozzle Loadings for Example 2B

Force	Value lbf	Moment	Value ft × lbf
—	—	NPS 4 suction	—
F_{XSA}	−500	M_{XSA}	+100
F_{YSA}	−1200	M_{YSA}	−1500
F_{ZSA}	+300	M_{ZSA}	+1000
—	—	NPS 3 discharge	—
F_{XDA}	+300	M_{XDA}	+2000
F_{YDA}	−500	M_{YDA}	+200
F_{ZDA}	+100	M_{ZDA}	+100

F.4.4.2.1 Suction nozzle calculations are as follows.

For Schedule 40 pipe with a nominal size of 4 in., $D_o = 4.500$ in. and $D_i = 4.026$ in. Therefore,

$$D_o^2 - D_i^2 = (4.500)^2 - (4.026)^2 = 4.04$$

$$D_o^4 - D_i^4 = (4.500)^4 - (4.026)^4 = 147.34$$

$$[(F_{XSA})^2 + (F_{ZSA})^2]^{0.5} = [(-500)^2 + (+300)^2]^{0.5} = 583$$

$$[(M_{XSA})^2 + (M_{ZSA})^2]^{0.5} = [(+100)^2 + (+1000)^2]^{0.5} = 1005$$

Equation (F.10) is used to determine the longitudinal stress for the suction nozzle, $\sigma_{l,s.}$

The applied F_{YSA} load acting on the suction nozzle is in the negative Y direction and produces a compressive stress; therefore, the negative sign on F_{YSA} is used.

$$\sigma_{l,s} = \left[1.27 F_{YSA}/(D_o{}^2 - D_i{}^2)\right] + \left[122 D_o (M_{XSA}{}^2 + M_{ZSA}{}^2)^{0.5}\right] / \left((D_o{}^4 - D_i{}^4)\right)$$

$$= [1.27 \times (-1200)/4.04] + [122 \times 4.500 \times 1005)]/147.34$$

$$= 3367$$

Equation (F.11) is used to determine the shear stress for the suction nozzle, τ_s.

$$\tau_s = [1.27 (F_{XSA}{}^2 + F_{ZSA}{}^2)^{0.5}]/(D_o{}^2 - D_i{}^2) + \left[61 D_o (|M_{YSA}|)\right]/(D_o{}^4 - D_i{}^4)$$

$$= (1.27 \times 583/4.04) + [61 \times 4500(|-1500|)]/147.34$$
$$= 2978$$

The principal stress for the suction nozzle, $\sigma_{p,s}$, is calculated using Equation (F.9):

$$\sigma_{p,s} = (\sigma_s/^2) + (\sigma_s{}^2/4 + \tau_s{}^2)^{0.5} < 5950$$

$$= (+3367/2) + \text{Skipping malformed tag: msup}$$
$$= +5105 < 950$$

Thus, the suction nozzle loads are satisfactory.

F.4.4.2.2 Discharge nozzle calculations are as follows.

For Schedule 40 pipe with a nominal size of 3 in., D_o = 3.500, and D_i = 3.068. Therefore,

$$D_o{}^2 - D_i{}^2 = (3.500)^2 - (3.068)^2 = 2.84$$

$$D_o{}^4 - D_i{}^4 = (3.500)^4 - (3.068)^4 = 61.47$$

$$\left[(F_{XDA})^2 + (F_{ZDA})^2\right]^{0.5} = \left[(+300)^2 + (+100)\right]^{2\,0.5} = 316$$

$$\left[(M_{XDA})^2 + (M_{ZDA})^2\right]^{0.5} = \left[(+2000)^2 + (+100)2\right]^{0.5} = 2002$$

Equation (F.10) is used to determine the longitudinal stress for the discharge nozzle, $\sigma_{l,D}$.

The applied F_{YDA} load acting on the discharge nozzle is in the negative Y direction and produces a tensile stress; therefore, a positive sign on F_{YDA} is used.

Equation (F.11) is used to determine the shear stress for the discharge nozzle, τ_D.

$$\tau_D = \left[1.27 (F_{XDA}{}^2 + F_{ZDA}{}^2)^{0.5}\right]/(D_o{}^2 - D_i{}^2) + \left[61 D_o (|M_{YDA}|)/(D_o{}^4 - D_i{}^4)\right]$$
$$= (1.27 \times 316/2.84) + [61 \times 3.500 \times (|+200|)/61.47] = 836$$

The principal stress for the discharge nozzle, $\sigma_{p,D}$, is calculated using Equation (F.9):

$$\sigma_{p,D} = \left(\sigma_D/2\right) + \left(\sigma_D{}^{2} + \tau_D{}^{2}\right)^{0.5} < 5950$$
$$= (+14,131/2) + [(+14,131)^{2} + (+836)^{2}]^{0.5} = +14,181 > 5950$$

Thus, the discharge nozzle loads are too large. By inspection, if M_{XDA} is reduced by 50 % to 1000 ft·lbf, the resulting principal stress still exceeds 5950 psi. Therefore, the maximum value for M_{XDA} is twice M_{XDT4}, or 1400 ft·lbf.

Annex G
(informative)

Materials Class Selection Guidance

Table G.1 is intended to provide general guidance for on-plot process plants and off-plot transfer and loading services. It should not be used without a knowledgeable review of the specific services involved.

Table G.1—Materials class Selection Guidance

Service	Temperature Range		Pressure Range	Materials Class	Ref. Note
	°F	°C			
Fresh water, condensate, cooling tower water (pH > 6)	< 200	< 93	All	S-5	—
Process water (pH > 6)	< 200	< 93	All	S-5 or S-6	a
Boiling water and boiler feedwater (pH > 6)	> 200	> 93	All	S-6 or C-6	a
Boiler circulator (pH > 6)	> 200	> 93	All	C-6	—
Foul water, reflux drum water, water draw, and hydrocarbons containing these waters, including reflux streams	< 350	< 175	All	S-8 or S-6	b
	> 350	> 175	All	C-6	—
Propane, butane, liquefied petroleum gas, ammonia, ethylene, low-temperature services (minimum metal temperature)	< 450	230	All	S-4	—
	>−50	> −46	All	S-4 LCB	h
	>−100	> −73	All	S-4 LC2	h
	>−150	> −100	All	S-4 LC3	h, i
	>−320	> −196	All	A-7 or A-8	h, i
Diesel oil; gasoline; naphtha; kerosene; gas oils; light, medium and heavy lubricating oils; fuel oil; residuum; crude oil; asphalt; synthetic crude bottoms	< 450	< 230	All	S-4	—
	450 to 700	230 to 370	All	S-6	b c
	> 700	> 370	All	C-6	b
Noncorrosive hydrocarbons, e.g. catalytic reformate, isomaxate, desulfurized oils	450 to 700	230 to 370	All	S-4	c
Xylene, toluene, acetone, benzene, furfural, methylethylketone (MEK), cumene	< 450	< 230	All	S-4	—
Sodium carbonate	< 350	< 175	All	C-6	—
Caustic (sodium hydroxide), concentration < 20 %	< 212	< 100	All	S-4	d
	> 212	> 100	All	—	e
Seawater	< 200	< 95	All	A-8, D-1, or D-2	f
Sour water	< 500	< 260	All	D-1	—
Produced water, formation water, and brine	All	All	All	A-8, D-1, or D-2	f
Sulfur (liquid state)	All	All	All	S-4	—
Fluid catalytic cracker (FCC) slurry	< 700	< 370	All	C-6	j
	°F	°C			
Potassium carbonate	< 350	< 175	All	C-6	—
	< 700	< 370	All	A-8	—
MEA (monoethylamine), diethylamine (DEA), triethylamine (TEA) stock solutions	< 250	< 120	All	S-4	—
DEA, TEA-lean solutions	< 250	< 120	All	S-4 or S-8	d, g
MEA-lean solution (CO_2 only)	175 to 300	80 to 150	All	S-9	d

Table G.1—Materials class Selection Guidance (Continued)

Service	Temperature Range		Pressure Range	Materials Class	Ref. Note
	°F	°C			
MEA-lean solution (CO_2 and H_2S)	175 to 300	80 to 150	All	S-8	d, g
MEA-, DEA-, TEA-rich solutions	175	< 80	All	S-4 or S-8	d
Sulfuric acid concentration > 85 %	< 100	< 38	All	S-4	b
85 % to < 1 %	< 450	< 230	All	A-8	b
Hydrofluoric acid concentration > 96 %	< 100	< 38	All	S-9 [k]	b

NOTE 1 The materials for pump parts for each material class are given in Annex H.

NOTE 2 Specific materials recommendations shall be obtained for services not clearly identified by the service descriptions listed in this table.

[a] High purity water requires steels with some chrome alloying to avoid flow accelerated corrosion (FAC) in high velocity areas. For water > 93C (200F) use C-6 materials by default unless the water chemistry (pH, conductivity, and dissolved oxygen content) can be actively controlled to prevent FAC. As an alternative, carbon steels with 1 % to 3 % chrome, specifically intended for boiler service may be considered.

[b] The corrosiveness of foul waters, hydrocarbons over 450 °F (230 °C), acids, and acid sludge may vary widely. Material recommendations from the end user shall be obtained for each service. The material class indicated above is satisfactory for many of these services but shall be verified. S-8 materials may also be considered for operating temperatures below 150 °F (66 °C). Depending on concentration, temperature and fluid velocities, higher alloy materials such as Alloy 20 for H2SO4 and Hastelloy C for HF services may be required.

[c] If product corrosivity is low, Class S-4 materials may be used for services at 451 °F to 700 °F (231 °C to 370 °C). Specific material recommendations shall be obtained in each instance.

[d] All welds shall be stress-relieved.

[e] UNS N08007 or Ni-Cu alloy pump material shall be used.

[f] For seawater, produced water, formation water, and brine services, the purchaser and the vendor shall agree on the construction materials that best suit the intended use.

[g] The vendor shall consider the effects of differential material expansion between casing and rotor and confirm suitability if operating temperatures can exceed 200 °F (95 °C).

[h] Materials selected for low-temperature services shall meet the requirements of 6.12.4. Casting alloy Grades LCB, LC2, and LC3 are shown only for reference. Grades LCB, LC2, and LC3 refer to ISO 4991. C23-45BL, C43E2aL, and C43L are equivalent to ASTM A352/A352M, Grades LCB, LC2, and LC3. Use equivalent materials for wrought alloys.

[i] Material alloys based on aluminum, bronze, aluminum bronze, and nickel, may also be considered for temperatures as low as −320 °F (−196 °C).

[j] Due to the abrasive nature of FCC slurry services, additional hard coating of wearing surfaces may be considered to increase the life of the part. Lined pumps may also be considered to increase life of the part.

[k] For HF applications, refer to the process licensor for guidance, due to variations in service.

Annex H
(normative)

Materials and Material Specifications for Pump Parts

Table H.1 lists material classes for the purchaser to select (see 6.12.1.3).

Table H.2, Table H.3, and Table H.4 may be used for guidance regarding materials specifications. If these tables are used, it should not be assumed that the material specifications are acceptable without taking full account of the service in which they will be applied. Table H.2 lists corresponding materials that may be considered acceptable. These materials represent family/type and grade only. The final required condition or hardness level (where appropriate) is not specified. These materials might not be interchangeable for all applications.

Alternative materials may be recommended for purchaser approval (see 6.12.1.1.3).

Table H.1—Material Classes for Pump Parts

Part	Material Classes and Abbreviations										
	Material Class	S-4 [k]	S-5 [k]	S-6 [f k]	S-8 [k]	S-9 [k]	C-6	A-7	A-8	D-1 [l]	D-2 [l]
	Full Compliance Material [a b]	STL	STL	STL	STL	STL	12 % Cr	AUS	316 AUS	Duplex	Super Duplex
	Trim Material [a b]	STL	STL 12 % Cr	12 % Cr	316 AUS	Ni-Cu Alloy	12 % Cr	AUS [c]	316 AUS	Duplex	Super Duplex
Pressure Casing	Yes	Carbon Steel	Carbon Steel [l]	Carbon Steel	Carbon Steel	Carbon Steel	12 % Cr	AUS	316 AUS	Duplex	Super Duplex
Inner Case Parts (Bowls, Diffusers, Diaphragms)	No	Carbon Steel	Carbon Steel	12 % Cr	316 AUS	Ni-Cu Alloy	12 % Cr	AUS	316 AUS	Duplex	Super Duplex
Impeller	Yes	Carbon Steel	Carbon Steel	12 % Cr	316 AUS	Ni-Cu Alloy	12 % Cr	AUS	316 AUS	Duplex	Super Duplex
Case Wear Rings [j]	No	Cast Iron	12 % Cr Hardened	12 % Cr Hardened	Hard-Faced 316 AUS [e]	Ni-Cu Alloy	12 % Cr Hardened	Hard-Faced AUS [e]	Hard-Faced 316 AUS [e]	Hard-Faced Duplex [e]	Hard-Faced Super Duplex [e]
Impeller Wear Rings [j]	No	Cast Iron	12 % Cr Hardened	12 % Cr Hardened	Hard-Faced 316 AUS [e]	Ni-Cu Alloy	12 % Cr Hardened	Hard-Faced AUS [e]	Hard-Faced 316 AUS [e]	Hard-Faced Duplex [e]	Hard-Faced Super Duplex [e]
Shaft	Yes	Carbon Steel	12 % Cr	12 % Cr	316 AUS	Ni-Cu Alloy	12 % Cr	AUS	316 AUS	Duplex	Super Duplex
Throat Bushings [j]	No	Cast Iron	12 % Cr Hardened	12 % Cr Hardened	316 AUS	Ni-Cu Alloy	12 % Cr Hardened	AUS	316 AUS	Duplex	Super Duplex
Interstage Sleeves [j]	No	Cast Iron	12 % Cr Hardened	12 % Cr Hardened	Hard-Faced 316 AUS [e]	Ni-Cu Alloy	12 % Cr Hardened	Hard-Faced AUS [e]	Hard-Faced 316 AUS [e]	Hard-Faced Duplex [e]	Hard-Faced Super Duplex [e]
Interstage Bushings [j]	No	Cast Iron	12 % Cr Hardened	12 % Cr Hardened	Hard-Faced 316 AUS [e]	Ni-Cu Alloy	12 % Cr Hardened	Hard-Faced AUS [e]	Hard-Faced 316 AUS [e]	Hard-Faced Duplex [e]	Hard-Faced Super Duplex [e]
Case and Gland Studs	Yes	4140 Alloy Steel	40 Alloy Steel	4140 Alloy Steel	4140 Alloy Steel	4140 Alloy Steel [d]	40 Alloy Steel	40 Alloy Steel	40 Alloy Steel	Duplex [h]	Super Duplex [h]

Table H.1—Material Classes for Pump Parts (Continued)

Part	Material Classes and Abbreviations										
	Material Class	S-4 [k]	S-5 [k]	S-6 [f k]	S-8 [k]	S-9 [k]	C-6	A-7	A-8	D-1 [l]	D-2 [l]
	Full Compliance Material [a b]	STL	STL	STL	STL	STL	12 % Cr	AUS	316 AUS	Duplex	Super Duplex
	Trim Material [a b]	STL	STL 12 % Cr	12 % Cr	316 AUS	Ni-Cu Alloy	12 % Cr	AUS [c]	316 AUS	Duplex	Super Duplex
Case Gasket	No	Aus, Spiral-Wound [g]	Aus, Spiral-Wound [g]	Aus, Spiral-Wound [g]	316 Aus, Spiral-Wound [g]	Ni-Cu Alloy, Spiral-Wound Ptfe Filled [g]	Aus, Spiral-Wound [g]	Aus, Spiral-Wound [g]	316 Aus Spiral-Wound [g]	Duplex Ss Spiral-Wound [g]	Duplex SS Spiral-Wound [g]
Discharge Head/ Suction Can	Yes	Carbon Steel	Carbon Steel	Carbon Steel	Carbon Steel	Carbon Steel	AUS	AUS	316 AUS	Duplex	Super Duplex
Column/Bowl Shaft Bushings	No	Filled Carbon	Filled Carbon	Filled Carbon	Filled Carbon	Filled Carbon	Filled Carbon	Filled Carbon	Filled Carbon	Filled Carbon	Filled Carbon
Wetted Fasteners (Bolts)	Yes	Carbon Steel	316 AUS [l]	316 AUS [l]	316 AUS	4140 Alloy Steel [d]	316 AUS [l]	316 AUS	316 AUS	Duplex	Super Duplex

Table H.1—Material Classes for Pump Parts (Continued)

[a] See 6.12.1.6.

[b] The abbreviations in the upper part of the second row indicate the case material; the abbreviations in the lower part of the second row indicate trim material. Abbreviations are as follows: 12 % CR, 12 % chromium; 316 AUS, austenitic stainless steel containing at least 2.0 % molybdenum; 4140 alloy steel, high-strength steels with approximately 0.40 % carbon, 1 % chromium, and 0.2 % molybdenum; AUS, austenitic stainless steel; CI, cast iron; STL, steel.

[c] Austenitic stainless steels include ISO types 683-13-10/19 (AISI standard types 302, 303, 304, 316, 321, and 347).

[d] A193 Gr. B7/B7M and A194 Gr. 2H/2HM Nut is the specified alloy for Hydrofluoric Acid Service.

[e] The requirement for hard-facing and the specific hard-facing material for each application is determined by the vendor and described in the proposal. Alternatives to hard-facing may include opening running clearances (6.7.7) or the use of non-galling materials or nonmetallic materials, depending on the corrosiveness of the pumped liquid.

[f] For Types VS6 and VS7 pumps in pipeline services with maximum operating temperatures of 130 °F (55 °C) or less, carbon steel bowls in S-6 material column may be considered by the purchaser. The purchaser to note on data sheet (Annex N).

[g] If pumps with axially split casings are furnished, a sheet gasket suitable for the service is acceptable. Spiral-wound gaskets shall contain a filler material suitable for the service. Gaskets other than spiral-wound may be proposed and furnished if proven suitable for service and approved by the purchaser. See 6.2.16.

[h] AISI 4140 alloy steel may be used for nonwetted case and gland studs.

[i] Some applications may require alloy grades higher than the duplex materials given in Table H.2. "Super duplex" material grades with pitting resistance equivalency (PRE) values greater than 40 can be necessary.

PRE → ≥ 40, where the PRE is based on actual chemical analysis.

PRE → $w_{Cr} + 3.3w_{Mo} + 16w_{N}$ where *w* is the percentage mass fraction of the element indicated by the subscript.

Note that alternative materials such as "super austenitic" may also be considered.

[j] Nonmetallic wear part materials, proven compatible with the specified process liquid, may be proposed within the applicable limits shown in Table H.3. [Also see 6.7.7 c)].

[k] The vendor shall consider the effects of differential material expansion between casing and rotor elements and confirm suitability if operating temperatures can exceed 200 °F (95 °C).

[l] For applications where large differences of thermal expansion can result if austenitic stainless steel fasteners are used, alternative fastener materials, such as 12 % or 17 % Cr martensitic steel, with appropriate corrosion resistance, may be used.

Table H.2—Material Specifications for Pump Parts

Material Class	Applications	ISO	USA		Europe			Japan
			ASTM	UNS [a]	EN [b]	Grade	Material No.	JIS
Cast Iron	General castings	185 Gr 300	A48/A48M class 25/30/40	F11701/ F12101	EN 1561	EN-GJL-250 EN-GJL-300	JL 1040JL 1050	G 5501 FC 250/300
Carbon Steel	Wrought/ forgings	683–18-C25	A266 Class 4	K03506	EN 10222-2	P 280 GH	1.0426	G 3202, Cl SFVC 2A
	Pressure castings	4991 C23-45 AH	A216/A216M Gr WCB	J03002	EN10213	GP 240 GH	1.0619	G 5151, Cl SCPH 2
	Bar stock: pressure	683-18-C25	A696 Gr B	G10200	EN 10273	P 295 GH	1.0481	G 4051, Cl S25C
	Bar stock: general	683-18-C45e	A576 Gr 1045	G10450	EN 10083-2	C 45	1.0503	G 4051, Cl S45C
	Bolts and studs	2604–2-F31	A193/A193M Gr B7	G41400	EN 10269	42 Cr Mo 4	1.7225	G 4107, Class 2, SNB7
	Nuts	683-1-C45	A194/A194Gr 2H	K04002	EN 10269	42 Cr Mo 4	1.1181	G 4051, Cl S45C
	Plate	9328-4, P 355 TN/ PL 355 TN	A516/A516Gr 65/70	K02403/ K02700	EN 10028-3	P 355 N P 355 NL1	1.0562 1.0566	G 3106, Gr SM400B
	Pipe	9329-2 PH26	A106/A106M Gr B or C	K03006	EN 10208-1	L 245 GA	1.0459	G 3456, Gr STPT 370/410
	Fittings	—	A105/A105M	K03504	—	—	—	G 4051 S25C G 3202 SFVC 2A
		—	A234/234MGr WPB/WPC	K03006/ K03501	—	—	—	G 3202 SFVC2B
4140 Alloy Steel	Bar stock	—	A434 Class BB A434 Class BC	G41400 [c]	EN 10083–1	42 Cr Mo 4	1.7225	G 4105, Cl SCM 440
	Bolts and studs	2604-2-F31	A193/A193M Gr B7	G41400	EN 10269	42 Cr Mo 4	1.7225	G 4107, Class 2, SNB7
	Nuts	683-1-C45	A194/A194M Gr 2H	K04002	EN 10269	C 45 E	1.1191	G 4051, Cl S45C

Table H.2—Material Specifications for Pump Parts (Continued)

Material Class	Applications	ISO	USA		Europe			Japan
			ASTM	UNS [a]	EN [b]	Grade	Material No.	JIS
12 % Chrome Steel	Pressure Castings	—	A487/A487M Gr CA6NM	J91540	EN 10283	GX 4 Cr Ni 13-4	1.4317	G 5121, CI SCS 6, SCS 6X
	Impellers and Diffusers	—	A487/A487M Gr CA6NM	J91540	EN 10213	GX 4 Cr Ni 13-4	1.4317	G 5121, CI SCS 6 SCS 6X
		—	A743/A743M Gr CA6NM	J91540	EN 10213	GX 4 Cr Ni 13-4	1.4317	G 5121, CI SCS 6 SCS 6X
	General Castings	—	A487/A487M Gr CA6NM	J91540	EN 10213	GX 4 Cr Ni 13–4	1.4317	G 5121, CI SCS 6 SCS 6X
		—	A743/A743M Gr CA6NM	J91540	EN 10283	GX 4 Cr Ni 13–4	1.4317	G 5121, CI SCS 6, SCS 6X
		—	A426/426M Gr CPCA15	J91150	EN 10283	GX12Cr12	1.4011	G5121 SCS1/ SCS1X
	Wrought/ forgings: Pressure	683-13-3	A182/A182M Gr F6a	S41000	EN 10250-4	X12 Cr13	1.4006	G 3214, Gr SUS 410-A G 3214,
		—	Class 1 or 2 A182/A182M Gr F6 NM	S41500	EN 10222-5	X 3 Cr NiMo 13-4-1	1.4313	CI SUS F6 NM
	Wrought/ Forgings: General	683-13-2	A473 Type 410	S41000	EN 10088-3	X 12 Cr 13	1.4006	G 3214, Gr SUS 410-A
	Bar Stock: Pressure	683-13-3	A479/A479M Type 410	S41000	EN 10272	X12 Cr 13	1.4006	G 4303 Gr SUS 410 or 403
	Bar Stock: General	683-13-3	A276 Type 410	S41000	EN 10088-3	X 12 Cr 13	1.4006	G 4303 Gr SUS 410 or 403
	Bar Stock: Forgings [c]	683-13-4	A276 Type 420 A473 Type 416 A582/A582M Type 416	S42000 S41600	EN 10088-3	X 20 Cr 13 X 20 Cr S 13 X 20 Cr S 13	1.4021 1.4005 1.4005	G 4303, Gr SUS 420J1 or 420J2

Table H.2—Material Specifications for Pump Parts (Continued)

Material Class	Applications	ISO	USA		Europe			Japan
			ASTM	UNS [a]	EN [b]	Grade	Material No.	JIS
12 % Chrome Steel	Bolts and Studs [d]	3506-1, C4-70	A193/A193M Gr B6	S41000	EN 10269	X22CrMo V 12-1	1.4923	G 4303 Gr SUS 410 or 403
	Nuts [d]	3506-2, C4-70	A194/A194M Gr 6	S41000	EN 10269	X22CrMo V 12-1	1.4923	G 4303 Gr SUS 410 or 403
	Plate	683-13-3	A240/A240M Type 410	S41000	EN 10088-2	X 12 Cr 13	1.4006	G 4304/4305 Gr SUS 403 or 410
	Pipe [g] ≤ 500 °F (260 °C) (316L)	683-13-10	B444 N06625 A312 Type 316L	N06625 S31603	EN 10095	NiCr22-M09NB	2.4856	G 3459 SUS316LTP
	Pipe [g] > 500 °F (260 °C) (N06625)	—	B444 N06625	N06625	EN 10095	N9Cr22-Mo9NB	2.4856	G 4903 NCF625TP
	Fittings [g] ≤ 500 °F (260 °C) (316L)	9327-5 X2CrNiMo 17-12	B446 N06625 A182 Gr 316L	N06625 S31603	EN 10088-2	NiCr22-M09NB	2.4856 1.4404	G 3214 SUSF316L
	Fittings [g] > 500 °F (260 °C) (N06625)	—	B446 N06625(g)	N06625	EN 10088-2	NiCr22-M09NB	2.4856	G 4901 NCF625
Austenitic Stainless Steel	Pressure Castings	683-13-10	A351/A351M Gr CF3	J92500	BSI/BS/ EN 10213-4	GX2 Cr Ni 19-11	1.4309	G 5121, Cl SCS 19A
		683-13-19	A351/A351M Gr CF3M	J92800	BSI/BS/ EN 10213-4	GX2 Cr Ni Mo 19-11-2	1.4409	G 5121, Cl SCS 16A SCS 16AX
	General Castings	—	A743/A743M Gr CF3	J92500	EN 10283	GX2 Cr Ni 19-11	1.4309	G 5121, Cl SCS 19A
		—	A743/A743M Gr CF3M	J92800	EN 10283	GX2 Cr Ni Mo 19-11-2	1.4409	G 5121, Cl SCS 16A, SCS 16AX

Table H.2—Material Specifications for Pump Parts (Continued)

Material Class	Applications	ISO	USA		Europe			Japan
			ASTM	UNS [a]	EN [b]	Grade	Material No.	JIS
Austenitic Stainless Steel	Wrought/ forgings	9327-5, XCrNi18-10	A182/A182M Gr F 304L [h]	S30403	EN 10222-5	X2 Cr Ni 19-11	1.4306	G 3214, Gr SUS F 304 L [h]
		9327-5, XCrNiMo 17-12	A182/A182M Gr F 316L	S31603	EN 10222-5 EN 10250-4	X2 Cr Ni Mo 17-12-2	1.4404	G 4304/4305, Gr SUS 304L [h]/ 316L
		—	A403 Gr WP304L/ WP316L	S30403/ S31603	General Purpose: EN10250-4 Pressure Purpose: EN 10222-5	X2 Cr Ni 19-11 X2 Cr Ni Mo 17-12-2	1.4306 1.4404	G3214 SUSF304L/ SUSF316L G4034,4035 SUS304L/ SUS316L
	Bar stock [e]	9327-5 X2CrNi18-10	A479/A479M Type 304L [h] A479/A479M Type 316L A276 Grade 316L	S30403 S31603	EN 10088-3 EN 10088-3	X2 Cr Ni 19-11 X2 Cr Ni Mo 17-12-2	1.4306 1.4404	G 4303 Gr SUS 304 L [h] G 4303 Gr SUS 316 L
		9327-5 X2CrNiMo 17-12	A479/A479M Type XM19	S20910	—	—	—	—
	Plate	9328-5 X2CrNiMo 17-12-2	A240/A240M Gr 304L [h]/316L	S30403 S31603	EN 10028-7 EN 10028-7	X2 Cr Ni 19-11 X2 Cr Ni Mo 17-12-2	1.4306 1.4404	G 4304/4305, Gr SUS 304 L [h]/316 L
	Pipe	683-13-10 683-13-19	A312/A312M Type 304L [h], 316L	S30403 S31603	—	—	—	G 3459 Gr SUS 304 LTP [h]/ 316 LTP
	Fittings	9327-5, X2CrNi18-10 9327-5, X2CrNiMo 17-12	A182/A182M Gr F304L [h], Gr 316L	S30403 S31603	EN 10222-5	X2 Cr Ni 19-11 X2 Cr Ni Mo 17-12-2	1.4306 1.4404	G 3214 Gr SUS F304L [h]/ F316L
	Bolts and Studs	3506-1, A4-70	A193/A193M Gr B 8 M	S31600	EN 10250-4	X6 Cr Ni Mo Ti 17-12-2	1.4571	G 4303, Gr SUS 316

Table H.2—Material Specifications for Pump Parts (Continued)

Material Class	Applications	ISO	USA		Europe			Japan
			ASTM	UNS [a]	EN [b]	Grade	Material No.	JIS
	Nuts	3506-2, A4-70	A194/A194M Gr B 8 M	S31600	EN 10250-4	X6 Cr Ni Mo Ti 17-12-2	1.4571	G 4303, Gr SUS 316
Ni-Cu Alloy	General Castings	—	A494/A494M Gr M30C or M35-1	N24130 or N24135	—	NiCu30Fe	2.4360 or 2.4361	—
Duplex Stainless Steel	General Castings	—	A890/A890M Gr 1B	J93372	BSI/BS/ EN 10213-4	GX2 CrNiMo CuN-25-6-3-3	1.4517	G 5121, Gr SCS 11/SCS 10
		—	A890/890M Gr 3A	J93371				
		—	A890/890M Gr 4A	J9220				
	Pressure Castings	—	A995/A995M Gr 1 B	J93372	BSI/BS/ EN 10213-4	GX2 CrNiMo CuN 25-6-3-3	1.4517	G 5121, Gr SCS 10/SCS 11
		—	A995/A995M Gr 3A	J93371				
		—	A995/A995M Gr 4A	J9220				
	Wrought/ Forgings	9327-5, X2CrNiMoN 22-5-3	A182/A182M Gr F 51	S31803	EN 10250-4 EN 10222-5	X2CrNiMoN 22-5-3	1.4462	—
	Bar Stock	—	A479/A479M	S32550	EN 10088-3	X2CrNiMo CuN-25-6-3	1.4507	—
		9327-5, X2CrNiMo N22-5-3	A276-S31803	S31803	EN 10088-3	X2CrNiMoN 22-5-3	1.4462	B 2312/B 2316 Gr SUS 329 J3L
	Plate	—	A240/ A240M-S31803	S31803	EN 10028-7	X2CrNiMoN 22-5-3	1.4462	G 4304/G 4305 Gr SUS 329 J3L
	Pipe	—	A790/ A790M-S31803	S31803	—	—	—	G 3459 Gr SUS 329 J3LTP
	Fittings	9327-5, X2CrNiMo N22-5-3	A182/A182M Gr F 51	S31803	EN 10250-4 EN 10222-5	X2CrNiMo N22-5-3	1.4462	B 2312/B 2316 Gr SUS329J3L
		—	A815/A815M Gr WPS 31803	S31803	EN 10222-5	X2CrNiMo N-22-5-3	1.4462	B2312, B2316 SUS329J3L

Table H.2—Material Specifications for Pump Parts (Continued)

Material Class	Applications	ISO	USA		Europe			Japan
			ASTM	UNS [a]	EN [b]	Grade	Material No.	JIS
Duplex Stainless Steel	Bolts and Studs	—	A276-S31803	S31803	EN 10088-3	X2CrNiMo N22-5-3	1.4462	G 4303 Gr SUS 329 J3L
		—	A479/A479M S31803	S31803	EN 10088-3	X2CrNi MoN-22-5-3	1.4462	G 4303 SUS329J3L
	Nuts	—	A276-S31803	S31803	EN 10088-3	X2CrNiMoN 22-5-3	1.4462	G 4303 Gr SUS 329 J3L
		—	A479/A479M S31803	S31803	EN 10088-3	X2CrNiMo N-22-5-3	1.4462	G4303 SUS329J3L
Super Duplex Stainless Steel [f]	General Castings	—	A890/A890M Gr 5A	J93404	BSI/BS/ EN 10213-4	GX2Cr NiMoN26-7-4	1.4469	—
		—	A890/A890M Gr 6A	J93380				—
	Pressure Castings	—	A995/A995M Gr 5A	J93404	BSI/BS/ EN 10213-4	GX2Cr NiMoN26-7-4	1.4469	—
		—	A995/995M Gr 6A	J93380	—	—	—	—
	Wrought/ forgings	—	A182/A182M Gr F55 and F53	S32750 S32760	EN 10250-4 EN 10088-3	X2CrNi MoCuWN 25-7-4	1.4501	G 4303, Gr SUS 329 J4L
	Bar stock	—	A276-S32760 A479/A479M-S32750/ S32760	S32750 S32760	EN 10088-3	X2CrNi MoCuWN 25-7-4	1.4501	G 4304/G 4305 Gr SUS 329 J4L
	Plate	—	A240/A240M-S32750/ S32760	S32750 S32760	EN 10028-7	X2CrNi MoCuWN 25-7-4	1.4501	—

Table H.2—Material Specifications for Pump Parts (Continued)

Material Class	Applications	ISO	USA		Europe			Japan
			ASTM	UNS [a]	EN [b]	Grade	Material No.	JIS
Super Duplex Stainless Steel [f]	Pipe	—	A790/A790M-S32750/ S32760	S32750 S32760	—	—	—	G 3459, Gr SUS 329 J4LTP
	Fittings	—	A182/A182M Gr F55 and F53 WPS32750/ S32760	S32750 S32760	EN 10250-4 EN 10088-3	X2CrNi MoCuWN 25-7-4	1.4501	B 2312/B 2316 Gr SUS 329 J4L
	Bolts and Studs	—	A276-S32750 A276-S32760	S32750 S32760	EN 10088-3	X2CrNiMo CuWN 25-7-4	1.4501	G 4303 Gr SUS 329 J4L
	Nuts	—	A276-S32750 A276-S32760	S32750 S32760	EN 10088-3	X2CrNiMo CuWN 25-7-4	1.4501	G 4303 Gr SUS 329 J4L

[a] UNS (unified numbering system) designation for chemistry only.

[b] Where EN standards do not yet exist, European national standards are available, e.g. AFNOR, BS, DIN.

[c] Do not use for shafts in the hardened condition (over 302 HB).

[d] Special, normally use 4140 alloy steel.

[e] For shafts, standard grades of austenitic stainless steel may be substituted in place of low-carbon (L) grades.

[f] Super duplex stainless steel classified with pitting resistance equivalent (PRE) number greater than or equal to 40:

PRE→ $w_{Cr} + 3.3w_{Mo} + 16w_{N}$ where w is the percentage mass fraction of the element indicated by the subscript.

[g] For temperatures ≤ 500 °F (260 °C), 316L stainless steel or UN N06625 piping and fittings may be used. For temperatures > 500 °F (260 °C), only UN N06625 piping and fittings shall be used. See 6.4.2.12.

[h] Many applications that have moderate corrosion properties do not require the 316/316L and the option of 304/304L is suitable. The option to use 304/304L in. for those moderate corrosion applications that do not require the greater corrosion resistance of 316/316L is advised by the vendor.

Table H.3—Nonmetallic Wear Part Materials

Material	Temperature Limits °F (°C) min.	max.	Application
Polyether ether ketone (PEEK) Chopped-carbon-fiber filled	−20 (−30)	275 (135)	Stationary parts
PEEK Continuous-carbon-fiber wound	−20 (−30)	450 (230)	Stationary or rotating
Perfluoroalkoxy/Carbon-fiber (PFA/CF) Carbon-fiber Filled	−50 (−46)	450 (230)	Stationary parts
Carbon graphite Resin-impregnated	−55 (−50)	550 (285)	Stationary parts
Babbitt-impregnated	−450 (−260)	300 (150)	
Nickel-impregnated	−450 (−260)	750 (400)	
Copper-impregnated	−450 (−260)	750 (400)	

Nonmetallic wear part materials that are proven compatible with the specified process liquid may be proposed within the above limits. See 6.7.7 d).

Such materials may be selected as wear components for mating against a suitably selected metallic component such as hardened 12 % Cr steel or hard-faced austenitic stainless steel. Materials may be used beyond these limits if proven application experience can be provided and if approved by the purchaser.

Table H.4—Piping Materials

Component	Fluid					
	Auxiliary Process Liquid		Steam		Cooling Water	
	Category		Gauge Pressure PSI (bar; kPa)		Nominal Size	
	Pipeline Services [c]	All Weldable Materials	≤ 75 (5; 500)	> 75 (5; 500)	Standard ≤ 1 NPS (DN 25)	Optional ≥ 1 1/2 NPS (DN 40)
Pipe	Seamless [a]	Seamless [a]	Seamless [a]	Seamless [a]	—	Carbon Steel, (galvanized to ISO 10684 or ASTM A153/ A153M)
Tubing [b]	Stainless Steel (Seamless Type 316)	Stainless Steel (Seamless Type 316)	Stainless Steel (Seamless Type 316)	Stainless Steel (Seamless Type 316)	Stainless Steel (Seamless Type 316)	—
All valves	Class 800	Class 800	Class 800	Class 800	Class 200 Bronze	Class 200 Bronze
Gate and Globe Valve	Bolted Bonnet and Gland	Bolted Bonnet and Gland	Bolted Bonnet and Gland	Bolted Bonnet and Gland	—	—
Pipe fittings and unions	Forged Class 3000	Forged Class 3000	Forged Class 3000	Forged Class 3000	Malleable iron (galvanized to ISO 10684 or ASTM A153/ A153M)	Malleable iron (galvanized to ISO 10684 or ASTM A153/ A153M)
Tube Fittings	Manufacturer's Standard			Manufacturer's Standard	Manufacturer's Standard	—

Fabricated Joints ≤ 1 NPS (DN 25)	Threaded	Socket-welded or Butt-welded	Threaded	Socket-welded or Butt-welded	Threaded	—
Fabricated joints ≥ 1 1/2 NPS (DN 40)	—	—	—	—	—	Purchaser to specify
Gaskets	—	Austenitic stainless steel spiral-wound	–	Austenitic stainless steel spiral-wound	—	—
Flange Bolting [d]	—	4140 alloy steel	—	4140 alloy steel	—	—

[a] Schedule 80 shall be used for pipe sizes from NPS 1/2 to NPS 1 1/2 (DN 15 to DN 40); schedule 40 shall be used for sizes 2 NPS (DN 50) and larger.

[b] Acceptable tubing sizes (in accordance with ISO 4200) are the following:

1/2 in. dia. × 0.065 in. wall (12.7 mm dia. × 1.66 mm wall);
3/4 in. dia. × 0.095 in. wall (19 mm dia. × 2.6 mm wall);
1-in. dia. × 0.109 in. wall (25 mm dia. × 2.9 mm wall).

[c] Threaded connections only apply to pipeline services with a maximum operating temperature of 130 °F (55 °C). See 6.4.3.5.

[d] 4140 alloy steel bolting shall be coated (or purchaser-approved equal). (see 7.6.1.10).

Table H.5—Miscellaneous Material Specifications

Babbit	ASTM B 23, Grades 1–9, as required by vendor for service conditions
Bronze	UNS C87200 (silicon bronze), C90700 or C92200 (tin bronze), C95200 (aluminum bronze) or C95800 (nickel aluminum bronze)
Hard-facing	Type 3 tungsten carbide; overlay-weld deposit of 0.030 in. (0.8 mm) minimum finished thickness or, if available, a solid cast part of equal material may be substituted.
	Type 1 tungsten carbide as required for service conditions, with cobalt binder (solid part, not overlay); Type 2 tungsten carbide as required for service conditions, with nickel binder (solid part, not overlay); Type 3 tungsten carbide-sprayed overlay as required for service conditions.
Low-carbon nickel-molybdenum-chromium alloy	ASTM B564, UNS N10276 (forgings) ASTM B574, UNS N10276 (bar and rod) ASTM B575, UNS N10276 (plate, sheet and strip) ASTM A494, Grade CW-2M (weldable cast)
Nickel copper alloy	ASTM B564, UNS N04400 (forgings) ASTM B164, class A, UNS N04400 (bar and rod) ASTM B127, UNS N04400 (plate, sheet and strip) ASTM A494, Grade M30C (weldable cast)
Ni resist	ASTM A436, Type 1, 2 or 3, UNS F41000, F41002 and F41004 respectively (austenitic cast iron), ASTM A439, Type D2, UNS F43000 (austenitic ductile iron)
Precipitation-hardening nickel alloy	ASTM B637, UNS N07718 (forgings and bars) ASTM B670, UNS N07718 (plate, sheet, and strip)
Precipitation-hardening stainless steel	ASTM A564, Grade 630, UNS S 17400 or Grade 631, UNS 17700 (wrought) ASTM A747, Grade CB7Cu-1, UNS J92180 (cast)
Sheet gasket	Long-fiber material with synthetic rubber binder suitable for service conditions, or spiral-wound stainless steel and equal gasket material

Annex I
(normative)

Lateral Analysis

I.1 Lateral Analysis

I.1.1 General

If a wet lateral critical speed analysis is required (see 9.2.4.1.2), the method and assessment of results shall be as specified in I.1.2 through I.1.5. Table I.1 illustrates the analysis process. The method and assessment specified are particular to horizontal-axis liquid-handling turbomachines.

Table I.1—Rotor Lateral Analysis Logic Diagram

Step	If…	then…
1	Design the Pump	
2	Do purchaser and vendor agree that the pump and conditions of service are identical (see definition 3.1.16) to an existing pump with a proven operating record,	analysis is not necessary.
3	the rotor is subcritical (see definition 3.1.51),	analysis is not necessary.
4	neither 2 nor 3 is true,	analysis is required.

I.1.2 Natural Frequencies

The lateral analysis report shall state the following:

a) the rotor's first, second, and third “dry” bending natural frequencies (see 6.9.1.2);

NOTE 1 The “dry” bending natural frequencies serve as useful reference points for subsequent analysis of the damped natural frequencies.

NOTE 2 Usual design practice is to investigate overhung modes, coupling, and thrust collar and set their first bending natural frequency at separation margin at least 20 % above the highest potential excitation frequency (based on maximum continuous speed) before carrying out lateral analysis of the rotor.

b) all the rotor’s damped natural frequencies within the frequency range zero to two times (2×) maximum continuous speed, which shall be calculated for the speed range (for variable speed) for the rated flow at rated speed, minimum speed, and maximum speed; and for constant speed at MCSF, rated flow, and maximum flow, taking account of the following:

NOTE Rated flow at varying speeds will be calculated by affinity laws based on rated speed and flow.

1) stiffness and damping at the following internal running clearances at the expected temperature:

— as-new clearances, with water,

— as-new clearances, with the pumped liquid,

— two times (2×) the as-new clearances, with the pumped liquid;

2) stiffness and damping at the shaft seals (if labyrinth type);

3) stiffness and damping within the bearings for the average clearance and oil temperature. The effect of bearing stiffness and damping in pumps is generally minor in comparison to that of the internal running clearances; therefore, it is acceptable to analyze the bearings at their average clearance and oil temperature;

4) dynamic stiffness of the bearing support structure;

5) mass and inertia of the pump half-coupling hub and half the coupling spacer;

NOTE Though the higher order damped natural frequencies can be close to impeller vane passing frequency, there is no experience in liquid-handling turbomachines pointing to rotor-dynamics problems caused by such proximity. This is deemed a consequence of the complex mode shape(s), relatively low excitation energy, and enough damping at the higher frequencies involved.

6) added mass effects for impellers;

c) values or the basis of the stiffness and damping coefficients used in the calculation.

I.1.3 Separation Margins and Damping

The damping factors calculated in I.1.2 b) 1) shall be within the “acceptable” region shown on Figure I.1. If this condition cannot be satisfied, the unbalanced response, including damping, shall be determined (see I.1.4).

NOTE 1 In liquid-handling turbomachines, the first assessment of a rotor's dynamic characteristics is based on damping vs separation margin, rather than amplification factor vs separation margin. Two factors account for this basis. First, the rotor's natural frequencies increase with rotational speed, a consequence of the differential pressure across internal clearances also increasing with rotational speed. On a Campbell diagram (see Figure I.2), this means that the closer separations are between the running speed and natural frequencies rather than between the running speed and the critical speeds. Because the amplification factor at the closer separations is not related to synchronous (unbalance) excitation of the rotor, it can be developed only by an approximate calculation based on the damping. Second, employing damping allows the specification of a minimum value for natural frequency to running speed ratios from 0.4 to 0.8, thereby assuring that the rotor is free from significant subsynchronous vibration.

The logarithmic decrement δ, is related to the damping factor ξ, as given in Equation (I.1):

$$\delta = (2\pi\xi)/(1 - \xi^2)^{0.5} \quad \text{(I.1)}$$

For ξ up to 0.4, the approximate relationships given in Equation (I.2) between ξ, δ, and amplification factor, F_a, are accurate enough for practical purposes:

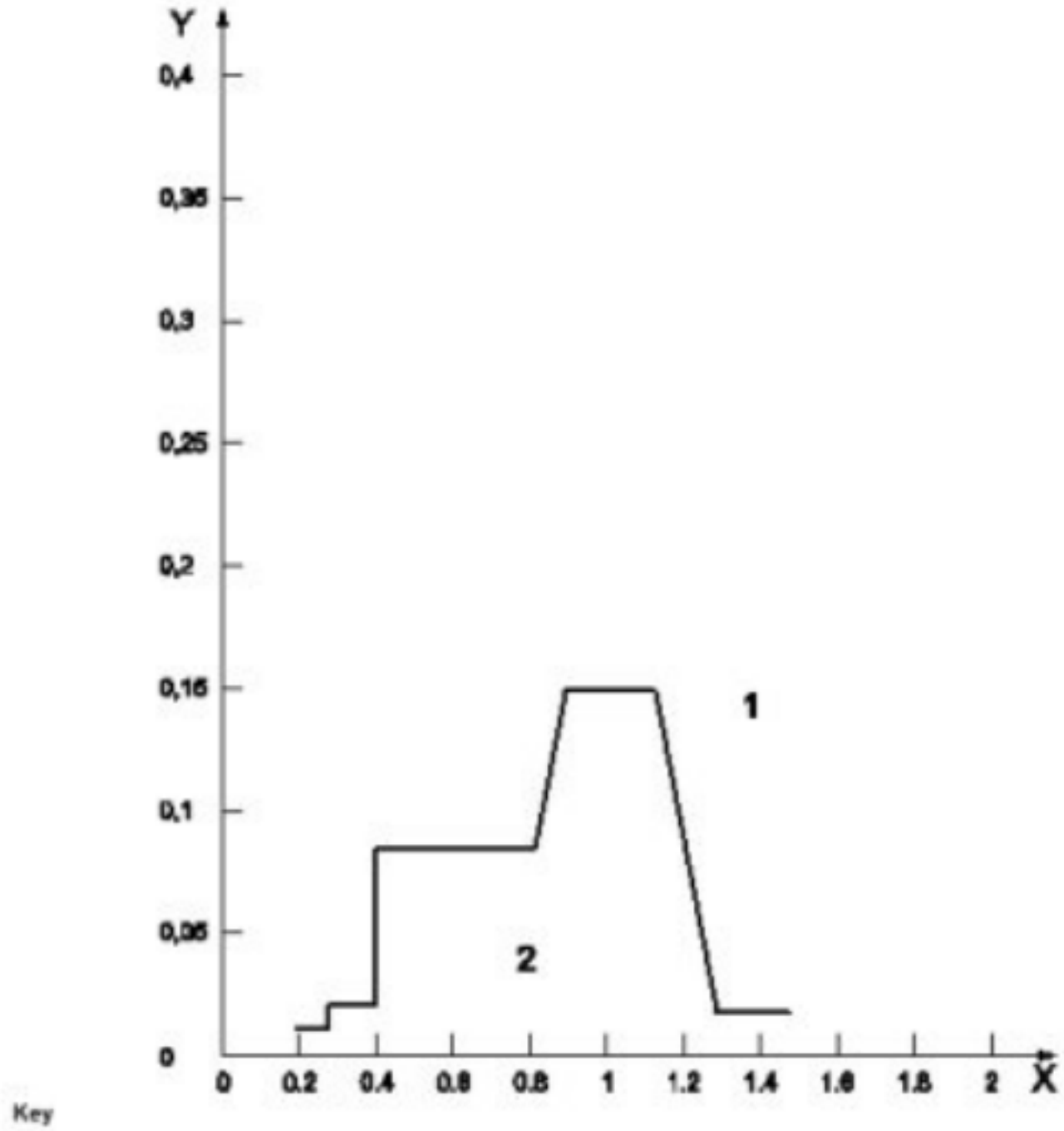
$$\xi = \delta/2\pi = 1/(2 \times F_a) \quad \text{(I.2)}$$

In liquid-handling turbomachines, critically damped conditions correspond to the following:

$$\xi \geq 0.15$$
$$\delta \geq 0.95$$
$$F_a \leq 3.33$$

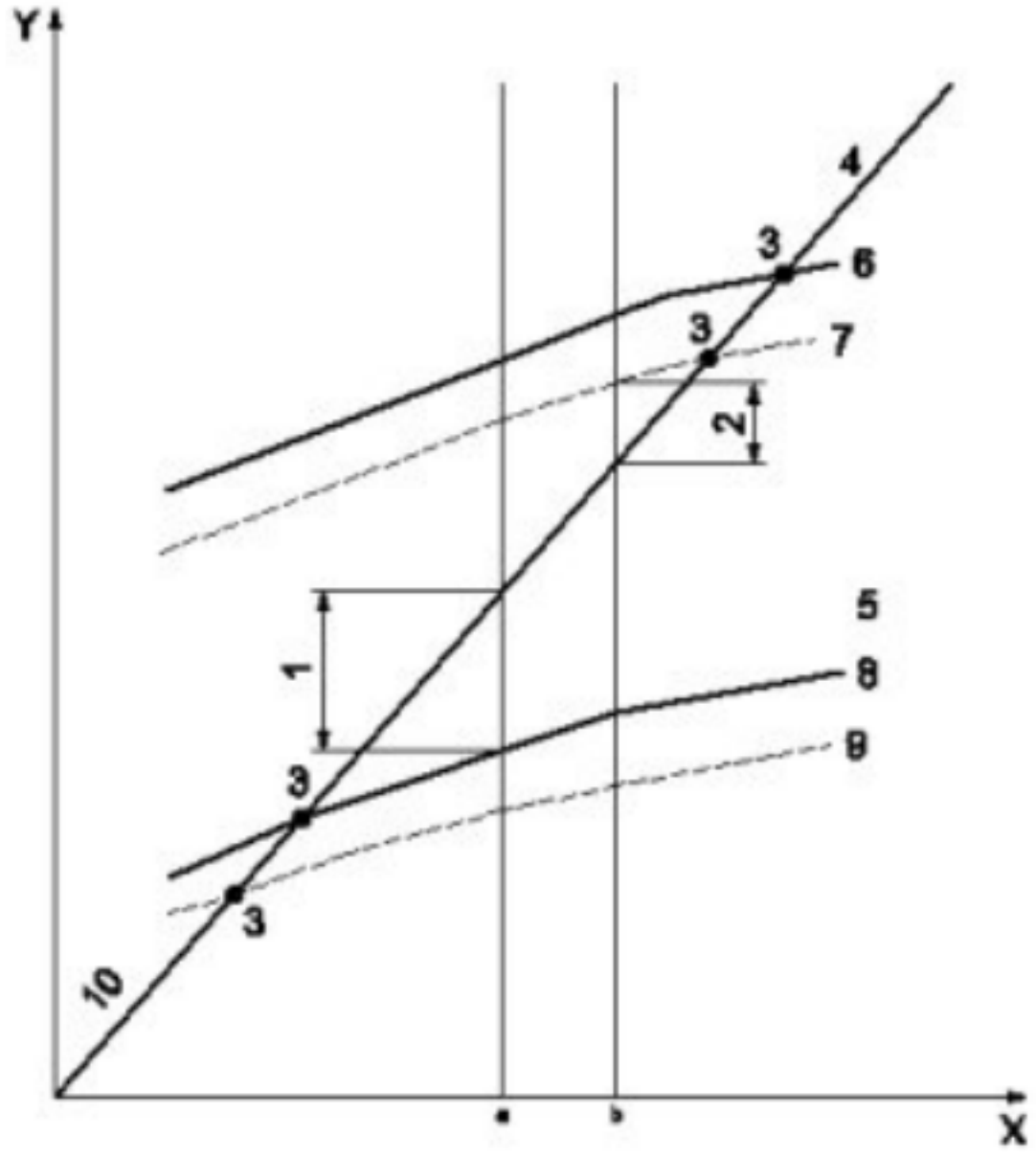
NOTE 2 The values given for critically damped conditions in liquid-handling turbomachines differ from those in API standards for gas- or vapor-handling turbomachines. The difference reflects successful operating experience with liquid-handling turbomachines designed using the values in this annex.

NOTE 3 Damping of $\xi \geq 0.08$ over the range f_{ni}/f_{run} 0.4 to 0.8 is supported by design and operating experience with liquid-handling turbomachines, which demonstrates that designs satisfying this requirement have not suffered problems with subsynchronous rotor vibration.



Key
X frequency ratio, f_{ni}/f_{run}
Y damping factor, ξ
1 acceptable region
2 unacceptable region

Figure I.1—Damping Factor vs Frequency Ratio



Key

X	pumping speed, expressed in revolutions per minute	6	as-new clearance
Y	frequency, f_n	7	2× clearance
1	minimum separation margin, 1st f_n	8	as-new clearance
2	minimum separation margin, 2nd f_n	9	2× clearance
3	critical speeds	10	run line
4	second bending	a	Minimum.
5	first bending	b	Maximum.

Figure I.2—Typical Campbell Diagram

I.1.4 Damped Unbalance Response Analysis

If the damping factor vs separation margin for a mode or modes is not acceptable by the criteria in Figure I.1, the rotor's damped response to unbalance shall be determined for the mode(s) in question on the following basis:

a) the pumped liquid;

b) clearance condition(s), as-new or two times (2×) as-new, causing inadequate separation margin vs damping;

c) total unbalance of four times (4×) the allowable value (see 9.2.4.2.2) lumped at one or more points to excite the mode(s) being investigated individually.

I.1.5 Allowable Displacement

The peak-to-peak displacement of the unbalanced rotor shall not exceed 35 % of the diametral running clearance at any close clearance location.

NOTE In centrifugal pumps, the typical damped response to unbalance does not show a peak in displacement at resonance large enough to assess the amplification factor. With this limitation, assessment of the damped response to unbalance is restricted to comparing rotor displacement to the available clearance.

I.2 Shop Verification of Rotor Dynamic Characteristics

[•] **I.2.1** If specified, the dynamic characteristics of the rotor shall be verified during the shop test. The rotor's actual response to unbalance shall be the basis for confirming the validity of the damped lateral analysis. This response is measured during either variable-speed operation from rated speed down to 75 % of the first critical speed or during coast-down. If the damped response to unbalance was not determined in the original rotor analysis (see I.1.4), this response shall be determined for a pump with new clearances handling water before proceeding with shop verification. The test unbalances shall be vectorially added in phase with the residual unbalance, at locations determined by the manufacturer (usually at the coupling or thrust collar).

NOTE The principal objective of shop verification by response to unbalance is to verify the existence of a critical speed (vibration peak) within the tolerance of the calculated value or, if the analysis predicted a highly damped critical speed, the absence of a vibration peak within tolerance of the calculated value. Shop verification by this method is feasible only for pumps that have sleeve bearings and that are furnished with proximity probe pairs at each journal bearing.

I.2.2 The magnitude and location of the test unbalance(s) shall be determined from a calibration of the rotor's sensitivity to unbalance. The calibration shall be performed by obtaining the vibration orbits at each bearing, filtered to rotor speed (1×), during two trial runs as follows:

a) with the rotor as-built;

b) with trial unbalance weights added 90° from the maximum displacement in Run a).

The magnitude of the test unbalances shall be such that the calculated maximum shaft displacement caused by the resultant total unbalance (residual plus test) is 150 % to 200 % of the allowable displacement from Table 8 or Table 9 at the bearing probes but shall not exceed eight times the maximum allowable rotor unbalance.

I.2.3 During the test, the rotor's speed, vibration displacement, and corresponding phase angle, filtered to rotor speed (1×), shall be measured and recorded.

I.2.4 The rotor's characteristics shall be considered verified if the following requirements are met:

a) observed critical speed(s) (distinct vibration peak and appropriate phase shift) within ± 10 % of the calculated value(s);

b) measured vibration amplitudes within 35 % of the calculated values.

Highly damped critical speeds might not be observable; therefore, agreement between measured and predicted response is verification of the analysis.

I.2.5 If the acceptance criteria given in I.2.4 are not met, the stiffness or damping coefficients, or both, used in the natural frequency calculation shall be adjusted to produce agreement between the calculated and measured results. The coefficients of one type of element, annular clearances with $L/D < 0.15$, annular clearances $L/D > 0.15$, impeller interaction, and bearings shall be adjusted with the same correction factor. Once agreement is reached, the same correction factors shall be applied to the calculation of the rotor's natural frequencies and damping for the pumped liquid, and the rotor's separation margins vs damping factors rechecked for acceptability.

Of the coefficients used in rotor lateral analysis, those for damping in annular clearances have the highest uncertainty and are, therefore, usually the first that are adjusted. The stiffness coefficients of annular clearances typically have a low uncertainty and, therefore, shall be adjusted only on the basis of supporting data. Adjustments of bearing coefficients require specific justification because the typical values are based on reliable empirical data.

I.2.6 Alternative methods of verifying the rotor's dynamic characteristics, e.g. variable-frequency excitation with the pump at running speed to determine the rotor's natural frequencies, are available. The use of alternative methods and the interpretation of the results shall be agreed between the purchaser and manufacturer.

I.3 Documentation

I.3.1 The report on a lateral analysis shall include the following:

a) rotor data used for the analysis, which may include the rotor model, seal and bearing coefficients, and supporting data;

b) Campbell diagram (see Figure I.2);

c) plot of damping factor vs frequency ratio;

d) mode shape for all modes that fall within the speed range investigated;

e) conclusions and recommendations resulting from the analysis;

f) Bode plot(s) from shop verification by unbalance (see I.2.3);

g) summary of analysis corrections to reach agreement with shop verification (see I.2.5).

Items f) and g) shall be furnished only if the activity documented is required by the analysis or if specified by the purchaser.

I.3.2 The rotor design information included in the report shall match the physical design of the pump rotor, including major dimensions, diameters, and weights.

NOTE API 684 can be used for additional references for rotor dynamic analysis.

Annex J
(normative)

Procedure for the Verification of Residual Unbalance

J.1 General

This annex describes a procedure to verify residual unbalance in rotors by determining the calibration accuracy of the balancing equipment. Balancing machines may be configured to display the amount of rotor unbalance; however, the calibration can be in error. To determine the actual residual unbalance, a known amount of unbalance should be added using an appropriate procedure.

J.2 Residual Unbalance

Residual unbalance is the amount of unbalance remaining in a rotor after balancing. Residual unbalance shall be expressed in g-mm (g-in.).

J.3 Maximum Allowable Residual Unbalance

J.3.1 The maximum allowable residual unbalance, per plane, shall be calculated in accordance with Equation (J.1).

J.3.2 The static weight on each journal shall be determined by rotordynamic calculations. If the static loadings cannot be obtained from rotordynamic calculations, then the method by which the journal weight was determined shall be identified. It should NOT be assumed that rotor weight is equally divided between the two journals. There can be great discrepancies in the journal weight to the point of being very low (even negative on overhung rotors).

J.4 Residual Unbalance Check

J.4.1 General

J.4.1.1 When the balancing machine readings indicate that the rotor has been balanced within the specified tolerance, a residual unbalance check shall be performed before the rotor is removed from the balancing machine. Record and plot the indicated residual unbalance heavy spot of both planes on the Residual Unbalance Worksheet (one for each plane).

NOTE Due to the possibility of machine calibration errors, the residual unbalance check can be performed prior to final correction of the unbalance, typically after the placement of temporary weights.

J.4.1.2 To check the residual unbalance, a known trial weight, equal to the multiplier from Table J.1 times the maximum allowable unbalance from Equation J.1, is attached to the rotor at the same angular location as the indicated heavy spot. The check is run at each balance machine readout plane, and the readings in each plane are tabulated. This run is then repeated with the weight placed 180° opposite of the heavy spot at the same radius. The check is run at each balance machine readout plane, and the readings in each plane are tabulated.

Table J.1—Trial Weight Multiplier vs N_{mc}

Maximum Continuous Speed (N_{mc}) of Part/Assembly	Trial Weight Multiplier
$N_{mc} \leq 7500$ rpm	1.5
$\leq 7500 < N_{mc} \leq 12{,}500$ rpm	2.0
$N_{mc} > 12{,}500$ rpm	2.5

J.4.2 Procedure

J.4.2.1 Select a trial weight and radius that will be equivalent to the trial weight multiplier times the maximum allowable residual unbalance as defined by Equation (J.1)

NOTE If U_r is 19.2 g-in. (488.4 g-mm), for a rotor with MCS £ 7500 rpm, the trial weight magnitude should equal 28.8 g-in. (732.6 g-mm).

In g-mm units:

$$U_r = 6350\frac{W}{N_{mc}} \text{ (for } N_{mc} < 25{,}000 \text{ rpm)} \quad \text{(J.1a)}$$

$$U_r = \frac{W}{3.937} \text{ (for } N_{mc} \ ^3 \ 25{,}000 \text{ rpm)}$$

Where W is static journal weight in kilograms, Kg.

In g-in units:

$$U_r = 113.4\frac{W}{N_{mc}} \text{ (for } N_{mc} < 25{,}000 \text{ rpm)} \quad \text{(J.1b)}$$

$$U_r = \frac{W}{220.46} \text{ (for } N_{mc} \ ^3 \ 25{,}000 \text{ rpm)}$$

where

W is static journal weight in pound mass, Lbm.

J.4.2.2 At the heavy spot, add the first trial weight at the selected radius in J.4.2.1 to the first balance readout plane. Trial weight magnitude is a linear function with radial location. Every effort should be made to place the weight accurately, both radially and circumferentially.

J.4.2.3 Verify that the balancing machine's readings are stable without faulty sensors or displays.

NOTE When the trial weight is added to the last known heavy spot, the first meter reading should easily exceed the balance tolerance in that plane. Little or no meter reading generally indicates that the rotor was not balanced to the correct tolerance, the balancing machine was not sensitive enough, or that a balancing machine fault exists (i.e. a faulty pickup).

J.4.2.4 Remove the trial weight and rotate the trial weight to the second position (that is 180° from the initial trial weight position). All verification shall be performed using only one sensitivity range on the balance machine.

J.4.2.5 Record and plot the balancing machine unbalance amplitude and phase readout (heavy spot) on the Residual Unbalance Worksheet (Figure J.1) for the readout plane in question. If the indicated unbalance phase angle for the residual unbalance (J.4.1.1) differs by more than 10° from the first trial weight phase angle or the 2nd trial weight phase angle plus 180°, then the angular location of the trial weight should be adjusted to lessen

the difference. Once the phase angle difference is less than 10°, the actual amount of residual unbalance (refer to worksheets, Figure J.2 and Figure J.3) can be calculated.

J.4.2.5.1 The difference in magnitudes of each trial weight run relative to the indicated unbalance should be within 20 %.

J.4.2.5.2 If this is exceeded, weight placement and magnitude should be reviewed. A larger trial weight can be used with the value entered into the "User Selected Trial Weight", otherwise this value should be zero.

NOTE 1 Not meeting this tolerance will generate errors in the calculated residual unbalance or indicate a problem with the balance machine.

NOTE 2 In Figure J.3, a larger trial weight was needed to enable the 2nd reading to be 180° different than the indicated reading (basically cross over the center of the plot.) The larger trial weight (9 gr) was inputted into the "User Selected Trial Weight" cell. This value of the trial weight "TW" is then used to calculate the "Actual Residual Unbalance". For this example, the machine was determined to be reading $^1/_2$ of the actual unbalance.

J.4.2.6 Repeat the steps described in J.4.2.1 through J.4.2.5 for each balance machine readout plane. If the specified maximum allowable residual unbalance has been exceeded in any balance machine readout plane when calculating the actual residual unbalance, the rotor shall be balanced more precisely and checked for compliance using the calibration factors determined above.

J.4.2.7 For stack component balanced rotors, a residual unbalance check shall be performed after the addition and balancing of the rotor after the addition of the first rotor component and at the completion of balancing of the entire rotor, as a minimum.

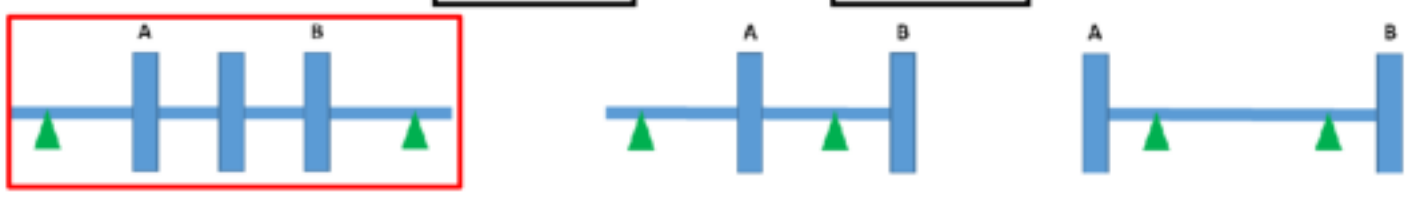
NOTE 1 This ensures that time is not wasted, and rotor components are not subjected to unnecessary material removal in attempting to balance a multiple component rotor with a faulty balancing machine.

NOTE 2 For large multistage rotors, the journal reactions can be considerably different from the case of a partially stacked to a completely stacked rotor.

Customer:
Job / Project Number:
OEM Equipment S/N:
Rotor Identification Number:
Repair Purchase Order Number:
Vendor Job Number:
Correction Plane (A or B) - see sketch:

Balancing Speed:
Maximum Continuous Operating Speed: Nmc rpm
Static Journal Weight Closest to This Plane: W kg lbm
W obtained from: Rotordynamics Specify if Other
Correction Radius - radius at which the readings are taken and trial weight will be placed: R mm in

Balance Plane Description: End A End B



Calculate the Maximum Allowable Residual Unbalance (Ur):

Nmc ≤ 25000 rpm Nmc > 25000 rpm

SI Units:

$$U_r = \frac{6350 * W}{N_{mc}} \quad \text{or} \quad U_r = \frac{W}{3.937}$$

Ur g-mm Unbalance @ Balance Plane g

Customary Units: (USC expressed in grams)

$$U_r = \frac{113.4 * W}{N_{mc}} \quad \text{or} \quad U_r = \frac{W}{220.46}$$

Ur g-in Unbalance @ Balance Plane g

Trial Weight Calculation:

$$TW = \frac{Um * U_r}{R}$$

Trial Weight Multiplier (Um):

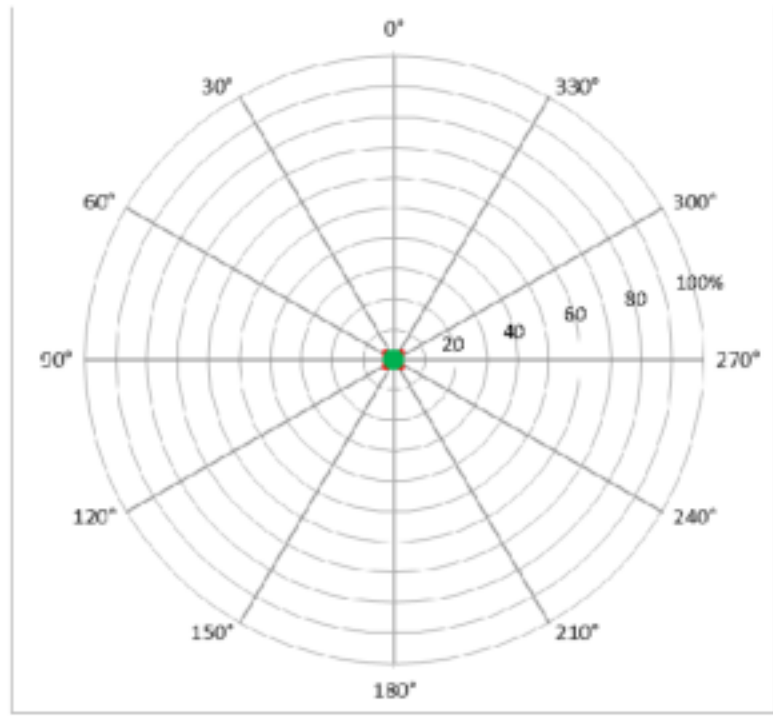
Recommended Trial Weight: g

User Selected Trial Weight: 0.00 g

Record Indicated Residual Unbalance (J.4.1.1) and the Indicated Unbalance with Trial Weight (J.4.2.5)

Test Data

Unbalance Readings		Magnitude grams (g)		Phase Angle degrees (°)
A: Residual Unbalance (IR)*				
B: Unbalance w/ 1st Trial Weight (Mx)*				
C: Unbalance w/ 2nd Trial Weight (Mn)*				
	Y = Mx - Mn	0.00	Angle Diff |B-A| <10	0.00
	Z = Mx + Mn	0.00	Angle Diff |C-A| <10	180.00
Trial Weight Effect	R1 = Mx - IR	0.00	R2 = Mn + IR	0.00
	R1/R2		R2/R1	
Uniformity Of Trial Weight Placement & Effect:		Caution: Potential Inaccuracies		
Angle diff < 10 and Magnitude ratio (0.8 < (R1/R2 & R2/R1) < 1.2)		*If contained, increase weight or adjust weight placement angle*		
		Metric (g-mm)		Customary (g-in)
Residual Unbalance Reading (IR*R)				
Actual Residual Unbalance (AR=R*TW*Y/Z)				
Within Specification (AR < Ur)				



100% = #NUM! g

** denotes indicated values*

Indicated Residual Unbalance
Indicated Unbalance w/ Trial Weight #1
Indicated Unbalance w/ Trial Weight #2

Indicates Input Required

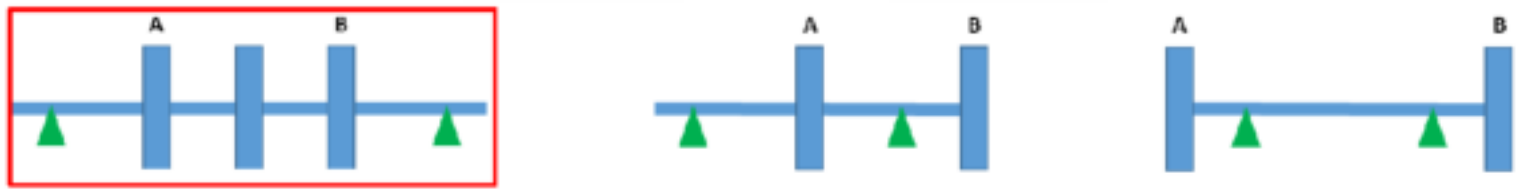
These cells should be hidden

Figure J.1—Residual Unbalance Worksheet

Customer:	International Oil Company
Job / Project Number:	JN 2398-IRG-48
OEM Equipment S/N:	345-678
Rotor Identification Number:	C4784XL
Repair Purchase Order Number:	
Vendor Job Number:	123634-297483
Correction Plane (A or B) - see sketch:	Plane A

Balancing Speed:		400	rpm		
Maximum Continuous Operating Speed:	Nmc	11000	rpm		
Static Journal Weight Closest to This Plane:	W	1300	kg		lbm
W obtained from:		Rotordynamics			Specify if Other
Correction Radius - radius at which the readings are taken and trial weight will be placed:	R	200	mm		in

Balance Plane Description: Thrust End A | Coupling End B



Calculate the Maximum Allowable Residual Unbalance (Ur):

Nmc ≤ 25000 rpm Nmc > 25000 rpm

SI Units:

$$U_r = \frac{6350 * W}{N_{mc}} \quad \text{or} \quad U_r = \frac{W}{3.937}$$

Ur		Unbalance @ Balance Plane	
750.45	g-mm	3.75	g

Customary Units: (USC expressed in grams)

$$U_r = \frac{113.4 * W}{N_{mc}} \quad \text{or} \quad U_r = \frac{W}{220.46}$$

Ur		Unbalance @ Balance Plane	
	g-in		g

Trial Weight Calculation:

$$TW = \frac{Um * U_r}{R}$$

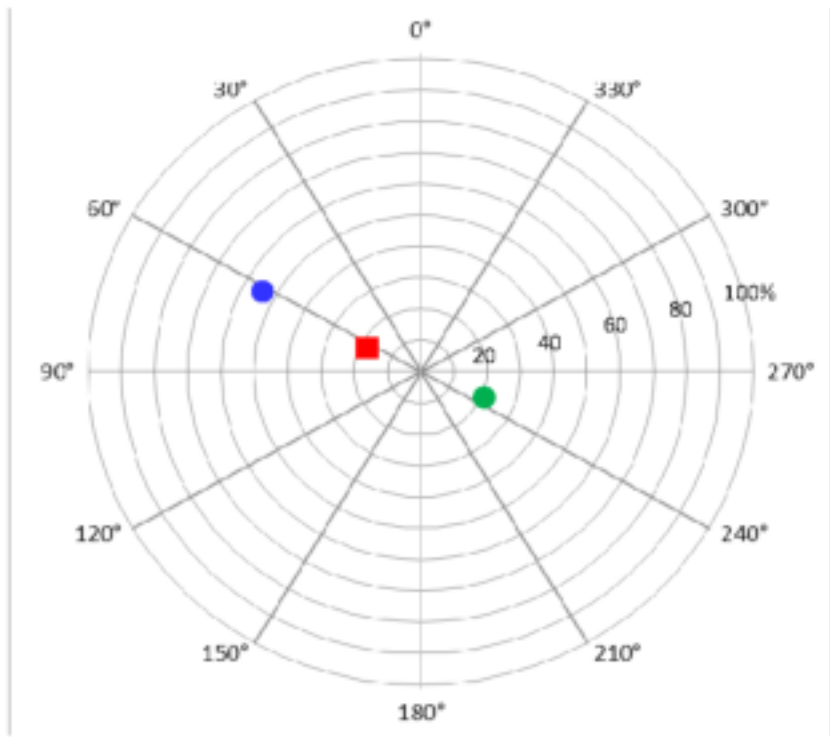
Trial Weight Multiplier (Um): 2

Recommended Trial Weight:		User Selected Trial Weight:	
7.50	g	0.00	g

Record Indicated Residual Unbalance (J.4.1.1) and the Indicated Unbalance with Trial Weight (J.4.2.5)

Test Data

Unbalance Readings		Magnitude grams (g)		Phase Angle degrees (°)
A: Residual Unbalance (IR)*		3.5		65
B: Unbalance w/ 1st Trial Weight (Mx)*		10.7		62
C: Unbalance w/ 2nd Trial Weight (Mn)*		4.2		247
	Y = Mx - Mn	6.50	Angle Diff |B-A| <10	3.00
	Z = Mx + Mn	14.90	Angle Diff |C-A| <10	2.00
Trial Weight Effect	R1 = Mx - IR	7.20	R2 = Mn + IR	7.70
	R1/R2	0.94	R2/R1	1.07
Uniformity Of Trial Weight Placement & Effect:		Acceptable		
Angle diff < 10 and Magnitude ratio [0.8 < (R1/R2 & R2/R1) < 1.2]		*If cautioned, increase weight or adjust weight placement angle*		
		Metric (g-mm)		Customary (g-in)
Residual Unbalance Reading (IR*R)		700.00		
Actual Residual Unbalance (AR=R*TW*Y/Z)		654.76		
Within Specification (AR < Ur)		YES		



100% = 20 g

* *denotes indicated values*

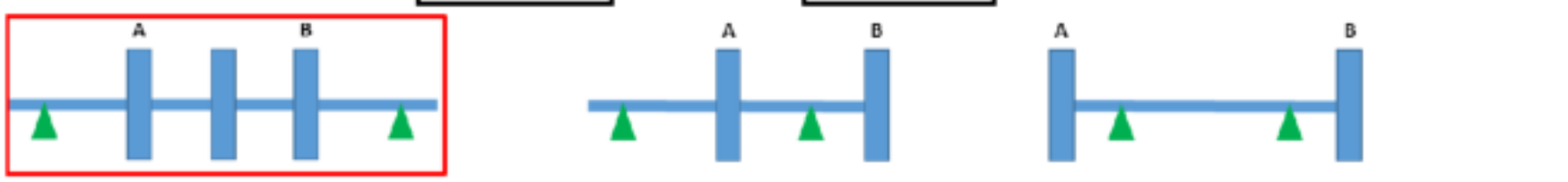
- Indicated Residual Unbalance
- Indicated Unbalance w/ Trial Weight #1
- Indicated Unbalance w/ Trial Weight #2

Indicates Input Required

Figure J.2—Sample Residual Unbalance Worksheet for Left Plane (metric)

Customer:		International Oil Company	
Job / Project Number:		JN 2398-IRG-48	
OEM Equipment S/N:		345-678	
Rotor Identification Number:		C4784XL	
Repair Purchase Order Number:			
Vendor Job Number:		123634-297483	
Correction Plane (A or B) - see sketch:		Plane A	
Balancing Speed:		400 rpm	
Maximum Continuous Operating Speed:	Nmc	11000 rpm	
Static Journal Weight Closest to This Plane:	W	kg	2866 lbm
W obtained from:		Rotordynamics	Specify if Other
Correction Radius - radius at which the readings are taken and trial weight will be placed:	R	mm	8 in

Balance Plane Description: Thrust End A — Coupling End B



Calculate the Maximum Allowable Residual Unbalance (Ur):

Nmc ≤ 25000 rpm — Nmc > 25000 rpm

SI Units:

$$U_r = \frac{6350 * W}{N_{mc}} \quad \text{or} \quad U_r = \frac{W}{3.937}$$

Ur		Unbalance @ Balance Plane	
	g-mm		g

Customary Units: (USC expressed in grams)

$$U_r = \frac{113.4 * W}{N_{mc}} \quad \text{or} \quad U_r = \frac{W}{220.46}$$

Ur		Unbalance @ Balance Plane	
29.55	g-in	3.69	g

Trial Weight Calculation:

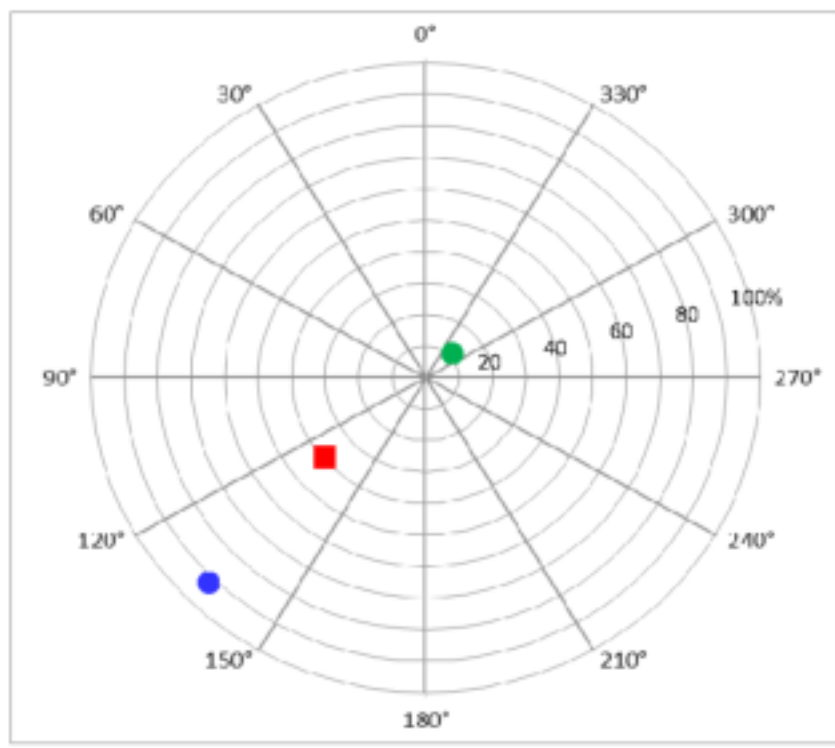
$$TW = \frac{Um * U_r}{R}$$

Trial Weight Multiplier (Um):	Recommended Trial Weight:		User Selected Trial Weight:	
2	7.39	g	0.00	g

Record Indicated Residual Unbalance (J.4.1.1) and the Indicated Unbalance with Trial Weight (J.4.2.5)

Test Data

Unbalance Readings		Magnitude grams (g)		Phase Angle degrees (°)
A: Residual Unbalance (IR)*		3.5		130
B: Unbalance w/ 1st Trial Weight (Mx)*		8.2		135
C: Unbalance w/ 2nd Trial Weight (Mn)*		1		312
	Y = Mx - Mn	7.20	Angle Diff |B-A| <10	5.00
	Z = Mx + Mn	9.20	Angle Diff |C-A| <10	2.00
Trial Weight Effect	R1 = Mx - IR	4.70	R2 = Mn + IR	4.50
	R1/R2	1.04	R2/R1	0.96
Uniformity Of Trial Weight Placement & Effect:		Acceptable		
Angle diff < 10 and Magnitude ratio (0.8 < (R1/R2 & R2/R1) < 1.2)		*if cautioned, increase weight or adjust weight placement angle*		
		Metric (g-mm)		Customary (g-in)
Residual Unbalance Reading (IR*R)				28.00
Actual Residual Unbalance (AR=R*TW*Y/Z)				46.25
Within Specification (AR < Ur)				NO



100% = 9 g

** denotes indicated values*

- Indicated Residual Unbalance
- Indicated Unbalance w/ Trial Weight #1
- Indicated Unbalance w/ Trial Weight #2

Indicates Input Required

Figure J.3—Sample Residual Unbalance Worksheet for Right Plane (USC)

Annex K
(informative)

Shaft Stiffness and Bearing System Life

K.1 Guideline on Shaft Stiffness for Overhung Pump Types OH2 and OH3

K.1 presents a standardized method for calculation of overhung pump SFI.

The design and operation requirements for overhung pump rotors are detailed in several areas of this standard. This subsection lists these requirements and establishes a standardized process of calculating a SFI that may be used to evaluate these latter parameters and to establish a baseline for the comparison of shaft flexibility.

For a shaft of two diameters, D_1 under the seal sleeve and D_2 between the bearings (see Figure K.1), the shaft stiffness is inversely proportional to what is generally termed the shaft flexibility index, SFI or I_{SF}, defined as given in Equation (K.1):

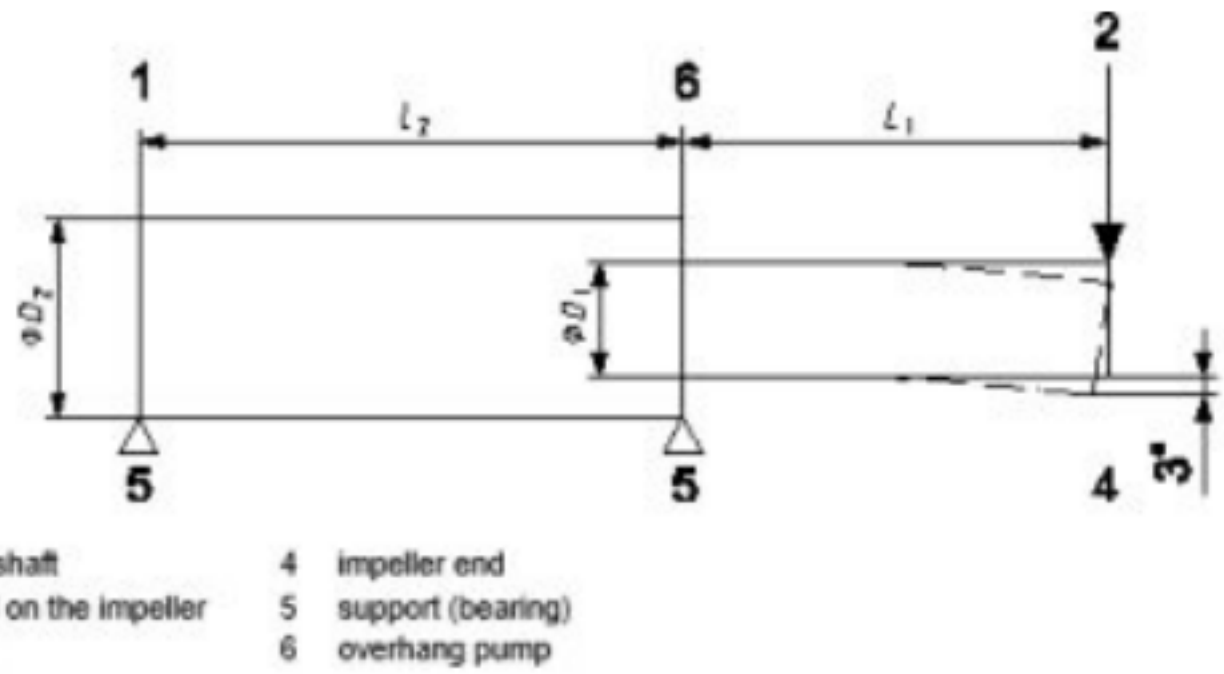
$$I_{SF} = L_1{}^3/D_1{}^4 + L_1 L_2{}^2/D_2{}^4 \quad \text{(K.1)}$$

where

L_1 is the overhang centerline of impeller to line bearing) and L_2 is the bearing span.

In rotor designs typical of refinery pumps, $D_2 > D_1$ and $L_2 < L_1$, the second term accounts for only about 20 % of the total value of I_{SF}, so the convention is to assess overhung pump shaft stiffness using a shortened expression as given in Equation (K.2):

$$I_{SF} = L_1{}^3/D_1{}^4 \quad \text{(K.2)}$$



Key

1	simplified shaft	4	impeller end
2	radial load on the impeller	5	support (bearing)
3	deflection	6	overhang pump

Figure K.1—Simple Overhung Rotor

The shortened calculation for I_{SF}, Equation (K.2), was used extensively by refiners in the 1970s and 1980s to compare overhung pump rotor stiffness and assign a maintenance cost factor to the price of pumps whose I_{SF} was some multiple, typically 1.2, of the lowest value of the pumps offered for the service. That practice led to the development of the higher stiffness rotors necessary to achieve longer pump mean time between repair (MTBR) and in the 1990s, lower shaft-seal leakage for reduced VOC emissions. Because the assessments were applied to pumps for a given application, the comparison was between pumps of similar size. To provide general guidance on values of I_{SF}, it is, therefore, necessary to relate I_{SF} to pump size.

Overhung-refinery-pump bearing frames are designed in discrete sizes. The shaft for each is, therefore, designed for the maximum torque, impeller mass, and impeller radial load (static and dynamic) of the largest liquid end the frame is intended to serve. Impeller mass is significant in that it is necessary for the rotor's first dry bending critical speed to be 120 % of the pump's maximum continuous operating speed (see 6.9.1.2). At the same time, deflection at the seal faces produced by radial thrust shall not exceed 0.002 in. (50 µm) (6.6.16).

The loads imposed on the shaft are directly related to the size of the impeller and, hence, to the flow, total head and speed of the pump. This allows the definition of a "size" factor, K_t, as given in Equation (K.3):

$$K_t = (Q \times H)/N \tag{K.3}$$

where

Q is flow at BEP of maximum diameter impeller;

H is the corresponding total head;

N is the rotational speed.

The size factor is related to torque. A log-log plot of I_{SF} vs K_t for various overhung pump designs ranging from 35 hp to 500,000 hp (25 kW to 350,000 kW), with the higher value for large pump-turbines, shows that the data for modern designs fall about a straight line; see Figure K.2 and Figure K.3. The line of best fit through the high side of the data is also shown.

The high-side line of best fit is defined by Equation (K.4) in SI units and by Equation (K.5) for USC units:

$$I_{SF,SI} = 32 \times K_t^{-0.76} \tag{K.4}$$

$$I_{SF,USC} = 6200 \times K_t^{-0.76} \tag{K.5}$$

Equations (K.4) and (K.5) cover refinery pumps whose rotors, at the limit of each frame, reportedly just meet the static deflection and rotor dynamics requirements of this standard based on rated speeds up to 3600 rpm. The liquid-end designs generally have double volutes for all pumps of 4 in. (100 mm) discharge and larger. In some cases the designs may be limited to 3000 rpm.

Figure K.2 and Figure K.3 or Equations (K.4) and (K.5) can be used to make a first assessment of the rotor stiffness of a given overhung pump design or a number of similar designs for a given application. An overhung pump whose I_{SF} is more than 1.2 times the chart or equation value is cause to seek justification of the design from the pump vendor.

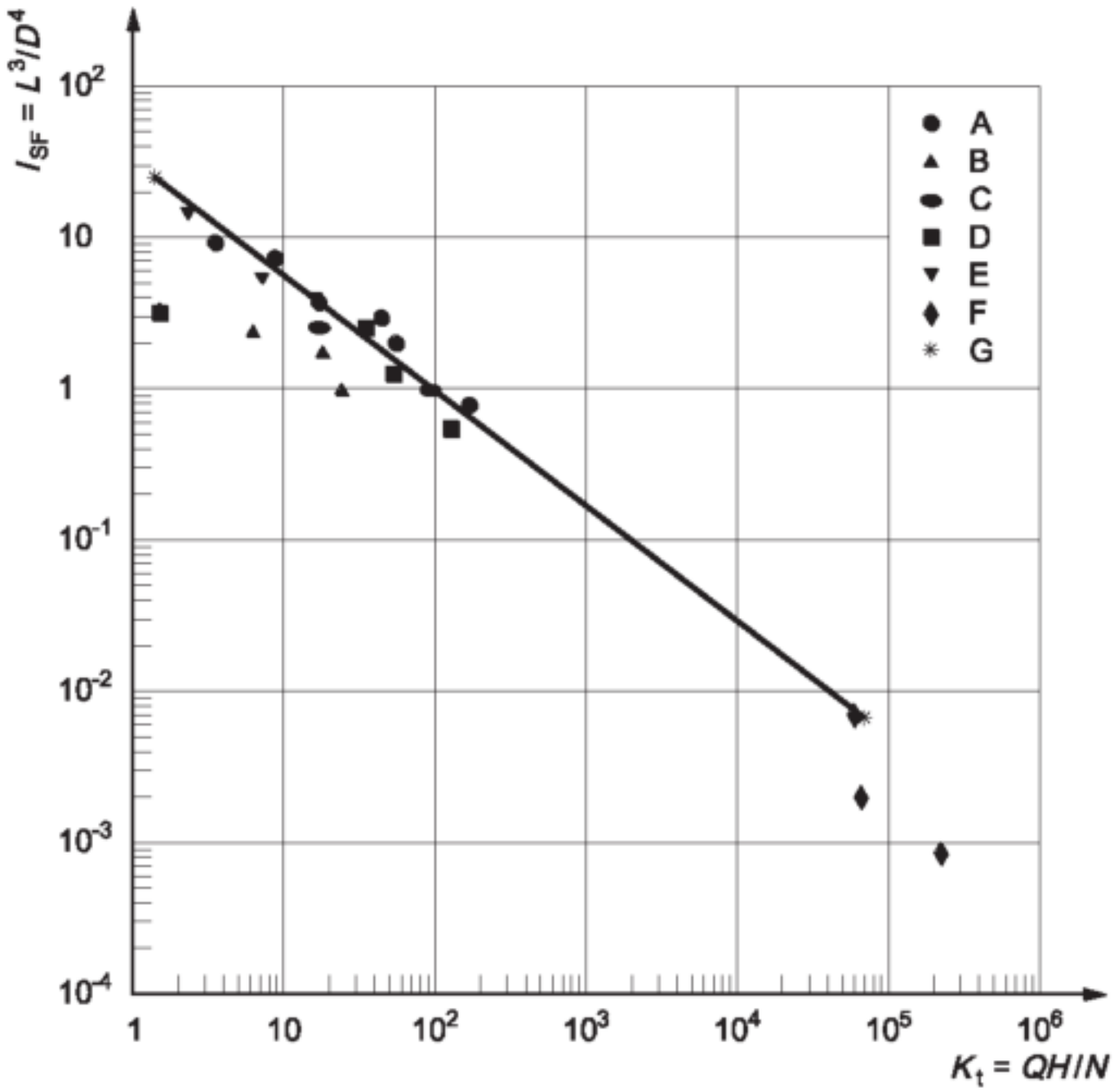


Figure K.2—Overhung Pump Shaft Flexibility Index vs Size Factor (SI Units)

K.2 Bearing System Life Considerations for OH2, OH3, BB1, BB2, and BB3 Pumps

K.2 presents a method of calculating bearing system life. If specified, bearing system life calculations shall be furnished (see 6.10.1.14).

Bearing system life (the calculated life of the combined system of bearings in the pump) shall be equivalent to at least 25,000 h with continuous operation at rated conditions, and at least 16,000 h at maximum radial and axial loads and rated speed. This section contains a discussion of these requirements.

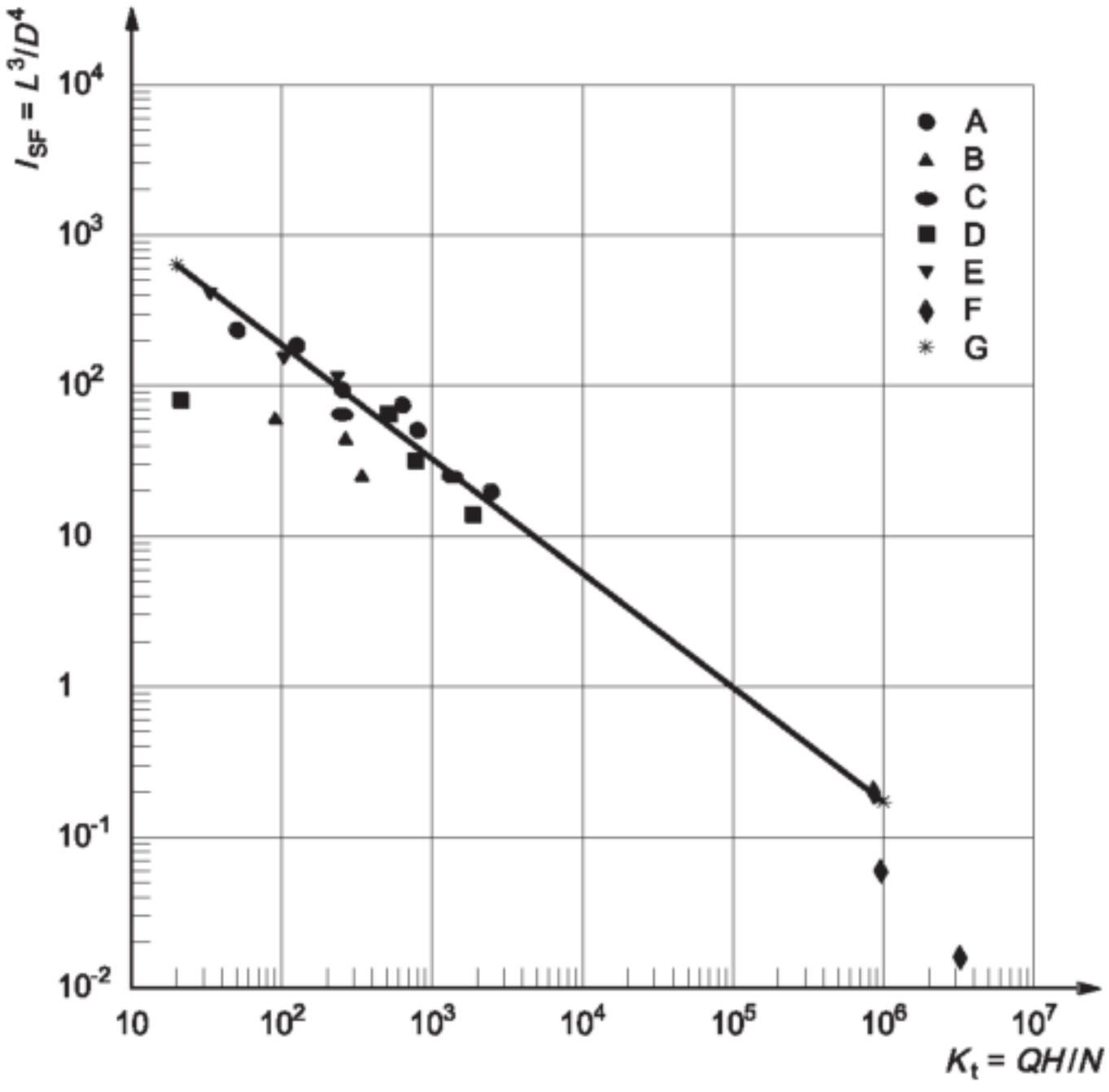


Figure K.3—Overhung Pump Shaft Flexibility Index vs Size Factor (USC Units)

Historically, this standard has required that "individual" bearings be designed for a minimum L_{10h} life of 25,000 h and 16,000 h at the maximum radial and axial loads and rated speed. Bearing-system life, $L_{10h,sys,}$ is calculated using Equation (K.6) and, therefore, is shorter than the shortest life of the individual bearings in the system:

$$L_{10h,sys} = \left[\left(1/L_{10hA}\right)^{3/2} + \left(1/_{L10hB}\right)^{3/2} + \ldots + \left(1/L_{10hN}\right)^{3/2}\right]^{-2/3} \quad \text{(K.6)}$$

where

L_{10hA} is the basic rating life, L_{10h}, in accordance with ISO 281 for bearing A;

L_{10hB} is the basic rating life, L_{10h}, in accordance with ISO 281 for bearing B;

L_{10hN} is the basic rating life, L_{10h}, in accordance with ISO 281 for bearing *N*.

EXAMPLE 1—If a pump has two bearings of equal L_{10h} life (not very likely) and a bearing system life of 25,000 h, the individual bearings would have a L_{10h} life of approximately 37,500 h. Note that because rolling-element bearings are manufactured in standard sizes, it is unlikely that any particular pump would have a calculated L_{10h} life of exactly 37,500 h for both bearings in the system.

EXAMPLE 2—If one bearing has a calculated L_{10h} life of 100,000 h (not unusual), it is necessary for the other bearing in the system to have a calculated L_{10h} life of only about 25,700 h to give a bearing system life of approximately 25,000 h.

OH2, OH3, and to a lesser extent, BB1, BB2, and BB3, pumps are not totally "engineered pumps" in that each component is not necessarily "designed" for a particular purchaser's order requirements. They are generally

"design lines of pumps," or pumps that are "pre-engineered" by the manufacturer to meet the requirements of this standard within a particular range of operating conditions for specific applications. This is particularly true for BB designs that can have multiple rotor and volute designs within one case pattern using several bearing housing and seal chamber designs. These "pre-engineered" pumps may then be modified, as required, to meet unique applications and purchaser requirements. In the design/product line development process for these pumps, the manufacturer chooses a set of extreme conditions within which he/she desires to sell "standard pumps." These conditions vary according to the manufacturer's experience with the breadth of services his/her customers require. These conditions can be chosen to encompass 98 % of all refinery services into which he/she has sold pumps in, say, a 10-year period.

Once the range of operating conditions has been established, the manufacturer chooses a number of bearing-frame sizes. For most pump manufacturers, this is either three or four sizes for these types of pumps. The manufacturer then matches his/her available or anticipated sets of hydraulics to the various bearing frames. For each bearing frame, there is a set of hydraulics that imposes the greatest loads on the bearings and frame. For this set of hydraulics, it is necessary that the minimum bearing system life meet the requirements of this standard. However, it is completely possible that there are sets of operating conditions or service parameters (such as high suction pressure, low running speed, operating pumps outside the preferred or allowable operating regions) where this pump does not meet all the requirements of this standard. For these conditions, the manufacturer has a number of options, including modifying the pump design to meet all the requirements, limiting the pump operating range, or negotiation with the purchaser to accept a slightly lower bearing-system life in order to reduce cost or improve overall bearing performance. This can be beneficial if operating conditions are such that bearing loads vary significantly and can result in loads that are too low for optimal bearing performance.

For all other sets of hydraulics, the bearing loads are lower. The equation for bearing life, $L_{10h,}$ as a function of load is given by Equation (K.7), taken from ISO 281:2007, Equation (4):

$$L_{10} = \left(C_r / P_r\right)^x \tag{K.7}$$

where

C_r is the dynamic load rating of the bearing;

P_r is the equivalent dynamic load;

x is 3 for ball bearings and 10/3 for roller bearings.

One can find methods of determining the bearing loads for a pump in the Hydraulic Institute standards. Examining Equation (K.6), one can see that for a given bearing, with a given load, a reduction in applied load of 10 % results in an increase in bearing life of approximately 37 %. This means that the calculated bearing-system life for all pumps with a given bearing frame is much greater than the bearing-system life requirement of this standard that is applied to the largest set of hydraulics for that bearing frame. Further, for the largest bearing frame, the system life is also greater if the impeller is at less-than-maximum diameter, if the specific gravity is low or if the suction pressure is lower than that used in the limiting case. Further, because bearings with the smallest size having ratings that exceed the calculated loads are chosen, even the largest set of hydraulics can well have a much higher calculated bearing-system life. This gives some insight as to why the historical requirement of 25,000 h L_{10h} life for individual bearings has not been a problem.

It is noted that there are rolling-element bearings much larger and with much higher dynamic load ratings that are used in pumps compliant with this standard. The application of rolling-element bearings is limited by bearing size and the speed of rotation. It has been found by the pump industry that large bearings running at two-pole speeds (3000 and 3600 nominal r/min) tend to run "hot" and can exceed the maximum bearing temperature requirements of this standard. As a result of this, all manufacturers limit angular-contact thrust bearings to about 7315 or 7316 sizes for two-pole machines. These correspond to shaft sizes of 2.95 in. and 3.1 in. (75 mm and 80 mm), respectively.

While hot running is detrimental to lubricant and bearing life, the most problematic issue is ball skidding in lightly loaded bearings. If a pump manufacturer applies larger bearings for each frame size, it is necessary that he/she limit the application of each bearing frame to hydraulics that provide bearing loads to minimize ball skidding.

This means additional bearing-frame sizes can be required to cover all pump sizes in a given product line, thus reducing component production volume, interchangeability, and stocking opportunities. The issue of bearing-system life is an application limitation for pumps of “standard designs.

Annex L
(informative)

Contract Documents and Engineering Design Data

L.1 When specified by the purchaser in 10.2, the contract documents and engineering design data shall be supplied by the vendor, as listed in this annex.

L.1.1 The following data shall be identified on transmittal (cover) letters, title pages and correspondence:

a) purchaser's/owner's corporate name;

b) job/project number;

c) equipment item number and service name;

d) inquiry or purchaser order number;

e) any other identification specified in the inquiry or purchaser order;

f) vendor's identifying proposal number, shop order number, serial number, or other reference required to identify return correspondence.

L.1.2 Each drawing shall have a title block in the lower right-hand corner with the date of certification, identification data specified in L.1.1, revision number and date, and title. Similar information shall be provided on all other documents including subvendor items.

L.2 Proposals

L.2.1 General

L.2.1.1 The vendor shall forward the original proposal, with the specified number of copies, to the addressee specified in the inquiry documents.

L.2.1.2 The proposal shall include, as a minimum, the data specified in L.2.2 through L.2.4.

L.2.1.3 The proposal shall include a specific statement that the system and all its components are in strict accordance with this standard.

L.2.1.4 If the equipment or any of its components or auxiliaries are not in strict accordance, the vendor shall include a list that details and explains each deviation.

L.2.1.5 The vendor shall provide details to enable the purchaser to technically evaluate any proposed alternative designs.

L.2.1.6 All correspondence shall be clearly identified in accordance with L.1.2.

L.2.1.7 Clearances less than those required by 6.7.7 and Table 6, shall be stated as exceptions to this standard in the proposal.

L.2.2 Drawings

L.2.2.1 The drawings indicated on the vendor drawing and data requirements (VDDR) form in this annex shall be included in the proposal. As a minimum, the following data shall be furnished.

a) A general arrangement or outline drawing for each major skid or system, showing direction of rotation, size and location of major purchaser connections, overall dimensions, maintenance clearance dimensions, overall weights, erection weights, maximum maintenance weights (indicated for each piece), lifting points, and methods of lifting the assembled machine;

b) Cross-sectional drawings showing the details of the proposed equipment;

c) Schematics of all auxiliary systems, including control, and electrical systems;

d) Bills of materials;

e) Sketches that show methods of lifting the assembled pump, packages, and major components and auxiliaries. [This information may be included on the drawings specified in Item a) above.]

f) The skid weight and dimensions to the skid's center of gravity (CoG) with the pump and all auxiliary equipment installed.

L.2.2.2 If "typical" drawings, schematics, and bills of material are used, they shall be marked up to show the correct weight and dimension data and to reflect the actual equipment and scope proposed.

L.2.3 Technical Data for Proposal

L.2.3.1 All technical data shall be given in units of measurement according to the purchase order. If needed, the technical data in alternate units can be included in parentheses.

L.2.3.2 The following data shall be included in the proposal.

a) Purchaser's data sheets, with complete vendor's information entered thereon and literature to fully describe details of the offering.

b) Predicted noise data.

c) VDDR form (see Figure L.1 for example form), indicating the schedule according to which the vendor agrees to transmit all the data specified.

d) Schedule for shipment of the equipment, in weeks after receipt of the order.

e) List of major wearing components, showing interchangeability with other items on the project or the owner's existing machines.

f) List of spare parts recommended for start-up and normal maintenance purposes (see Figure L.1).

g) List of the special tools furnished for maintenance (see 7.7.1).

h) Description of any special weather and winterization required for start-up, operation, and periods of idleness, under the site conditions specified on the data sheets. This description shall clearly indicate the protection to be furnished by the purchaser as well as that included in the vendor's scope of supply.

i) Complete tabulation of utility requirements, e.g. steam, water, electricity, and external flush liquid (including the quantity and supply pressure of the liquid required). Approximate data shall be clearly indicated as such.

j) Description of any optional or additional tests and inspection procedures for materials, as required by 6.12.1.7.

k) Description of any special requirements whether specified in the purchaser's inquiry or as outlined in 6.1.11, 6.1.14, 6.1.15, 6.1.24, 6.1.27, 6.1.35, 6.1.36, 6.2.1, 6.6.1, 6.10.1.1, 6.10.1.3, 9.1.3.1, and 9.1.3.6.

l) List of machines similar to the proposed machine(s) that have been installed and operating under conditions analogous to those in the inquiry.

m) Any start-up, shutdown, or operating restrictions required to protect the integrity of the equipment.

n) A list of any components that can be construed as being of alternative design, hence requiring purchaser's acceptance (6.1.1.3).

o) Component designed for a finite life (6.1.2).

p) Calculated specific speed and suction-specific speed.

q) Notification if testing at rated speed, per 8.3.3.5, is not possible or recommended. A recommendation for correction of pump power from rated to test conditions shall also be included.

L.2.4 Curves

The vendor shall provide complete performance curves, including differential head, typical efficiency, water NPSH3, and power expressed as functions of flowrate. The curves shall include the following:

a) except for low specific-speed designs where it would not be feasible, the curves shall be extended to at least 120 % of flowrate at peak efficiency, and the rated operating point shall be indicated;

b) the head curve for maximum and minimum impeller diameters shall be included;

c) the impeller identification number, specific speed, and suction-specific speed shall be shown on the curves;

d) if applicable, the curves shall indicate viscosity corrections;

e) minimum flow (both thermal and stable), preferred and allowable operating regions, and any limitations of operation shall be indicated.

f) the efficiency and power curves shall include the effect of alternate running clearances (see 6.7.7), e.g. for hot clearances and non-metallic wear parts.

g) curves for hydraulic turbines shall include plots of head, power, and efficiency versus capacity. Supplier shall also supply plots of power, efficiency, and capacity versus speed from zero-power speed to runaway speed.

h) if a pump is driven with an ASD, the head curve for maximum and minimum pump speed at the rated impeller diameter shall be included

L.2.5 Optional Tests

The vendor shall furnish an outline of the procedures to be used for each of the special or optional tests that have been specified by the purchaser or proposed by the vendor.

L.3 Engineering Design Data

L.3.1 General

L.3.1.1 Engineering data shall be furnished by the vendor in accordance with the agreed VDDR form (see Figure L.1 for example form).

L.3.1.2 The purchaser shall review the vendor's data upon receipt; however, this review shall not constitute permission to deviate from any requirements in the order unless specifically agreed in writing. After the data has been reviewed and accepted, the vendor shall furnish certified copies in the quantities specified.

L.3.1.3 A complete list of vendor data shall be included with the first issue of major drawings. This list shall contain titles, drawing numbers, and a schedule for transmittal of each item listed. This list shall cross-reference data with respect to the VDDR form.

L.3.2 Drawings and Technical Data

The drawings and data furnished by the vendor shall contain enough information so that together with the manuals specified in L.3.5, the purchaser can properly install, operate, and maintain the equipment covered by the purchase order. All contract drawings and data shall be clearly legible (8-point minimum font size even if reduced from a larger size drawing), shall cover the scope of the agreed VDDR form, and shall satisfy the applicable detailed descriptions in this annex.

L.3.3 Progress Reports

The vendor shall submit progress reports to the purchaser at intervals specified that shall, as a minimum, include the following:

a) overall progress summary,

b) status of engineering,

c) status of document submittals,

d) status of major suborders,

e) updated production schedule,

f) inspection/testing highlights for the month,

g) any pending issues.

L.3.4 Parts Lists and Recommended Spares

L.3.4.1 The vendor shall submit complete parts lists for all equipment and accessories supplied.

L.3.4.2 These lists shall include part names, manufacturers' unique part numbers and materials of construction (identified by applicable international standards).

L.3.4.3 Each part shall be completely identified and shown on appropriate cross-sectional, assembly-type cutaway, or exploded-view isometric drawings.

L.3.4.4 Interchangeable parts shall be identified as such.

L.3.4.5 Parts that have been modified from standard dimensions or finish to satisfy specific performance requirements shall be uniquely identified by part number.

L.3.4.6 The vendor shall indicate on each of these complete parts lists all those parts that are recommended as start-up or maintenance spares, and the recommended stocking quantities of each. These shall include spare parts recommendations of subvendors that were not available for inclusion in the vendor's original proposal. (Sample list is shown in Figure L.1 and Figure L.2 below.)

L.3.5 Installation, Operation, Maintenance, and Technical Data Manuals

L.3.5.1 General

The vendor shall provide proper written instructions and all necessary drawings to enable the purchaser to install, operate, and maintain all of the equipment covered by the purchase order. This information shall be compiled in a manual or manuals with a cover sheet showing the information listed in L.1.2, an index sheet, and a complete list of the enclosed drawings by title and drawing number. The manual pages and drawings shall be numbered. The manual or manuals shall be prepared specifically for the equipment covered by the purchase order. "Typical" manuals are unacceptable.

L.3.5.1.1 A draft manual(s) shall be issued to purchaser 8 weeks prior to mechanical testing for review and comment.

L.3.5.1.2 Refer to the VDDR form for number of copies. Hardcopies as well as electronic copies shall be provided as described on VDDR.

L.3.5.2 Installation Manual

L.3.5.2.1 All information required for the proper installation of the equipment shall be compiled in a manual that shall be issued no later than the time of issue of final certified drawings. For this reason, it may be separate from the operating and maintenance instructions.

L.3.5.2.2 This manual shall contain information on alignment and grouting procedures, normal and maximum utility requirements, centers of mass, rigging provisions and procedures, and all other installation data.

L.3.5.2.3 All drawings and data specified in L.2.2 and L.2.3 that are pertinent to proper installation shall be included as part of this manual [reference L.3.6.1 cc) below for listing].

L.3.5.2.4 One extra manual, over and above the specified quantity, shall be included with the first equipment shipped.

L.3.5.2.5 All recommended receiving and storage procedures shall be included.

NOTE Refer to API 686 for data required for installation.

L.3.5.3 Operating and Maintenance Manual

A manual containing all required operating and maintenance instructions shall be supplied at shipment. In addition to covering operation at all specified process conditions, this manual shall also contain separate sections covering operation under any specified extreme environmental conditions.

L.3.5.4 Technical Data Manual

The vendor shall provide the purchaser with a technical data manual at shipment.

	Number of Pumps with Identical Parts *n*						
	1 to 3	4 to 6	≥7	1 to 3	4 to 6	7 to 9	≥ 10
	Recommended Number of Spare Parts						
	Start-up			Normal Maintenance			
Cartridge [b, e]	—	—	—	1	1	1	1
Bundle [b, f]	—	—	—	1	1	1	1
Rotor [c, g]	—	—	—	1	1	1	1
Case [a]	—	—	—	—	—	—	1
Head (case cover and stuffing box)	—	—	—	—	—	—	1
Bearing bracket [a]	—	—	—	—	—	—	1
Shaft (with key)	—	—	—	1	1	2	*n*/3
Impeller	—	—	—	1	1	2	*n*/3
Wear rings (set) [h]	1	1	1	1	1	2	*n*/3
Bearings, complete (rolling element, radial) [a, i]	1	1	2	1	2	*n*/3	*n*/3
Bearings, complete (rolling element, thrust) [a, i]	1	1	2	1	2	*n*/3	*n*/3
Bearings, complete (hydrodynamic, radial) [a, i]	1	1	2	1	2	*n*/3	*n*/3
Bearing pads only (hydrodynamic, radial) [a, i]	1	1	2	1	2	*n*/3	*n*/3
Bearings, complete (hydrodynamic, thrust) [a, i]	1	1	2	1	2	*n*/3	*n*/3
Bearing pads only (hydrodynamic, thrust) [a, i]	1	1	2	1	2	*n*/3	*n*/3
Mechanical seal/packing [d, h, i]	1	2	*n*/3	1	2	*n*/3	*n*/3
Shaft sleeve [h]	1	2	*n*/3	1	2	*n*/3	*n*/3
Gaskets, shims, O-rings (set) [h]	1	2	*n*/3	1	2	*n*/3	*n*/3
Add for vertical pump:							
Bowls	—	—	—	—	—	*n*/3	n/3
Spiders or spider bushings (set)	—	—	1	1	1	*n*/3	*n*/3
Bearings, bushings (set)	1	1	2	1	1	*n*/3	*n*/3
Add for high-speed integral gear:							
Gear box	—	1	1	1	1	1	*n*/3
Diffuser and cover	1	1	1	1	1	1	*n*/3
Splined shaft	1	1	1	1	1	1	*n*/3
Gear-box housing	—	—	—	1	1	1	*n*/3
Oil pump, internal	—	1	1	1	1	1	*n*/3
Oil pump, external	—	1	1	1	1	1	*n*/3
Oil filter	1	2	*n*/3	1	2	3	*n*/3

a Horizontal pumps only.

b Vital-service pumps are generally unspared, partially spared, or multistage. When a vital machine is down, production loss or violation of environmental permits results.

c Essential-service pumps are required for operation and have an installed spare. A production loss occurs only if main and spare fail simultaneously.

d Cartridge-type mechanical seals include sleeve and gland.

e Cartridge bundle assembly consists of assembled components plus discharge head, seal(s), and bearing housing(s).

f The bundle consists of assembled rotor plus stationary hydraulic parts [diffuser(s) or volute(s)].

g Rotor consists of all rotating parts attached to the shaft, except the half-coupling.

h Normal-wear parts (see 3.1.31).

i Per pump set.

Figure L.1—Sample List

	TYPICAL VENDOR DRAWING AND DATA REQUIREMENTS					JOB NO. _____ PURCHASE ORDER NO. _____ REQUISITION NO. _____ ENQUIRY NO. _____				ITEM NO. _____ DATE _____ DATE _____ DATE _____			
						PAGE 1 OF 2 BY _____							
	FOR _____ SITE _____ SERVICE _____					REVISION _____ UNIT _____ NO. REQUIRED _____							
	Proposal [a]		Bidder shall furnish _____ copies of data for all items indicated by an X.										
		Review [b]	Vendor shall furnish _____ copies and _____ transparencies of drawings and data indicated.										
			Final [c]	Vendor shall furnish _____ copies and _____ transparencies of drawings and data indicated.									
				Vendor shall furnish _____ operating and maintenance manuals.									
					Distribution record	Final—Received from vendor							
						Due from vendor [c]							
						Review—Returned from vendor							
						Review—Received from vendor							
						Review—Due from vendor [c]							
					DESCRIPTION								
				Pump									
				a	Certified dimensional outline drawing								
				b	Cross-sectional drawings and bills of materials								
				c	Shaft seal drawing and bills of materials								
				d	Shaft coupling assembly drawing and bill of materials								
				e	Primary and auxiliary flush piping schematics and bills of materials								
				f	Cooling or heating schematic and bill of materials								
				g	Lubricating oil schematic and bill of materials								
				h	Lubricating oil system arrangement drawing								
				i	Lubricating oil component drawings								
				j	Electrical and instrumentation schematics, wiring diagrams, and bills of materials								
				k	Electrical and instrumentation arrangement drawing and list of connections								
				l	Performance curves								
				m	Vibration analysis data								
				n	Damped unbalanced response analysis								
				o	Lateral critical speed analysis								
				p	Torsional critical speed analysis								
				q	Certified hydrostatic test data								
				r	Material certifications								
				s	Progress reports								
				t	Weld procedures								
				u	Performance test data								
				v	Optional test data and reports								
				w	Certified rotor balance data for multistage pumps								
				x	Residual unbalance check								
				y	Rotor mechanical and electrical runout for pumps with noncontacting shaft vibration probes								
				z	Data sheets applicable to proposals, purchase and as-built								
				aa	Noise data sheets								
				bb	As-built clearances								
				cc	Installation, operation, and maintenance manuals								
				dd	Spare parts recommendations and price list								
				ee	Preservation, packaging, and shipping procedures								
				ff	Material safety data sheets								

Figure L.2a—Example Distribution Record

TYPICAL
VENDOR DRAWING AND DATA REQUIREMENTS

JOB NO. _____ ITEM NO. _____
PURCHASE ORDER NO. _____ DATE _____
REQUISITION NO. _____ DATE _____
ENQUIRY NO. _____ DATE _____

PAGE __2__ OF __2__ BY _____

FOR _____ REVISION _____
SITE _____ UNIT _____
SERVICE _____ NO. REQUIRED _____

Proposal [a] Bidder shall furnish _____ copies of data for all items indicated by an X.

Review [b] Vendor shall furnish _____ copies and _____ transparencies of drawings and data indicated.

Final [c] Vendor shall furnish _____ copies and _____ transparencies of drawings and data indicated.

Vendor shall furnish _____ operating and maintenance manuals.

DISTRIBUTION RECORD

Final – Received from vendor
Due from vendor [c]
Review – Returned from vendor
Review – Received from vendor
Review – Due from vendor [c]

			DESCRIPTION						
			Motor						
			a	Certified dimensional outline drawing					
			b	Cross-sectional drawing and bill of materials					
			c	Data sheets applicable to proposals, purchase and as-built					
			d	Noise data sheets					
			e	Performance data					
			f	Certified drawings of auxiliary systems					
			g	Installation operation and maintenance manuals					
			h	Spare parts recommendations and price list					
			i	Material safety data sheets					

[a] It is not necessary that proposal drawings and data be certified or as-built. Typical data shall be clearly identified as such.
[b] For single-stage units: these items are normally provided only in instruction manuals.
[c] These items are normally applicable to multistage units only.

Send all drawings and data to _____

_____

All drawings and data shall show project, appropriation, purchase order, and item numbers in addition to the plant location and unit. In addition to the copies specified above, one set of the drawings/instructions necessary for field installation shall be forwarded with the shipment.

Nomenclature:

_____ S—number of weeks prior to shipment.
_____ F—number of weeks after firm order.
_____ D—number of weeks after receipt of approved drawings.

Vendor _____
Date _____ Vendor reference _____
Signature _____

Figure L.2b—Example Distribution Record (Continued)

L.3.6 Description of Documents in Figure L.1

L.3.6.1 Pump

a) Certified dimensional outline drawing, including the following:

   1) Size, rating, and location of all purchaser connections;

   2) Approximate overall and handling masses;

   3) Overall dimensions, and maintenance and dismantling clearances;

   4) Shaft centerline height;

   5) Dimensions of baseplates (if furnished) complete with diameters, number and locations of bolt holes, and the thicknesses of sections through which it is necessary that the bolts pass;

   6) Grouting details;

   7) Forces and moments for suction and discharge nozzles;

   8) Center of gravity and lifting points;

   9) Shaft end separation and alignment data;

   10) Direction of rotation;

   11) Winterization, tropicalization, or noise attenuation details, if required.

   12) Piping/Valve arrangement for all auxiliary connections.

   13) For pump skids that will be mounted to steel modules, separate installation drawings for mounting the baseplates and anti-vibration mount (AVM) plates (including the adjustment procedure)

b) Cross-sectional drawings and bills of materials.

c) Shaft seal drawing and bill of materials.

d) Shaft coupling assembly drawing and bill of materials, including allowable misalignment tolerances and the style of the coupling guard.

e) Primary and auxiliary sealing schematic and bill of materials, including seal fluid, liquid flows, pressure, pipe and valve sizes, instrumentation, and orifice sizes.

f) Cooling or heating schematic and bill of materials, including cooling or heating media, liquid flows, pressure, pipe and valve sizes, instrumentation, and orifice sizes.

g) Lubricating oil schematic and bill of materials, including the following:

   1) oil flowrates, temperatures, and pressures at each use point;

   2) control, alarm, and trip settings (pressure and recommended temperatures);

   3) total head loads;

   4) utility requirements, including electricity, water, and air;

5) pipe, valve, and orifice sizes;

6) instrumentation, safety devices, control schemes, and wiring diagrams.

h) Lubricating oil system arrangement drawing, including size, rating, and location of all purchaser connections.

i) Lubricating oil component drawings and data, including the following:

1) pumps and drivers;

2) coolers, filters, and reservoir;

3) instrumentation;

4) spare parts lists and recommendations.

j) Electrical and instrumentation schematics, wiring diagrams, and bills of materials, including the following:

1) vibration alarm and shutdown limits,

2) bearing temperature alarm and shutdown limits,

3) lubricating oil temperature alarm and shutdown limits,

4) driver.

k) Electrical and instrumentation arrangement drawing and list of connections.

l) Performance curves.

m) Vibration analysis data.

n) Damped unbalanced response analysis.

o) Lateral critical speed analysis: the required number of lateral critical analysis reports, no later than 3 months after the date of order. The reports shall be as required in I.1.2 and I.1.3.

p) Torsional critical speed analysis: the required number of torsional critical analysis reports, no later than 3 months after the date of order. The reports shall be as required in 6.9.2.10.

q) Certified hydrostatic test data.

r) Material certifications: the vendor's physical and chemical data from mill reports (or certification) of pressure parts, impellers, and shafts.

s) Progress reports detailing the cause of any delays including engineering, purchasing, manufacturing, and testing schedules for all major components (planned and actual dates, and the percentage completed, shall be indicated for each milestone in the schedule).

t) Weld procedures.

u) Performance test data: certified shop logs of the performance test, record of shop test data (which the vendor shall maintain for at least 20 years after the date of shipment); the vendor shall submit certified copies of the test data to the purchaser before shipment.

v) Optional tests data and reports: optional tests data and reports include NPSH required test, complete unit test, sound level test, auxiliary equipment test, bearing housing resonance test, and any other tests mutually agreed upon by the purchaser and vendor.

w) Certified rotor balance data for multistage pumps.

x) Residual unbalance check.

y) Rotor mechanical and electrical runout for pumps designed to use noncontacting shaft vibration probes.

z) Data sheets applicable to proposals, purchase, and as-built.

aa) Noise data sheets.

bb) As-built clearances.

cc) Instruction manuals describing installation, operation, and maintenance procedures; each manual shall include the following sections.

1) Section 1—Installation:

   i) storage;

   ii) foundation;

   iii) grouting;

   iv) setting equipment, rigging procedures, component masses, and lifting diagram;

   v) alignment;

   vi) piping recommendations;

   vii) composite outline drawing for pump/driver train, including anchor-bolt locations;

   viii) dismantling clearances.

2) Section 2—Operation:

   i) start-up, including tests and checks before start-up;

   ii) routine operational procedures;

   iii) lubricating oil recommendations.

3) Section 3—Disassembly and assembly:

   i) rotor in pump casing;

   ii) journal bearings;

   iii) thrust bearings (including clearance and preload on rolling-element bearings);

   iv) seals;

   v) thrust collars, if applicable;

vi) allowable wear of running clearances;

vii) fits and clearances for rebuilding;

viii) routine maintenance procedures and intervals.

4) Section 4—Performance curves, including differential head, efficiency, water NPSH3, and brake horsepower vs flowrate for all operating conditions specified.

5) Section 5—Vibration data:

i) vibration analysis data,

ii) lateral critical speed analysis,

iii) torsional critical speed analysis.

6) Section 6—As-built data:

i) as-built data sheets,

ii) as-built clearances,

iii) rotor balance data for multistage pumps,

iv) noise data sheets,

v) performance data.

7) Section 7—Drawing and data requirements:

i) certified dimensional outline drawing and list of connections;

ii) cross-sectional drawing and bill of materials;

iii) shaft seal drawing and bill of materials;

iv) lubricating oil arrangement drawing and list of connections;

v) lubricating oil component drawings and data, and bills of materials;

vi) electrical and instrumentation schematics, wiring diagrams, and bills of materials;

vii) electrical and instrumentation arrangement drawing and list of connections;

viii) coupling assembly drawing and bill of materials;

ix) primary and auxiliary seal schematic and bill of materials;

x) primary and auxiliary seal piping, instrumentation, arrangement, and list of connections;

xi) cooling and heating schematic and bill of materials;

xii) cooling or heating piping, instrumentation arrangement, and list of connections;

xiii) a generic casing drawing defining casing retirement thickness(es) in critical areas shall be provided.

dd) Spare parts recommendations and price list.

ee) Preservation, packaging, and shipping procedure.

ff) Material safety data sheets.

L.3.6.2 Motor

a) Certified dimensional outline drawing for motor and all auxiliary equipment, including the following:

   1) size, location, and purpose of all purchaser connections, including conduit, instrumentation, and any piping or ducting;

   2) ASME rating and facing for any flanged connections;

   3) size and location of anchor bolt holes and thicknesses of sections through which bolts must pass;

   4) total mass of each item of equipment (motor and auxiliary equipment) plus loading diagrams, heaviest mass, and name of the part;

   5) center of gravity and lifting points

   6) overall dimensions and all horizontal and vertical clearances necessary for dismantling, and the approximate location of lifting lugs;

   7) shaft centerline height;

   8) shaft end dimensions, plus tolerances for the coupling;

   9) direction of rotation.

b) Cross-sectional drawing and bill of materials, including the axial rotor float.

c) Data sheets applicable to proposals, purchase, and as-built.

d) Noise data sheets.

e) Performance data including the following:

   1) for induction motors 200 hp (150 kW) and smaller:

      i) efficiency and power factor at one-half, three-quarter, and full load,

      ii) speed-torque curves;

   2) for induction motors larger than 200 hp (150 kW) and larger, certified test reports for all test run and performance curves as follows:

      i) time-current heating curve;

      ii) speed-torque curves at 70 %, 80 %, 90 %, and 100 % of rated voltage;

      iii) efficiency and power factor curves from zero to rated service factor;

      iv) current vs load curves from zero to rated service factor;

v) current vs speed curves from 0 % to 100 % of rated speed.

f) Certified drawings of auxiliary systems, including wiring diagrams, for each auxiliary system supplied; the drawings shall clearly indicate the extent of the system being supplied by the vendor and the extent being supplied by others.

g) Motor instruction manuals describing installation, operating, and maintenance procedures. Each manual shall include the following sections.

1) Section 1—Installation:

i) storage;

ii) setting motor, rigging procedures, component masses, and lifting diagram;

iii) piping and conduit recommendations;

iv) composite outline drawing for motor, including locations of anchor-bolt holes;

v) dismantling clearances.

2) Section 2—Operation:

i) start-up, including check before start-up;

ii) normal shutdown;

iii) operating limits, including number of successive starts;

iv) lubricating oil recommendations.

3) Section 3—Disassembly and assembly instructions:

i) rotor in motor,

ii) journal bearings,

iii) seals,

iv) routine maintenance procedures and intervals.

4) Section 4—Performance data required by L.2.2.1 e).

5) Section 5—Data sheets:

i) as-built data sheets,

ii) noise data sheets.

6) Section 6—Drawing and data requirements:

i) certified dimensional outline drawing for motor and all auxiliary equipment, with list of connections;

ii) cross-sectional drawing and bill of materials.

h) Spare parts recommendations and price list.

i) Material safety data sheets.

Annex M
(informative)

Test Data Summary

Figure M.1a and Figure M.1b show an example of a test data summary form. Figure M.2 and Figure M.3 show examples of a test curve format in USC units and SI units, respectively. Figure M.4 shows the test points referenced in 8.3.3.4.2.

TEST DATA SUMMARY				
Customer			Curve no.	
Purchaser			Test date	
Purchase order no.				
Item no.			Certified by:	
Pump serial no.			(Vendor representative)	
Size and type			Witnessed by:	
No. of stages			(Purchaser representative)	
Overall Pump Performance (Table 16)				
	Rated	**Interpolate Value**	**Actual Deviation ± %**	**Acceptance Tolerance ± %**
Flow				
Head				
Power				
NPSH3				
Shutoff head				
Speed, r/min				
Pump Construction Data				
Stage 1			**Series Stages**	
Impeller diameter	in. (mm)		Impeller diameter	in. (mm)
Impeller pattern no.			Impeller pattern no.	
No. of vanes			No. of vanes	
Volute/diffuser pattern no.			Volute/diffuser pattern no.	
Blade tip clearance (6.1.21)	%		Blade tip clearance (6.1.27)	%

Figure M.1a—Test Data Summary Form

Mechanical Performance						
Maximum Vibration Levels Recorded Within Specified Flow Region (6.9.4)						
	Rated Flow		**Preferred Operating Region**		**Allowable Operating Region**	
	Tested	**Specified**	**Tested**	**Specified**	**Tested**	**Specified**
Housing velocity: Drive end: Overall/filtered						
Nondrive end: Overall/filtered						
Shaft displacement: Drive end: Overall/filtered						
Nondrive end: Overall/filtered						

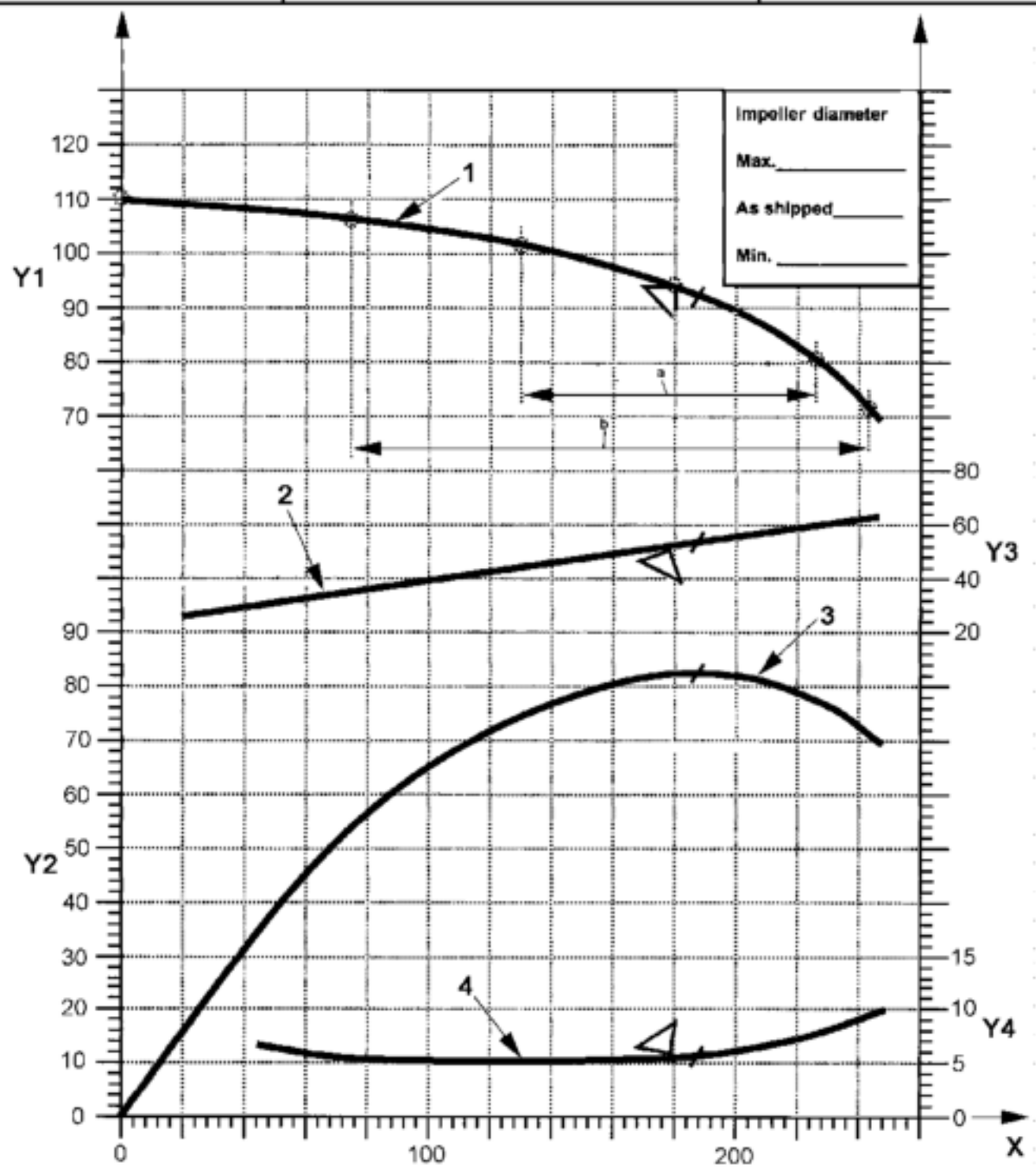
Bearing Temperatures °F (°C) [6.10.2.7, 9.2.5.2.5 c), and 9.2.5.3]			
Pressurized Lubrication Systems		**Ring Oil or Splash Lubrication**	
Ambient temp.		Ambient temp.	
Oil temp. rise		Oil temp. rise	
Oil return temp.		Oil sump temp.	
Max. bearing metal temp.			
Drive end journal			
Nondrive end journal			
Thrust bearing			

This mechanical performance summary is for recording test levels for each operating region relative to specified values. It is not intended to replace shop test data logs.

Units of measurement shall be in./s (mm/s) RMS for velocity, mils (mm) peak/peak for displacement, and °F (°C) for temperature.

Figure M.1b—Test Data Summary Form (Continued)

Pump serial no. _____	Pumped liquid _____	**Curve No.**
Size and type _____	Relative density _____	Rated point
No. of stages _____	Temperature _____ °C	Flowrate m^3/h = 180.0
Speed, r/min _____	Kinematic viscosity _____ mm^2/s	Head m = 94
Impeller no. _____	Impeller eye area _____ mm^2	NPSH3 m = 6.3
		Power kW = 55.9
		Calculated efficiency %: 82.3 Ref.



Date _____

Key

X	flowrate, expressed in cubic meters per hour	1	head
Y1	head, expressed in meters	2	power
Y2	efficiency, expressed in percentage	3	efficiency
Y3	power, expressed in kilowatts	4	NPSH3
Y4	NPSH3, expressed in meters		

[a] preferred operating region

[b] allowable operating region

NOTE Values for scales, flow, head, NPSH3, and power efficiency are for illustration only.

Figure M.3—Example of Test Curve Format (SI Units)

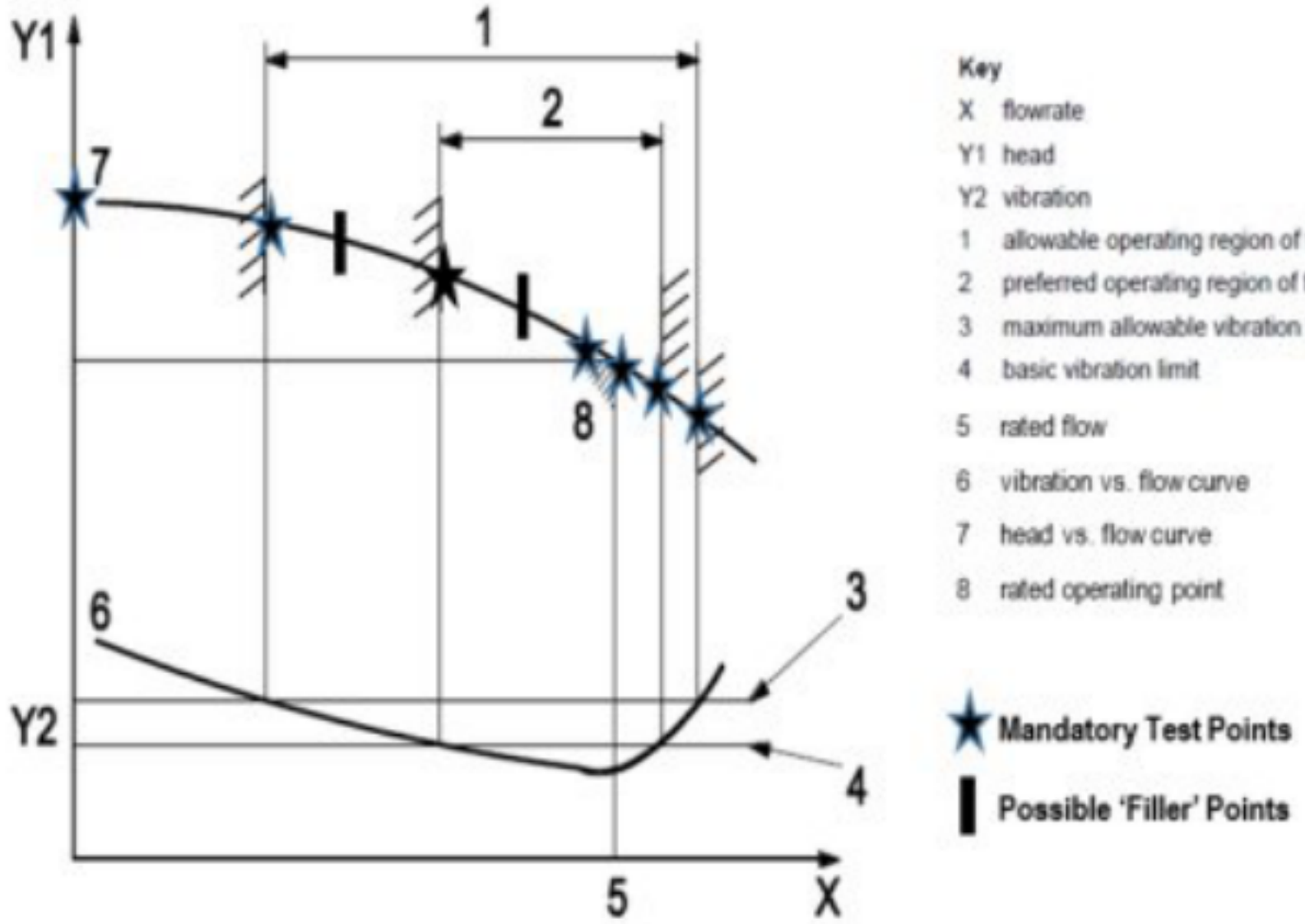


Figure M.4—Test Points Shown on Typical Performance Curve

Annex N
(informative)

Pump Data Sheets and Electronic Data Exchange

N.1 Pump Data Sheets

This annex contains typical data sheets for use by the purchaser and the vendor as shown in Figure N.1 and Figure N.2. The data sheets, in both USC and SI units, are available as Microsoft Excel Spreadsheets. Figure N.3 contains the API Project Design Data Sheet, which summarizes the Site Design Conditions and Site Utilities. These documents can be combined as needed to fully define the scope of the pump design requirements.

N.2 Electronic Data Exchange

The information contained in pump data sheets can also be transmitted digitally rather than via a conventional data sheet format. This is suitable if the pump purchaser and the pump vendor have systems that can process digital information rather than paper-based data sheets. Direct electronic transfer can be achieved with a transfer protocol that is adopted by both the purchaser and vendor. It is necessary that this transfer protocol also be commercially neutral for it to be accepted by all parties. Such a method improves the operating efficiencies of both parties if their internal data systems can import/export via this neutral protocol. Examples of these systems include

a) for the purchaser:

    1) pump specifications database,

    2) bid tab program,

    3) system analysis program,

    4) as-built asset management program;

b) for the vendor:

    1) pump selection system,

    2) pump configuration and quotation development system,

    3) order management system and bill-of-material management system.

Those interested in utilizing electronic data exchange (EDE) are encouraged to reference the EDE technology and implementation standard, HI 50.7. Additional interpretive information is also available at www.pumps.org/ede.

HI 50.7 was developed and supported by the Hydraulic Institute and the FIATECH automating equipment information exchange (AEX) project, in cooperation with the API. Information on the EDE technology and XML schemas is available online at www.pumps.org/ede.

A complete listing of all data fields in the electronic data sheets and their corresponding XML structures are found in HI 50.7 www.pumps.org/ede.

CLIENT: _____

PROJECT TITLE: _____

JOB NUMBER: _____

EQUIPMENT NUMBER: _____

EQUIPMENT SERVICE: _____

SERIAL NUMBER: _____

REQ / SPEC NUMBER : _____

PURCHASE ORDER NUMBER : _____

Cells coloured thus [ ] contain drop-down options
Text in BLACK color are by Purchaser
Text in BLUE color are by Supplier
Text in RED color are by Purchaser OR Supplier

COMMENTS: _____

	ADDITIONAL / RELATED DATASHEETS					
	ITEM / TAG NUMBERS	ATT	ITEM / TAG NUMBERS	ATT	ITEM / TAG NUMBERS	ATT
PUMP(s)						
MOTOR(s)						
GEAR(s)						
TURBINE(s)						

APPLICABLE OVERLAY STANDARDS _____

Rev	Date	Description	By	Checked				

	CENTRIFUGAL PUMP DATASHEET	DATASHEET No.
	API 610 - 13th EDITION	Sheet of

Figure N.1a—USC Units Pump Process Data Sheet

CENTRIFUGAL PUMP DATASHEET
API 610 - 13th EDITION

GENERAL

Note			Rev
APPLICABLE TO:		APPLICABLE NATIONAL / INTERNATIONAL STANDARD :	
CLIENT		UNIT	
SITE		SERVICE	
NO. REQ	PUMP SIZE	TYPE / No. STAGES	
MANUFACTURER		MODEL / SERIAL NO.	
EQUIPMENT NUMBER			

LIQUID CHARACTERISTICS

	Units	@ RATED TEMP	@ MAXIMUM TEMP	@ MINIMUM TEMP	*Alternate LIQUID*
LIQUID TYPE OR NAME :					
PUMPING TEMPERATURE :	°F				
VAPOR PRESSURE :	psia				
RELATIVE DENSITY (3.1.53) :					
SPECIFIC HEAT :	Btu/(lbm-°F)				
VISCOSITY :	cP				

OPERATING CONDITIONS (6.1.3)

	Units	Rated	Normal	Alt Condition 1 *[Name 1]*	Alt Condition 2 *[Name 2]*	Alt Condition 3 *[Name 3]*	Alt Condition 4 *[Name 4]*
NPSHA Datum:							
PUMPING TEMPERATURE :	°F						
FLOW :	gpm						
DISCHARGE PRESSURE (6.3.2) :	psia						
SUCTION PRESSURE :	psia						
DIFFERENTIAL PRESSURE :	psi						
DIFFERENTIAL HEAD :	ft						
NPSHA :	ft						
HYDRAULIC POWER :	HP						

SERVICE (CONTINUOUS/INTERMITTENT) :
• IF INTERMITTENT, NO. OF STARTS :
PUMPS OPERATE IN:
CORROSION DUE TO : (6.12.1.9)
EROSION DUE TO : (6.12.1.9)
H2S CONCENTRATION (ppm) : (6.12.1.13)
CHLORIDE CONCENTRATION (ppm) :
PARTICULATE SIZE (DIA IN MICRONS)
PARTICULATE CONCENTRATION (ppm)
CORROSION ALLOWANCE (6.3.10) :
DESIGN NOTES:

INSTALLATION LOCATION :
IF INDOOR, TEMPERATURE: MAX: °F MIN: °F
ELECTRIC AREA CLASSIFICATION (6.1.29):
DIVISION ZONE
GROUP
TEMP CLASS
PUMP VALVE START POSITION :

PERFORMANCE

PROPOSAL CURVE NO. RPM
IMPELLER DIA.: RATED MAX. MIN. in
RATED POWER : HP EFFICIENCY (%)
RATED CURVE BEP FLOW (at rated impeller diameter) (6.1.16) : gpm
MIN FLOW : THERMAL gpm STABLE gpm
PREFERRED OPERATING REGION (6.1.16) to gpm
ALLOWABLE OPERATING REGION (6.9.4.1) to gpm
MAX HEAD @ RATED IMPELLER ft
MAX POWER @ RATED IMPELLER HP
NPSH3 AT RATED FLOW (6.1.9) : ft
CENTERLINE OF PUMP TO NPSHA DATUM ft
NPSH MARGIN AT RATED FLOW (6.1.10) : ft
SPECIFIC SPEED (6.1.17) gpm,rpm,ft
SUCTION SPECIFIC SPEED LIMIT (6.1.11) gpm,rpm,ft
SUCTION SPECIFIC SPEED (6.1.11) gpm,rpm,ft
MAX. ALLOWABLE SOUND PRESSURE LEVEL (6.1.19) (dBA)
ESTIMATED MAX. SOUND PRESSURE LEVEL (dBA)
MAX. ALLOWABLE SOUND POWER LEVEL (6.1.19) (dB)
ESTIMATED MAX SOUND POWER LEVEL (dB)

DRIVER (7.1)

DRIVER TYPE
GEAR
VARIABLE SPEED REQUIRED
SOURCE OF VARIABLE SPEED
MANUFACTURER
NAMEPLATE POWER : HP
NOMINAL RPM
RATED LOAD RPM
FRAME
ORIENTATION
LUBRICATION
BEARING TYPE
RADIAL
THRUST
STARTING METHOD
DRIVER DATASHEET NUMBER
DRIVE DESIGN STANDARD

DATASHEET No. Rev: SHEET of

Figure N.1b—USC Units Pump Process Data Sheet (Continued)

CENTRIFUGAL PUMP DATASHEET
API 610 - 13th EDITION

CONSTRUCTION

API PUMP TYPE (Table 1):

NOZZLE CONNECTIONS (6.4.2):

	Size	Facing	Rating	Position
SUCTION				
DISCHARGE				

PRESSURE CASING AUX. CONNECTIONS: (6.4.3)

	No.	Size	Type	Facing	Rating	Posn.
BALANCE/LEAK OFF						
DRAIN						
VENT						
PRESSURE GAGE						
TEMP GAGE						
WARM-UP LINE						

DRAIN VALVE SUPPLIED BY (7.6.2.7)
DRAINS MANIFOLDED (7.6.2.7)
VENT VALVE SUPPLIED BY
VENTS MANIFOLDED
THREADED CONS FOR PIPELINE SERVICE & < 55ºC (6.4.3.5)
SPECIAL FITTINGS FOR TRANSITIONING (6.4.3.6)
CYLINDRICAL THREADS REQUIRED (6.4.3.5.1)
MACHINED AND STUDDED AUX CONNECTIONS (6.4.3.8)
ROUTE DRAIN TO SKID EDGE

CASING MOUNTING (6.3, 9.3.8.3):
CASING TYPE:
OH3 BACK-PULLOUT LIFTING DEVICE REQD. (9.1.2.6)
CASING PRESSURE RATING:
MAWP (6.2.8) : ____ kPa @ ____ ºC
HYDROTEST (8.3.2.3) : ____ kPa @ ____ ºC
SYSTEM DESIGN PRESSURE ____ kPa
WETTING AGENT REQUIRED FOR HYDROTEST (8.3.2.8) :
HYDROTEST OH PUMP AS ASSEMBLY :
SUCTN PRESS. REGIONS DESIGNED FOR MAWP (9.3.12.15) :
ROTOR:
SHAFT FLEXIBILITY INDEX (SFI) (9.1.1.3)
FIRST CRITICAL SPEED WET (MULTISTAGE PUMPS ONLY)
COMPONENT BALANCE TO ISO 21940-11, G1 (6.9.3.4) :
SHRINK-FIT-LIMITED MOVEMENT IMPELLERS (9.2.2.3)
ROTATION (VIEWED FROM COUPLING END) :
NUMBER OF IMPELLER VANES PER IMPELLER
NUMBER OF DIFFUSER VANES PER STAGE
NUMBER OF VOLUTES PER STAGE
IMPELLERS INDIVIDUALLY SECURED (9.2.2.1) :

COUPLING: (7.2)
MANUFACTURER
MODEL
RATING (kW / 100 RPM)
SPACER LENGTH (7.2.2.d) ____ mm
SERVICE FACTOR (7.2.4) 1.5
RIGID-TYPE
COUPLING WITH HYDRAULIC FIT (7.2.11)
COUPLING WITH PROPRIETARY CLAMPING DEVICE (7.2.12)
COUPLING IN COMPLIANCE WITH (7.2.5)(7.2.2.f)
COUPLING BALANCE TOLERANCE (7.2.2.g)

GUARDS (7.3)
COUPLING AND SHAFT GUARD STANDARD (7.3)
IGNITION HAZARD ASSESSMENT PER EN 13463-1 (7.3.2.2; 7.3.3.4)
COUPLING GUARD MATERIAL (7.3.2.1; 7.3.3.3) :
SHAFT GUARD MATERIAL (7.3.2.1; 7.3.3.3) :
SPARK RESISTANT MATERIAL REQUIRED (7.3.3.4) :

MATERIAL (6.12.1.1)

ANNEX H CLASS
MINIMUM DESIGN METAL TEMPERATURE (6.12.4.1) ____ ºC
MAXIMUM OPERATING TEMPERATURE (3.1.20) ____ ºC
REDUCED-HARDNESS MATERIALS REQ'D (6.12.1.15)
APPLICABLE HARDNESS STANDARD (6.12.1.15)
BARREL :
CASE :
DIFFUSERS
IMPELLER :
IMPELLER WEAR RING :
CASE WEAR RING :
SHAFT:
BOWL (IF VS-TYPE)

BASEPLATE

BASEPLATE REQUIRED
BASEPLATE DIMENSIONS (LxW) (mm) :
BASEPLATE CONSTRUCTION (7.4)
BASEPLATE DRAINAGE (7.4.1)
MOUNTING :
NON-GROUT CONSTRUCTION (7.4.3):
SUPPLIED WITH : • GROUT AND VENT HOLES
• DRAIN CONNECTION
PROVIDE SPACER PLATE UNDER ALL EQUIPMENT FEET (7.4.14)
BOLT OH 3/4/5 PUMP TO PAD / FOUNDATION :
PROVIDE SOLEPLATE FOR OH 3/4/5 PUMPS :
MECHANICALLY ADJUSTABLE CHOCKS BY VENDOR (7.4.15)

BEARINGS AND BEARING HOUSINGS (6.10.1)

BEARING (TYPE / NUMBER):
RADIAL / ____
THRUST / ____
REVIEW AND APPROVE THRUST BEARING SIZE : (9.2.5.2.7)
LUBRICATION (6.11) :
PRESSURE LUBE SYSTEM TO API-614, CHAPTER :
API 614 DATASHEETS ATTACHED (9.2.6.2)
PRESSURIZED LUBE OIL SYSTEM MOUNTED ON BASEPLATE:
LOCATION OF PRESSURIZED LUBE OIL SYSTEM:
INTERCONNECTING PIPING PROVIDED BY
OIL VISC. ISO GRADE VG
VENT-TO-HOUSING CONSTANT LEVEL OILER (6.10.2.4) :
OIL MIST PROVISIONS (6.11.4)
GREASE LUBRICATION (6.11.5)

DATASHEET No. _____ Rev: _____ SHEET _____ of _____

Figure N.1c—USC Units Pump Process Data Sheet (Continued)

CENTRIFUGAL PUMP DATASHEET
API 610 - 13th EDITION

INSTRUMENTATION

Note		
	INSTRUMENTATION PER API-670 (7.5.2)	
	ACCELEROMETERS (7.5.2.1)	
	NUMBER OF ACCELEROMETERS	
	MOUNTING LOCATION:	
	PROVISION FOR MTG ONLY (6.10.2.13)	
	FLAT SURFACE REQUIRED (6.10.2.14)	
	VIBRATION PROBES (7.5.2.2)	
	VIBRATION PROBES REQUIRED (7.5.2.2)	
	NUMBER PER RADIAL BEARING	
	NUMBER PER AXIAL BEARING	
	THREADED PROVISION FOR MTG ONLY (6.10.2.13; 6.6.12)	
	FLAT SURFACE PROVISION ONLY (6.10.2.14)	
	MONITORS AND CABLES SUPPLIED BY (7.5.2.4)	
	TEMPERATURE DETECTORS (7.5.2.3)	
	TEMP. PROBES REQUIRED (7.5.2.3)	
	PROVISIONS FOR MOUNTING ONLY (6.10.2.2)	
	RADIAL BEARING TEMP.	
	NUMBER PER RADIAL BEARING	
	THRUST BEARING TEMP.	
	NUMBER PER THRUST BEARING ACTIVE SIDE	
	NUMBER PER THRUST BEARING INACTIVE SIDE	
	TEMP. GAUGES (WITH THERMOWELLS) (9.1.3.5)	
	TEMP. GAUGE LOCATION	
	SUPPLY UPPER/LOWER CASING RTD'S FOR WARMUP	
	PRESSURE GAUGE TYPE	
	PRESSURE GAUGE LOCATION	

SEAL SUPPORT SYSTEM MOUNTING

MOUNTED ON PUMP BASEPLATE (7.6.1.4)	
LOCATION ON OR OFF BASEPLATE (7.4.6) :	
INTERCONNECTING PIPING BY	

MECHANICAL SEAL (6.8)

API 682 DATASHEET ATTACHED :	
ADDITIONAL CENTRAL FLUSH PORT (6.8.9)	
HEATING JACKET REQ'D. (6.8.11)	

HEATING AND COOLING (6.1.23-6.1.27)

COOLING REQ'D		
COOLING WATER PIPING PLAN (7.6.3.1)		
COOLING WATER PIPING FITTINGS		
COOLING WATER PIPING MATERIALS		
COOLING WATER REQUIREMENTS:	SUPPLY PRESSURE	psig
	BEARING HOUSING	gpm
	HEAT EXCHANGER	gpm
	TOTAL COOLING WATER	gpm
HEATING MEDIUM		
HEATING PIPING		

PIPING & APPURTENANCES

TAG ALL ORIFICES (7.6.2.4)	
SOCKET WELD UNION ON 1st SEAL GLAND NIPPLE (7.6.2.8)	
MANIFOLD AUX PIPING SYSTEMS AT SKID EDGE (7.6.1.6)	

PRESSURE VESSEL DESIGN CODE REFERENCES

THESE REFERENCES SHALL BE PROVIDED BY THE MANUFACTURER

CASTING FACTORS USED IN DESIGN (PER TABLE 4)	
SOURCE OF MATERIAL PROPERTIES (6.3.5)	

WELDING AND REPAIRS (6.12.3.1)

THESE REFERENCES SHALL BE PROVIDED BY THE PURCHASER. (DEFAULT TO TABLE 11 IF NO PURCHASER PREFERENCE IS STATED)

ALTERNATIVE WELDING CODES AND STANDARDS	
WELDING REQUIREMENT (APPLICABLE CODE OR STANDARD)	
ALTERNATIVE WELDER/OPERATOR QUALIFICATION STANDARD	
ALTERNATIVE WELDING PROCEDURE QUALIFICATION STANDARD	
NON-PRESSURE RETAINING STRUCTURAL WELDING STANDARD (BASEPLATES OR SUPPORTS)	
STANDARD FOR MAGNETIC PARTICLE OR LIQUID PENETRANT EXAMINATION (PLATE EDGES)	
STANDARD FOR POSTWELD HEAT TREATMENT	
STANDARD FOR POSTWELD HEAT TREATMENT OF CASING FABRICATION WELDS	

MATERIAL INSPECTION

THESE REFERENCES SHALL BE PROVIDED BY THE PURCHASER

DEFAULT TO TABLE 14 TABLE 14 INSPECTION CLASS:

ALTERNATIVE MATERIAL INSPECTIONS AND ACCEPTANCE CRITERIA (SEE TABLE 15, 8.2.2.5)

TYPE OF INSPECTION	METHOD	FOR FABRICATIONS	FOR CASTINGS
RADIOGRAPHY			
ULTRASONIC INSPECTION			
MAGNETIC PARTICLE INSPECTION			
LIQUID PENETRANT INSPECTION			
VISUAL INSPECTION (all surfaces)			

DATASHEET No. _____ Rev: _____ SHEET _____ OF _____

Figure N.1d—USC Units Pump Process Data Sheet (Continued)

CENTRIFUGAL PUMP DATASHEET
API 610 - 13th EDITION

Note	SURFACE PREPARATION, PAINT & SPARES	TEST	Rev
	MANUFACTURER'S STANDARD (8.4.3.4)	SHOP INSPECTION (8.1.1.1)	
	OTHER (SEE BELOW)	PERF. CURVE & DATA APPROVAL REQ'D PRIOR TO SHIPMENT (8.3.3.4.6) :	
	SPECIFICATION NO.	TEST WITH SUBSTITUTE SEAL (8.3.3.3.1)	
	PUMP:	MATERIAL CERTIFICATION REQUIRED (6.12.1.8): CASING	
	PUMP SURFACE PREPARATION	IMPELLER	
	PRIMER	SHAFT	
	FINISH COAT	OTHER	
	BASEPLATE:	CASTING REPAIR WELD PROCEDURE APPROVAL REQD (6.12.2.5):	
	SURFACE PREPARATION	INSPECTION REQUIRED FOR CONNECTION WELDS (6.12.3.4)	
	PRIMER:	MAG PARTICLE	
	FINISH COAT	RADIOGRAPHY	
	UNDERSIDE	LIQUID PENETRANT	
		ULTRASONIC	
	SHIPMENT: (8.4)	HARDNESS TEST REQUIRED (6.12.1.14; 8.2.2.7)	
	EXPORT BOXING REQUIRED	ADDITIONAL SUBSURFACE EXAMINATION (6.12.1.6) (8.2.1.3)	
	PREPARE FOR OUTDOOR STORAGE (NUMBER OF MONTHS) :	FOR:	
		METHOD:	
	SPARE ROTOR ASSEMBLY PACKAGED FOR:	PMI TESTING REQUIRED (8.2.2.8)	
	ROTOR STORAGE ORIENTATION (9.2.8.2)	COMPONENTS TO BE TESTED :	
	SHIPPING & STORAGE CONTAINER SUITABLE FOR	RESIDUAL UNBALANCE TEST (J.4.1.2)	
	VERTICAL STORAGE (9.2.8.3) :	NOTIFICATION OF SUCCESSFUL PERFORMANCE TEST (8.1.1.3) (8.3.3.4.6)	
	N2 PURGE REQUIRED (9.2.8.4) :	BASEPLATE (NOZZLE LOAD) TEST (7.4.24)	
		HYDROSTATIC TEST (8.3.2)	
	SPARE PARTS (10.3.4.2)	HYDROSTATIC TEST OF BOWLS & COLUMN (9.3.13.1)	
	START-UP	PERFORMANCE TEST (8.3.3)	
	NORMAL MAINTENANCE	RETEST ON SEAL LEAKAGE (8.3.3.3.2)	
		TEST DATA POINTS TO (8.3.3.4)	
		ALTERNATE TEST TOLERANCES PER (8.3.3.5)	
	MASSES lb	NPSH (8.3.4.3)	
	ITEM No / PUMP / DRIVER / GEAR / BASE / TOTAL	NPSH BASED ON 1ST STG ONLY ALLOWED (8.3.4.3.2)	
		TEST NPSHA LIMITED TO 110% SITE NPSHA (8.3.3.7)	
		RETEST REQUIRED AFTER FINAL HEAD ADJUSTMENT (8.3.3.8.2)	
		COMPLETE UNIT TEST (8.3.4.4.1)	
		SOUND LEVEL TEST (8.3.4.5)	
	OTHER PURCHASER REQUIREMENTS	CLEANLINESS PRIOR TO FINAL ASSEMBLY (8.2.2.6)	
	COORDINATION MEETING REQUIRED	LOCATION OF CLEANLINESS INSPECTION:	
	MAXIMUM DISCHARGE PRESSURE TO INCLUDE (6.3.2)	CHECK FOR CO-PLANAR MOUNTING PAD FLATNESS (7.4.8)	
	MAX RELATIVE DENSITY	MECHANICAL RUN TEST UNTIL OIL TEMP STABLE (8.3.4.2.1)	
	OPERATION TO TRIP SPEED	4 HR. MECH RUN AFTER OIL TEMP STABLE (8.3.4.2.2)	
	MAX DIA. IMPELLERS AND/OR NO OF STAGES	BRG HSG RESONANCE TEST (8.3.4.7)	
	CONNECTION DESIGN APPROVAL (9.2.1.4)	STRUCTURAL RESONANCE TEST (9.3.9.2)	
	TORSIONAL ANALYSIS / REPORT (6.9.2)	REMOVE / INSPECT HYDRODYNAMIC BEARINGS AFTER TEST (9.2.7.4)	
	PROGRESS REPORTS (L.3.3)	AUXILIARY EQUIPMENT TEST (8.3.4.6)	
	OUTLINE OF PROCEDURES USED FOR OPTIONAL TESTS (L.2.5)	EQUIPMENT TO BE INCLUDED IN AUXILLIARY TESTS :	
	ADDITIONNAL DATA REQUIRING 20 YEARS RETENTION (8.2.1.1)		
	LATERAL ANALYSIS REQUIRED (9.2.4.1.2)	LOCATION OF AUXILIARY EQUIPMENT TEST :	
	ROTOR DYNAMIC BALANCE TO 4W/N (6.9.3.5)		
	VFD STEADY STATE FORCED RESPONSE ANALYSIS (6.9.2.3)	IMPACT TEST (6.12.4.3) PER EN 13445	
	TRANSIENT FORCED RESPONSE (6.9.2.4)	PER ASME SECTION VIII	
	BEARING LIFE CALCULATIONS REQUIRED (6.10.1.11)	CASING DISASSEMBLY AFTER TEST (8.3.3.9)	
	CASING RETIREMENT THICKNESS DRAWING [L.3.6.1 cc) 7) xiii)]	SPARE PARTS TEST (8.3.4.8)	
	CONNECTION BOLTING (7.6.1.7)		
	VENDOR TO KEEP REPAIR AND HT RCDS (8.2.1.1.c)		
	VENDOR SUBMIT TEST PROCEDURES (8.3.1.1)		
	SUBMIT INSPECTION CHECK LIST (8.1.5)		
	ACOUSTIC ANALYSIS OF CROSSOVER PASSAGE (BB3, BB5)(9.2.1.5)		
	API-691 DOCUMENTATION REQUIRED (6.1.3.1)		

DATASHEET No. _____ Rev: _____ SHEET _____ OF _____

Figure N.1e—USC Units Pump Process Data Sheet (Continued)

VERTICAL PUMP SUPPLEMENTAL DATASHEET
API 610 - 13th EDITION

VERTICAL PUMP TYPE (Table 1) _____ Rev

REMARKS _____

VERTICAL PUMPS

	(+) UP	(-) DOWN
PUMP THRUST:		
STATIC THRUST	____ lbf	____ lbf
AT MIN FLOW	____ lbf	____ lbf
AT RATED FLOW	____ lbf	____ lbf
AT MAX FLOW	____ lbf	____ lbf
MAX THRUST	____ lbf	____ lbf
SEPARATE SOLEPLATE REQUIRED (9.3.8.3.2)		____
SOLEPLATE Length x Width	____ in	X ____ in
SOLEPLATE THICKNESS		____ in
SEPARATE MOUNTING FLANGE REQUIRED (9.3.8.3.1)		____

COLUMN PIPE:	____
DIAMETER	____ in
TOTAL COLUMN LENGTH	____ ft
NUMBER OF SECTIONS	____
SPACING (LENGTH PER SECTION)	____ ft
GUIDE BUSHINGS:	
NUMBER	____
LINE SHAFT BEARING SPACING	____ in
GUIDE BUSHING LUBE:	____

VERTICAL PUMPS (CONT'D)

LINE SHAFT:	____
LINE SHAFT DIAMETER	____ in
TUBE DIAMETER	____ in
LINESHAFT CONNECTION	____
LEVEL CONTROL	____
SUCTION STRAINER TYPE	____
IMPELLER COLLETS ACCEPTABLE (6.6.3)	____
HARDENED SLEEVES UNDER BEARINGS (9.3.10.4)	____
STRUCTURAL ANALYSIS (9.3.5)	____
SUCTION CAN	
THICKNESS	____ in
LENGTH	____ ft
DIAMETER	____ in
CAN DRAIN PIPED TO SURFACE (9.3.13.4)	____
ELLIPTICAL BOTTOM HEAD REQUIRED (9.3.2.5)	____
PRESSURE CASING WALLS USING SEAMLESS PIPE (9.3.2.7)	____
HYDROTEST BOWLS/COLUMN @ MAX DIFF PRESSURE (9.3.13.1)	____

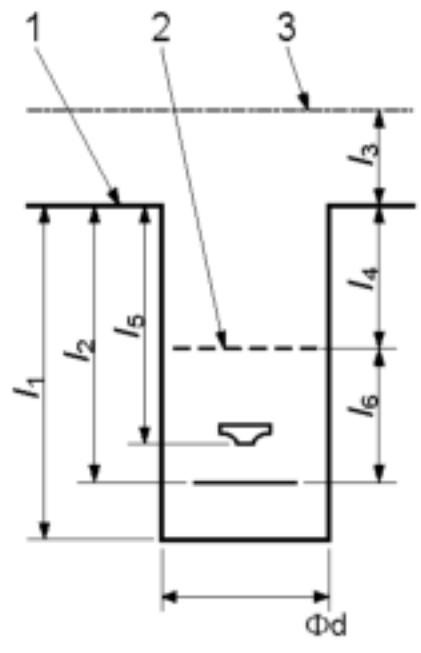
MATERIALS (additional)

SUCTION CAN / BARREL:	____	LINESHAFT SLEEVES :	____
DISCHARGE HEAD :	____	BEARING RETAINER :	____
BOWL SHAFT :	____	SHAFT ENCLOSING TUBE :	____
LINESHAFT :	____	DISCHARGE COLUMN :	____
LINESHAFT HARDFACING :	____	**PRESSURE RATING:**	**MAWP (psig)** / **HYDRO (psig)**
BELLMOUTH :	____	HEAD	____ / ____
BOWL BEARING :	____	COLUMN PIPE	____ / ____
LINESHAFT BEARING :	____	BOWL	____ / ____

SUMP ARRANGEMENT

SUMP DIMENSIONS :

GRADE ELEVATION	1 ____	ft
LOW LIQUID LEVEL ($l_3 + l_4$)	2 ____	ft
C.L. OF DISCHARGE	3 ____	ft
SUMP DEPTH	l_1 ____	ft
PUMP LENGTH	l_2 ____	ft
GRADE TO DISCH.	l_3 ____	ft
GRADE TO LOW LIQUID LVL	l_4 ____	ft
GRADE TO 1ST STG IMPL'R.	l_5 ____	ft
SUBMERGENCE REQ'D	l_6 ____	ft
SUMP DIAMETER	Φd ____	ft



DATASHEET No. _____ Rev: _____ SHEET _____ OF _____

Figure N.1f—USC Units Pump Process Data Sheet (Continued)

NOTES

	Note #		Rev
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DATASHEET No. _____ Rev: _____ SHEET _____OF_____

Figure N.1g—USC Units Pump Process Data Sheet (Continued)

CLIENT: _____

PROJECT TITLE: _____

JOB NUMBER: _____

EQUIPMENT NUMBER: _____

EQUIPMENT SERVICE: _____

SERIAL NUMBER: _____

REQ / SPEC NUMBER : _____

PURCHASE ORDER NUMBER : _____

Cells coloured thus [ ] contain drop-down options
Text in BLACK color are by Purchaser
Text in BLUE color are by Supplier
Text in RED color are by Purchaser OR Supplier

COMMENTS: _____

	ADDITIONAL / RELATED DATASHEETS					
	ITEM / TAG NUMBERS	ATT	ITEM / TAG NUMBERS	ATT	ITEM / TAG NUMBERS	ATT
PUMP(s)						
MOTOR(s)						
GEAR(s)						
TURBINE(s)						

APPLICABLE OVERLAY STANDARDS _____

Rev	Date	Description	By	Checked				

	CENTRIFUGAL PUMP DATASHEET **API 610 - 13th EDITION**	DATASHEET No.
		Sheet / of

Figure N.2a—SI Units Pump Process Data Sheet

CENTRIFUGAL PUMP DATASHEET
API 610 - 13th EDITION

GENERAL

1	Note	APPLICABLE TO:		APPLICABLE NATIONAL / INTERNATIONAL STANDARD:		Rev
2		CLIENT		UNIT		
3		SITE		SERVICE		
4		NO. REQ.	PUMP SIZE	TYPE	No. STAGES	
5		MANUFACTURER		MODEL	SERIAL NO.	
6		EQUIPMENT NUMBER				

LIQUID CHARACTERISTICS

	Units	@ RATED TEMP	@ MAXIMUM TEMP	@ MINIMUM TEMP	*Alternate LIQUID*
LIQUID TYPE OR NAME:					
PUMPING TEMPERATURE:	°C				
VAPOR PRESSURE:	kPa a				
RELATIVE DENSITY (3.1.46):					
SPECIFIC HEAT:	kJ/(kg-K)				
VISCOSITY:	Pa s				

OPERATING CONDITIONS (6.1.3)

	Units	Rated	Normal	Alt Condition 1 *(Name 1)*	Alt Condition 2 *(Name 2)*	Alt Condition 3 *(Name 3)*	Alt Condition 4 *(Name 4)*
NPSHA Datum:							
PUMPING TEMPERATURE:	°C						
FLOW:	m³/s						
DISCHARGE PRESSURE (6.2.1.2):	kPa a						
SUCTION PRESSURE:	kPa a						
DIFFERENTIAL PRESSURE:	kPa						
DIFFERENTIAL HEAD:	m						
NPSHA:	m						
HYDRAULIC (BRAKE) POWER:	kW						

SERVICE (CONTINUOUS/INTERMITTENT):
• IF INTERMITTENT, NO. OF STARTS:
PUMPS OPERATE IN:
CORROSION DUE TO: (6.12.1.10)
EROSION DUE TO: (6.12.1.10)
H2S CONCENTRATION (ppm): (6.12.1.14)
CHLORIDE CONCENTRATION (ppm):
PARTICULATE SIZE (DIA IN MICRONS)
PARTICULATE CONCENTRATION (ppm)
CORROSION ALLOWANCE (6.2.140):
ENTRAINED GAS (COMPOSITION / %-VOLUME)
EXTREMELY HAZARDOUS: HIGHLY CORROSIVE:
DESIGN NOTES:

INSTALLATION LOCATION:
IF INDOOR, TEMPERATURE: MAX: °C MIN: °C
ELECTRIC AREA CLASSIFICATION (6.1.36):
DIVISION ZONE
GROUP
TEMP CLASS
PUMP VALVE START POSITION:

PERFORMANCE

PROPOSAL CURVE NO. RPM
IMPELLER DIA.: RATED MAX. MIN. mm
RATED POWER: kW EFFICIENCY (%)
RATED CURVE BEP FLOW (at rated impeller diameter) (6.1.22): m³/s
MIN FLOW: THERMAL m³/s STABLE m³/s
PREFERRED OPERATING REGION (6.1.22) to m³/s
ALLOWABLE OPERATING REGION (6.9.4.1) to m³/s
MAX HEAD @ RATED IMPELLER m
MAX POWER @ RATED IMPELLER kW
NPSH3 AT RATED FLOW (6.1.12): m
CENTERLINE OF PUMP TO NPSHA DATUM m
NPSH MARGIN AT RATED FLOW (6.1.15): m
NPSH MARGIN REQUIRED m
SPECIFIC SPEED (6.1.16) m³/s, rpm, m
SUCTION SPECIFIC SPEED LIMIT (8.1.16) m³/s, rpm, m
SUCTION SPECIFIC SPEED (6.1.16) m³/s, rpm, m
MAX. ALLOWABLE SOUND PRESSURE LEVEL (6.1.25.2) (dBA)
ESTIMATED MAX. SOUND PRESSURE LEVEL (dBA)
MAX. ALLOWABLE SOUND POWER LEVEL (6.1.25.2) (dB)
ESTIMATED MAX SOUND POWER LEVEL (dB)

DRIVER (7.1)

DRIVER TYPE
GEAR
VARIABLE SPEED REQUIRED
SOURCE OF VARIABLE SPEED
MANUFACTURER
NAMEPLATE POWER: kW
NOMINAL RPM
RATED LOAD RPM
FRAME
ORIENTATION
LUBRICATION
BEARING TYPE
RADIAL
THRUST
STARTING METHOD
DRIVER DATASHEET NUMBER
DRIVE DESIGN STANDARD

DATASHEET No. Rev: SHEET of

Figure N.2b—SI Units Pump Process Data Sheet (Continued)

CENTRIFUGAL PUMP DATASHEET
API 610 - 13th EDITION

Note	CONSTRUCTION	Rev

API PUMP TYPE (Table 1):

NOZZLE CONNECTIONS (6.4.2):

	Size	Facing	Rating	Position
SUCTION				
DISCHARGE				

PRESSURE CASING AUX. CONNECTIONS: (6.4.3)

	No.	Size	Type	Facing	Rating	Posn.
BALANCE/LEAK OFF						
DRAIN						
VENT						
PRESSURE GAGE						
TEMP GAGE						
WARM-UP LINE						

DRAIN VALVE SUPPLIED BY (7.6.2.7)
DRAINS MANIFOLDED (7.6.2.7)
VENT VALVE SUPPLIED BY
VENTS MANIFOLDED
THREADED CONS FOR PIPELINE SERVICE & < 55°C (6.4.3.5)
SPECIAL FITTINGS FOR TRANSITIONING (6.4.3.6)
CYLINDRICAL THREADS REQUIRED (6.4.3.5.1)
MACHINED AND STUDDED AUX CONNECTIONS (6.4.3.8)
ROUTE DRAIN TO SKID EDGE

MATERIAL (6.12.1.1)

ANNEX H CLASS
MINIMUM DESIGN METAL TEMPERATURE (6.12.4.1) °C
MAXIMUM OPERATING TEMPERATURE (3.1.20) °C
REDUCED-HARDNESS MATERIALS REQ'D (6.12.1.15)
APPLICABLE HARDNESS STANDARD (6.12.1.15)
BARREL :
CASE :
DIFFUSERS
IMPELLER :
IMPELLER WEAR RING :
CASE WEAR RING :
SHAFT:
BOWL (IF VS-TYPE)

BEARINGS AND BEARING HOUSINGS (6.10.1)

BEARING (TYPE / NUMBER):
RADIAL /
THRUST /
REVIEW AND APPROVE THRUST BEARING SIZE : (9.2.5.2.7)
LUBRICATION (6.11) :
PRESSURE LUBE SYSTEM TO API-614, CHAPTER :
API 614 DATASHEETS ATTACHED (9.2.6.2)
PRESSURIZED LUBE OIL SYSTEM MOUNTED ON BASEPLATE:
LOCATION OF PRESSURIZED LUBE OIL SYSTEM:
INTERCONNECTING PIPING PROVIDED BY
OIL VISC. ISO GRADE VG
VENT-TO-HOUSING CONSTANT LEVEL OILER (6.10.2.4) :
OIL MIST PROVISIONS (6.11.4)
GREASE LUBRICATION (6.11.5)

CASING MOUNTING (6.3, 9.3.8.3):
CASING TYPE:
OH3 BACK-PULLOUT LIFTING DEVICE REQD (9.1.2.6)
CASING PRESSURE RATING:
MAWP (6.2.8) : kPa @ °C
HYDROTEST (8.3.2.3) : kPa @ °C
SYSTEM DESIGN PRESSURE kPa
WETTING AGENT REQUIRED FOR HYDROTEST (8.3.2.8) :
HYDROTEST OH PUMP AS ASSEMBLY :
SUCTN PRESS. REGIONS DESIGNED FOR MAWP (9.3.12.15) :
ROTOR:
SHAFT FLEXIBILITY INDEX (SFI) (9.1.1.3)
FIRST CRITICAL SPEED WET (MULTISTAGE PUMPS ONLY)
COMPONENT BALANCE TO ISO 21940-11, G1 (6.9.3.4) :
SHRINK-FIT-LIMITED MOVEMENT IMPELLERS (9.2.2.3)
ROTATION (VIEWED FROM COUPLING END) :
NUMBER OF IMPELLER VANES PER IMPELLER
NUMBER OF DIFFUSER VANES PER STAGE
NUMBER OF VOLUTES PER STAGE
IMPELLERS INDIVIDUALLY SECURED (9.2.2.1) :

COUPLING: (7.2)
MANUFACTURER
MODEL
RATING (kW / 100 RPM)
SPACER LENGTH (7.2.2.d) mm
SERVICE FACTOR (7.2.4) 1.5
RIGID-TYPE
COUPLING WITH HYDRAULIC FIT (7.2.11)
COUPLING WITH PROPRIETARY CLAMPING DEVICE (7.2.12)
COUPLING IN COMPLIANCE WITH (7.2.5)(7.2.2.f)
COUPLING BALANCE TOLERANCE (7.2.2.g)

GUARDS (7.3)
COUPLING AND SHAFT GUARD STANDARD (7.3)
IGNITION HAZARD ASSESSMENT PER EN 13463-1 (7.3.2.2; 7.3.3.4)
COUPLING GUARD MATERIAL (7.3.2.1; 7.3.3.3) :
SHAFT GUARD MATERIAL (7.3.2.1; 7.3.3.3) :
SPARK RESISTANT MATERIAL REQUIRED (7.3.3.4) :

BASEPLATE

BASEPLATE REQUIRED
BASEPLATE DIMENSIONS (LxW) (mm) :
BASEPLATE CONSTRUCTION (7.4)
BASEPLATE DRAINAGE (7.4.1)
MOUNTING :
NON-GROUT CONSTRUCTION (7.4.3):
SUPPLIED WITH : • GROUT AND VENT HOLES
• DRAIN CONNECTION
PROVIDE SPACER PLATE UNDER ALL EQUIPMENT FEET (7.4.14)
BOLT OH 3/4/5 PUMP TO PAD / FOUNDATION :
PROVIDE SOLEPLATE FOR OH 3/4/5 PUMPS :
MECHANICALLY ADJUSTABLE CHOCKS BY VENDOR (7.4.15)

DATASHEET No. _____ Rev: _____ SHEET _____ of _____

Figure N.2c—SI Units Pump Process Data Sheet (Continued)

CENTRIFUGAL PUMP DATASHEET
API 610 - 13th EDITION

Note	INSTRUMENTATION		SEAL SUPPORT SYSTEM MOUNTING		Rev
	INSTRUMENTATION PER API-670 (7.5.2)		MOUNTED ON PUMP BASEPLATE (7.6.1.5)		
	ACCELEROMETERS (7.5.2.1)		LOCATION ON OR OFF BASEPLATE (7.4.6) :		
	NUMBER OF ACCELEROMETERS				
	MOUNTING LOCATION:		INTERCONNECTING PIPING BY		
	PROVISION FOR MTG ONLY (7.5.2.1.3)				
	FLAT THREADED SURFACE REQUIRED (6.10.2.13)		**MECHANICAL SEAL (6.8)**		
			API 682 DATASHEET ATTACHED :		
	VIBRATION PROBES (7.5.2.2)		ADDITIONAL CENTRAL FLUSH PORT (6.8.14)		
	VIBRATION PROBES REQUIRED (7.5.2.1.2)		HEATING JACKET REQ'D. (6.8.10)		
	NUMBER PER RADIAL BEARING				
	NUMBER PER AXIAL BEARING				
	FLAT THREADED SURFACE REQUIRED (6.10.2.13)		**HEATING AND COOLING (6.1.28-6.1.33)**		
	MONITORS AND CABLES SUPPLIED BY (7.5.2.4)		COOLING REQ'D		
			COOLING WATER PIPING PLAN (ANNEX B)		
			COOLING WATER PIPING		
	TEMPERATURE DETECTORS (7.5.2.2)		FITTINGS		
	TEMP. PROBES REQUIRED (7.5.2.2)		COOLING WATER PIPING MATERIALS		
	PROVISIONS FOR MOUNTING ONLY (6.10.2.2)		COOLING WATER REQUIREMENTS:	SUPPLY PRESSURE	kPa
	RADIAL BEARING TEMP.			BEARING HOUSING	m^3/s
	NUMBER PER RADIAL BEARING			HEAT EXCHANGER	m^3/s
	THRUST BEARING TEMP.			TOTAL COOLING WATER	m^3/s
	NUMBER PER THRUST BEARING ACTIVE SIDE		HEATING MEDIUM		
	NUMBER PER THRUST BEARING INACTIVE SIDE		HEATING PIPING		
	TEMP. GAUGES (WITH THERMOWELLS) (9.1.3.5)				
	TEMP. GAUGE LOCATION				
	SUPPLY UPPER/LOWER CASING RTD'S FOR WARMUP		**PIPING & APPURTENANCES**		
			TAG ALL ORIFICES (7.6.2.6)		
	PRESSURE GAUGE TYPE		SOCKET WELD UNION ON 1st SEAL GLAND NIPPLE (7.6.2.6.5)		
	PRESSURE GAUGE LOCATION		MANIFOLD AUX PIPING SYSTEMS AT SKID EDGE (7.6.2.26)		

PRESSURE VESSEL DESIGN CODE REFERENCES

THESE REFERENCES SHALL BE PROVIDED BY THE MANUFACTURER

CASTING FACTORS USED IN DESIGN (PER TABLE 3) _____

SOURCE OF MATERIAL PROPERTIES (6.2.65) _____

WELDING AND REPAIRS (6.12.3.1)

THESE REFERENCES SHALL BE PROVIDED BY THE PURCHASER. (DEFAULT TO TABLE 11 IF NO PURCHASER PREFERENCE IS STATED)

ALTERNATIVE WELDING CODES AND STANDARDS

WELDING REQUIREMENT (APPLICABLE CODE OR STANDARD)

ALTERNATIVE WELDER/OPERATOR QUALIFICATION STANDARD

ALTERNATIVE WELDING PROCEDURE QUALIFICATION STANDARD

NON-PRESSURE RETAINING STRUCTURAL WELDING STANDARD (BASEPLATES OR SUPPORTS)

STANDARD FOR MAGNETIC PARTICLE OR LIQUID PENETRANT EXAMINATION (PLATE EDGES)

STANDARD FOR POSTWELD HEAT TREATMENT

STANDARD FOR POSTWELD HEAT TREATMENT OF CASING FABRICATION WELDS

MATERIAL INSPECTION

THESE REFERENCES SHALL BE PROVIDED BY THE PURCHASER

DEFAULT TO TABLE 14 _____ TABLE 14 INSPECTION CLASS: _____

ALTERNATIVE MATERIAL INSPECTIONS AND ACCEPTANCE CRITERIA (SEE TABLE 15, 8.2.1.9.2)

TYPE OF INSPECTION	METHOD	FOR FABRICATIONS	FOR CASTINGS
RADIOGRAPHY			
ULTRASONIC INSPECTION			
MAGNETIC PARTICLE INSPECTION			
LIQUID PENETRANT INSPECTION			
VISUAL INSPECTION (all surfaces)			

DATASHEET No. _____ Rev: _____ SHEET _____ OF _____

Figure N.2d—SI Units Pump Process Data Sheet (Continued)

CENTRIFUGAL PUMP DATASHEET
API 610 - 13th EDITION

Note	SURFACE PREPARATION, PAINT & SPARES		TEST		Rev
	MANUFACTURER'S STANDARD (8.4.4.4)		SHOP INSPECTION (8.1.1.1)		
	OTHER (SEE BELOW)		PERF. CURVE & DATA APPROVAL REQ'D PRIOR TO SHIPMENT :		
	SPECIFICATION NO.		TEST WITH SUBSTITUTE SEAL (8.3.3.3.1)		
	PUMP:		MATERIAL CERTIFICATION REQUIRED (6.12.1.9):	CASING	
	PUMP SURFACE PREPARATION			IMPELLER	
	PRIMER			SHAFT	
	FINISH COAT			OTHER	
	BASEPLATE:		CASTING REPAIR WELD PROCEDURE APPROVAL REQD (6.12.2.7):		
	SURFACE PREPARATION		INSPECTION REQUIRED FOR CONNECTION WELDS (6.12.3.4.1)		
	PRIMER:			MAG PARTICLE	
	FINISH COAT			RADIOGRAPHY	
	UNDERSIDE			LIQUID PENETRANT	
				ULTRASONIC	
	SHIPMENT: (8.4)		HARDNESS TEST REQUIRED (6.12.1.15, 8.2.1.11)		
	EXPORT BOXING REQUIRED		ADDITIONAL SUBSURFACE EXAMINATION (6.12.1.6) (8.2.1.3)		
	PREPARE FOR OUTDOOR STORAGE (NUMBER OF MONTHS) :			FOR:	
				METHOD:	
	SPARE ROTOR ASSEMBLY PACKAGED FOR:		PMI TESTING REQUIRED (8.2.1.12)		
	ROTOR STORAGE ORIENTATION (9.2.8.2)		COMPONENTS TO BE TESTED :		
	SHIPPING & STORAGE CONTAINER SUITABLE FOR		RESIDUAL UNBALANCE TEST (J.4.1)		
	VERTICAL STORAGE (9.2.8.3) :		NOTIFICATION OF SUCCESSFUL PERFORMANCE TEST (8.1.1.3)		
	N2 PURGE REQUIRED (9.2.8.4) :		BASEPLATE (NOZZLE LOAD) TEST (7.4.26)		
			HYDROSTATIC TEST (8.3.2)		
	SPARE PARTS (L.3.4.6)		HYDROSTATIC TEST OF BOWLS & COLUMN (9.3.13.1)		
	START-UP		PERFORMANCE TEST (8.3.3)		
	NORMAL MAINTENANCE		RETEST ON SEAL LEAKAGE (8.3.3.3.2)		
			TEST DATA POINTS TO 8.3.3.4		
			ALTERNATE TEST TOLERANCES PER (8.3.3.9)		
	MASSES kg		NPSH (8.3.5.3)		

ITEM No	PUMP	DRIVER	GEAR	BASE	TOTAL

	NPSH BASED ON 1ST STG ONLY ALLOWED (8.3.5.3.3)		
	PERFORMANCE TEST NPSHA LIMITED TO 110% SITE NPSHA (8.3.3.11.8)		
	RETEST REQUIRED AFTER FINAL HEAD ADJUSTMENT (8.3.3.12.4)		
	COMPLETE UNIT TEST (8.3.5.4.1)		
	SOUND LEVEL TEST (8.3.5.5)		
	CLEANLINESS PRIOR TO FINAL ASSEMBLY (8.2.1.10)		
	OTHER PURCHASER REQUIREMENTS	LOCATION OF CLEANLINESS INSPECTION:	
	COORDINATION MEETING REQUIRED	CASING DISASSEMBLY AFTER TEST FOR DRYING (8.3.3.14)	
	MAXIMUM DISCHARGE PRESSURE TO INCLUDE	MECHANICAL RUN TEST UNTIL OIL TEMP STABLE (8.3.5.2.1)	
	MAX RELATIVE DENSITY (6.2.2)	4 HR. MECH RUN AFTER OIL TEMP STABLE (8.3.5.2.2)	
	MAX DIA. IMPELLERS AND/OR NO OF STAGES (6.2.3)	BRG HSG RESONANCE TEST (8.3.5.7)	
		STRUCTURAL RESONANCE TEST (9.3.9.2)	
	CONNECTION DESIGN APPROVAL (6.12.3.4.2)	REMOVE / INSPECT HYDRODYNAMIC BEARINGS AFTER TEST (9.2.7.4)	
	TORSIONAL ANALYSIS / REPORT (6.9.2.10)	AUXILIARY EQUIPMENT TEST (8.3.5.6)	
	PROGRESS REPORTS (L.3.3)	EQUIPMENT TO BE INCLUDED IN AUXILLIARY TESTS :	
	OUTLINE OF PROCEDURES USED FOR OPTIONAL TESTS (L.2.5)		
	ADDITIONNAL DATA REQUIRING 20 YEARS RETENTION (8.2.1.1)	LOCATION OF AUXILIARY EQUIPMENT TEST :	
	LATERAL ANALYSIS REQUIRED (9.2.4.1.2)		
	ROTOR DYNAMIC BALANCE TO 4W/N (6.9.3.4)		
	INSTALLATION LIST IN PROPOSAL (6.1.1.2)	IMPACT TEST (6.12.4.3) PER EN 13445	
	AFD STEADY STATE FORCED RESPONSE ANALYSIS (6.9.2.3)	PER ASME SECTION VIII	
	TRANSIENT FORCED RESPONSE (6.9.2.4)	ON LIFTING LUGS FOR LOW TEMP (7.4.22.2)	
	BEARING LIFE CALCULATIONS REQUIRED (6.10.1.14)	SPARE PARTS TEST (8.3.5.8)	
	CASING RETIREMENT THICKNESS DRAWING (L.3.6.1.cc.7.xii)		
	CONNECTION BOLTING (7.6.1.10)		
	VENDOR SUBMIT TEST PROCEDURES (8.3.1.1)		
	NO UNWITNESSED PRELIMINARY PERFORMANCE TEST ALLOWED (8.1.1.4)		
	SUBMIT INSPECTION CHECK LIST (8.1.5)		
	ACOUSTIC ANALYSIS OF CROSSOVER PASSAGE (BB3, BB5)(9.2.1.5)		
	API-691 DOCUMENTATION REQUIRED (6.1.3.1)		
	SPECIAL PURPOSE PUMP (6.3.2)		

DATASHEET No. _____ Rev: _____ SHEET _____ OF _____

Figure N.2e—SI Units Pump Process Data Sheet (Continued)

VERTICAL PUMP SUPPLEMENTAL DATASHEET
API 610 - 13th EDITION

VERTICAL PUMP TYPE (Table 1) _____ Rev

REMARKS _____

VERTICAL PUMPS

PUMP THRUST:	(+) UP	(-) DOWN
STATIC THRUST	____ N	____ N
AT MIN FLOW	____ N	____ N
AT RATED FLOW	____ N	____ N
AT MAX FLOW	____ N	____ N
MAX THRUST	____ N	____ N
THRUST BEARING LOCATION		
SEPARATE SOLEPLATE REQUIRED (9.3.8.3.2)		
SOLEPLATE Length x Width	____ mm X	____ mm
SOLEPLATE THICKNESS		____ mm
SEPARATE MOUNTING FLANGE REQUIRED (9.3.8.3.1)		

COLUMN PIPE: _____

- DIAMETER ____ mm
- TOTAL COLUMN LENGTH ____ m
- NUMBER OF SECTIONS ____
- SPACING (LENGTH PER SECTION) ____ m

GUIDE BUSHINGS:

- NUMBER ____
- LINE SHAFT BEARING SPACING ____ mm
- GUIDE BUSHING LUBE: _____

VERTICAL PUMPS (CONT'D)

LINE SHAFT: _____

- LINE SHAFT DIAMETER ____ mm
- TUBE DIAMETER ____ mm
- LINESHAFT CONNECTION _____

LEVEL CONTROL _____

SUCTION STRAINER TYPE _____

IMPELLER COLLETS ACCEPTABLE (6.6.3.2) _____

HARDENED SLEEVES UNDER BEARINGS (9.3.10.4) _____

STRUCTURAL ANALYSIS (9.3.5) _____

SUCTION CAN

- THICKNESS ____ mm
- LENGTH ____ m
- DIAMETER ____ mm

CAN DRAIN PIPED TO SURFACE (9.3.13.4) _____

PRESSURE CASING WALLS USING SEAMLESS PIPE (9.3.2.7) _____

MINIMUM SUBMERGENCE TEST REQUIRED (9.3.9.3) _____

HYDROTEST BOWLS/COLUMN @ MAX DIFF PRESSURE (9.3.13.1) _____

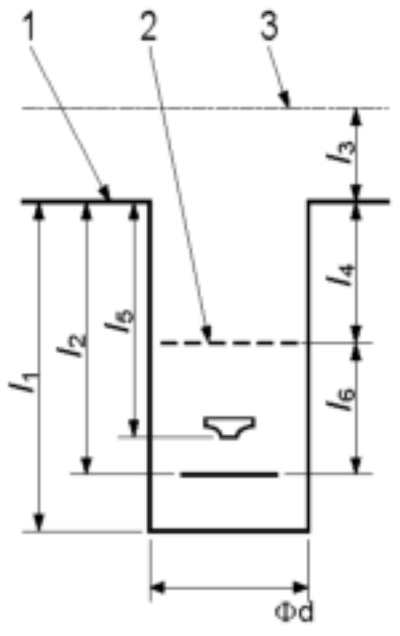
MATERIALS (additional)

SUCTION CAN / BARREL:		LINESHAFT SLEEVES :	
DISCHARGE HEAD :		BEARING RETAINER :	
BOWL SHAFT :		SHAFT ENCLOSING TUBE :	
LINESHAFT :		DISCHARGE COLUMN :	
LINESHAFT HARDFACING :		**PRESSURE RATING:**	**MAWP (kPa)** **HYDRO (kPa)**
BELLMOUTH :		HEAD	
BOWL BEARING :		COLUMN PIPE	
LINESHAFT BEARING :		BOWL	

SUMP ARRANGEMENT

SUMP DIMENSIONS :

GRADE ELEVATION	1 ____	m
LOW LIQUID LEVEL ($l_3 + l_4$)	2 ____	m
C L. OF DISCHARGE	3 ____	m
SUMP DEPTH	l_1 ____	m
PUMP LENGTH	l_2 ____	m
GRADE TO DISCH.	l_3 ____	m
GRADE TO LOW LIQUID LVL	l_4 ____	m
GRADE TO 1ST STG IMPL'R.	l_5 ____	m
SUBMERGENCE REQ'D	l_6 ____	m
SUMP DIAMETER	Φd ____	m



DATASHEET No. _____ Rev: _____ SHEET _____ OF _____

Figure N.2f—SI Units Pump Process Data Sheet (Continued)

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DATASHEET No. _____ Rev: _____ SHEET _____OF_____

Figure N.2g—SI Units Pump Process Data Sheet (Continued)

ProjectDesignData DOCUMENT NUMBER:

	REVISION	0	1	2	3	4
	DATE					
	BY					
	REV/APPR					

PROJECT DESIGN DATA SHEET

U.S. CUSTOMARY

JOB NO. _____ PAGE 1 OF 1
CLIENT _____
LOCATION _____
UNIT NO. _____
ITEM NO. _____
SERVICE _____

APPLICABLE TO: ◉ PROPOSAL ○ PURCHASE ○ AS BUILT REQ'N NO.

SITE DESIGN CONDITIONS

AMBIENT DRY BULB TEMPERATURE		RAINFALL / SNOWFALL (in)	
Average Annual (Summer / Winter)	(°F)	Average, Annual Rainfall	Max 24 Hour Period
Site Maximum (Design)	(°F)	Average, Annual Snowfall	
Site Minimum (Design)	(°F)	SNOW AND ICE LOADING	
Design Basis - Air Cooled Exchangers	(°F)	Ground Snow Load	(in)
RELATIVE HUMIDITY (%)		RAW WATER SOURCE:	
Average @	(°F)	Temperature: (min/max/design)	(°F)
Maximum @	(°F)	EARTHQUAKE LOADING:	
Minimum @	(°F)	Design Code:	
BAROMETRIC PRESSURE (MAX/MIN)	(psig)	Seismic Zone:	Importance Factor:
SOLAR RADIATION (Heat Flux Intensity)	(BTU/ft²)	Short Period Acceleration (S_S) in %g:	
SITE ELEVATION	(ft)	DESIGN WIND SPEED, 3 sec gust:	(mph)
SITE CONDITIONS:		Wind Design Code:	Site Class:
		Exposure Category:	

UTILITIES

STEAM

Pressure (psig)	Rated	Max/Min	Mech Design
High-high			
High			
Medium			
Low			
Temperature (°F)	Rated	Max/Min	Mech Design
High-high			
High			
Medium			
Low			

AIR & NITROGEN

	Pressure (psig)			Temperature (°F)	
	Min	Normal	Design	Normal	Design
Instrument					
Plant Air					
Nitrogen (LP)					
Nitrogen (HP)					

Remarks

COOLING WATER

	Press (psig)	Temp (°F)
Maximum (Supply/Return)		
Minimum (Supply/Return)		
Max. Differential Temperature		
Allowable Differential Pressure		
System Mech Design		

Source

ELECTRICAL POWER

MOTORS			Voltage	Phase	Hertz
Low Volt:	<=	(HP)			
Med Volt:	> & ·	(HP)			
High Volt:	> & ·	(HP)			
High-High Volt:	>	(HP)			
Fractional	<=	(HP)			
HEATERS (incl Lube Oil):					
	>	(kW)			
	<=	(kW)			
Motor Space Htrs >		(kW)			
Motor Space Htrs <		(kW)			
Heaters for panels					
MISCELLANEOUS ELECTRICAL					
Normal Lighting					
U.P.S. - Controls					
U.P.S. - Critical Equip					
Instrument Circuits					
Shutdown / Alarm - Instr.					
Emergency Generator					
Other					

Digital I/O Signals:
Analog I/O Signals:

REMARKS:

Figure N.3—API Project Design Data Sheet

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