ANSI/HI 12.1-12.6-2005

Approved American National Standard

ANSI

American National Standard for

Rotodynamic (Centrifugal) Slurry Pumps

for Nomenclature, Definitions, Applications, and Operation

ANSI/HI 12.1-12.6-2005

Hydraulic Institute

6 Campus Drive
First Floor North
Parsippany, New Jersey
07054-4406
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Approved May 31, 2005
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American National Standard

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Foreword (Not part of Standard)

Purpose and aims of the Hydraulic Institute

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

a) To develop and publish standards for pumps;

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

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

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

e) To support educational and research activities;

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

Purpose of Standards

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

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

Definition of a Standard of the Hydraulic Institute

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

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

Comments from users

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

If a dispute arises regarding contents of an Institute publication or an answer provided by the Institute to a question such as indicated above, the point in question shall be referred to the Executive Committee of the Hydraulic Institute, which then shall act as a Board of Appeals.

Revisions

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

Units of Measurement

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

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

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

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

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Black & Veatch Corp.
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S.A. Fitts Consulting C/O Exeter Energy L.P.
South Florida Water Management District
Southern Company
Suncor Energy, Oil Sands
Syncrude Canada Limited
Weir Specialty Pumps

Committee List

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

Chairman – Graeme Addie, GIW Industries, Inc.

Members	Alternates
Tom Angle, Weir Specialty Pumps	
Bill Beekman, Floway Pumps	
Ralph Gabriel, John Crane, Inc.	
Brian Long, ITT - Industrial Products Group, Goulds Pumps	Robert Manion, ITT – Bell & Gossett

Members—*continued*	Alternates—*continued*
Jim Osborne, A. F. Wilfley & Sons	
Aleks Roudnev, Weir Slurry Group, Inc.	Jim Mangano, Weir Slurry Group, Inc.
Thomas Stirling, Weir Hazleton, Inc.	
Roger Turley, Flowserve Corporation	Bill Adams, Flowserve Corporation
	Fred Buse, Flowserve Corporation
	Robert Stanbury, Flowserve Corporation

Other contributors

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Paul Schmarr, Weir Rubber Engineering
Anders Sellgren, Luleå University of Technology
Harry Tian, GIW Industries, Inc.
Eric Vanhie, EagleBurgmann, LP
Robert Visintainer, GIW Industries, Inc.
Kenneth Wilson - Queens University

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12 Rotodynamic (centrifugal) slurry pumps

12.0 Scope

This Standard is for rotodynamic (centrifugal), single-stage, overhung impeller slurry pumps, horizontal and vertical of industrial types, herein referred to as *slurry pumps*. It includes types and nomenclature; definitions; design and application; and installation, operation, and maintenance.

12.1 Objective

This standard is intended to clearly outline information necessary to define, select, apply, operate, and maintain slurry pumps.

12.1.1 Introduction

This Standard covers slurry pumps used for pumping and/or transporting mixtures of solids and liquids or so-called slurries. Slurries are abrasive and if not taken into account, abrasive slurries may cause high wear and shortened life of pumps. Unlike clear water, slurries alter the performance of the pumps and cause wear to the wet-end parts. Below a certain velocity, some slurries also settle out in the piping causing blockages. These differences are such that if they are not taken into account, the pumps will not work satisfactorily or not at all. For this reason, this Standard includes information about slurries and their effects, necessary to select, apply, operate, and maintain different designs and materials of construction for slurry pumps.

12.1.2 Pump types and nomenclature

Figure 12.1 shows classifications of slurry pumps based on mechanical configuration. Figures 12.3 through 12.10B show typical constructions commonly used for each pump type. Uppercase letter designations are for different manufacturer variants of the same type. Other variations are also acceptable.

While there are no rigid rules about where different mechanical configurations are to be applied, initial cost, wear parts (maintenance) cost, and arrangement

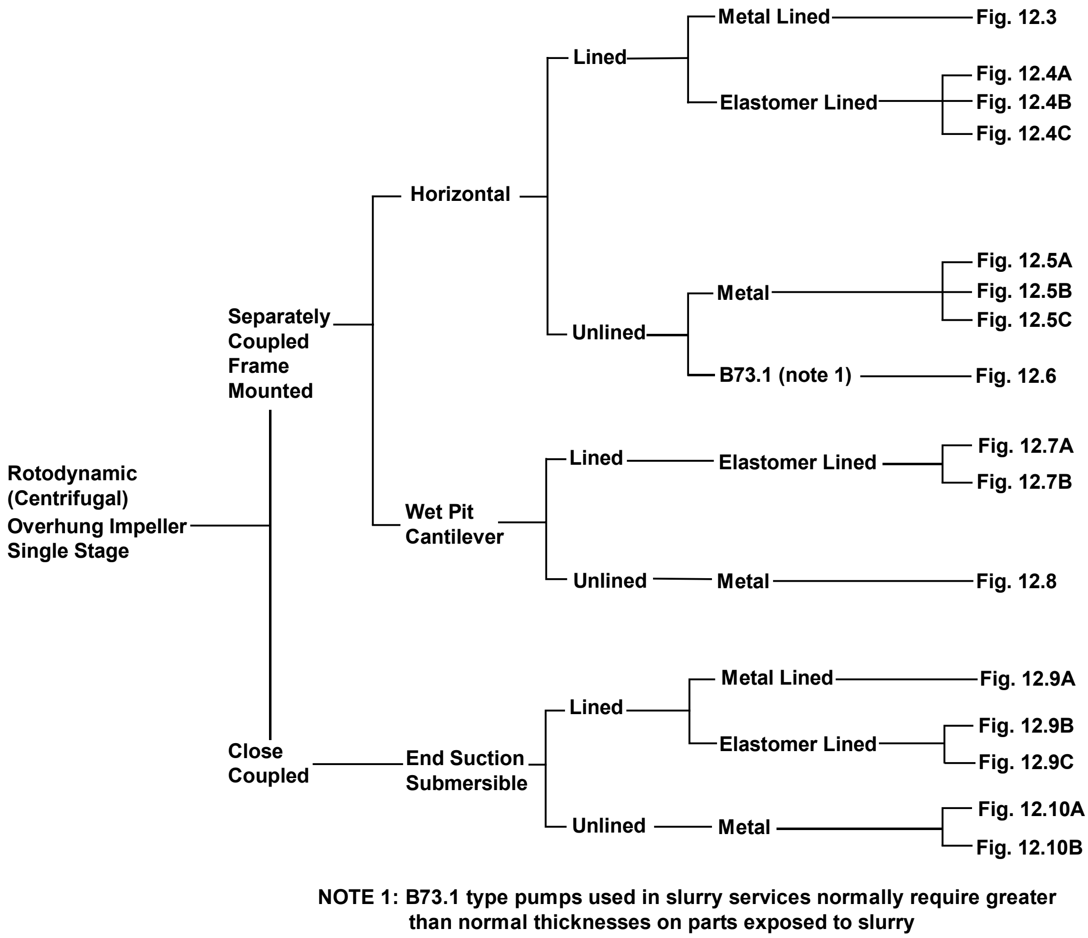


Figure 12.1 — Rotodynamic (centrifugal) slurry pump types

convenience are such that mechanical configurations tend to be aligned to certain services.

The separately coupled, frame-mounted mechanical configurations are preferred for the heavier solids transport wear services described later in Section 12.3.4.2. The hard metal pumps are preferred for services involving the largest sizes of solids. Elastomer pumps, by virtue of the needed support, must be of the lined type.

Cantilevered wet pit pumps are used in plant mining process service but are more widely used in the lighter class wear services for cleanup and lower concentration slurries. These pumps usually are limited to no more than 300-mm (12-in.) discharge size.

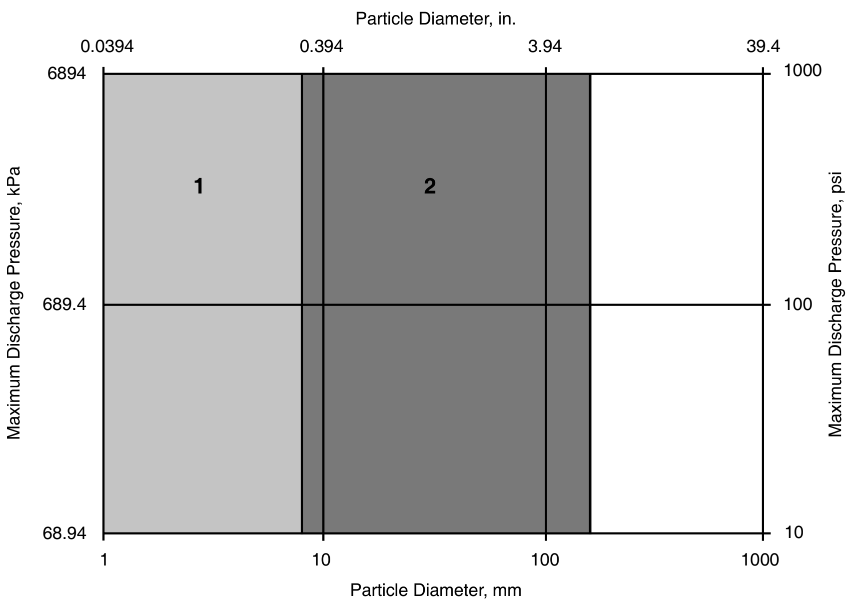
The close-coupled submersible pump types are similar to the cantilevered wet pit pumps, mostly employed on clean-up services, but there may be areas where they are used as process pumps. These are also limited to smaller sizes.

12.1.3 Definition of slurry

Slurry is a mixture of solids (specific gravity greater than 1) in a liquid carrier, usually water. It is often used as a means to transport solids. Slurries also occur when solids are present as an incidental part of the process. The properties of the solids and liquid as well as the amount of solids are variable. The solids size may vary from a few micrometers, often referred to as *microns*, up to hundreds of millimeters and the solids may settle below a certain transport velocity. The properties of slurry, therefore, are highly variable. Slurry may behave like a Newtonian or non-Newtonian fluid. It may be abrasive and/or corrosive depending on the composition. Slurry pumps are usually employed to move slurries with solids concentrations between 2% and 50% by volume and specific gravities of the slurry up to 5.3. Slurries with solids of wood, paper, and other organic materials also exist but are not covered by this document.

12.1.4 Definition of slurry pumps

A slurry pump here is defined as a pump suitable for pumping a liquid containing abrasive particles. Slurry pumps vary in construction depending upon the properties of the slurry to be pumped; they are generally more robust than the pumps used in clean-liquid services, and they often have replaceable wear parts. Figure 12.2 shows how various materials are typically used in slurry pumps.



Zone 1. Rubber or metal

NOTE: Total head generation with rubber impellers is generally limited to 40 m (131 ft)/stage

Zone 2. Metal

Figure 12.2 — Typical material types for discharge pressure and particle size

The maximum pressures noted are for one or more single-stage pumps, operating in series, at one location. The maximum permissible head per stage is shown in Table 12.6 located in Section 12.3.5 of this document.

12.1.5 Overhung impeller

The impeller is mounted on the end of a shaft, which is cantilevered or "overhung" from its bearing supports.

12.1.6 Frame mounted

In this group, the casing and impeller are mounted on a bearing frame that is separately coupled (not part of the driver). The frame may provide total support for the entire pump or the casing may also have feet for additional support.

12.1.7 Cantilevered wet pit

The impeller is overhung and the shaft is oriented vertically. The bearing housing is mounted above a pit. The casing and impeller extend into the pit below the liquid level.

12.1.8 Submersible

A close-coupled, overhung design, with a submersible motor. In service, these pumps are normally submerged in the pumped fluid.

12.1.9 Lined type

These designs incorporate field replaceable and permanently bonded liners. Field replaceable liners are commonly elastomers or wear-resistant metals located in critical high wear areas. Designs that utilize a permanently bonded layer, such as rubber vulcanized to a metal casing, are also included in this type.

12.1.10 Unlined type

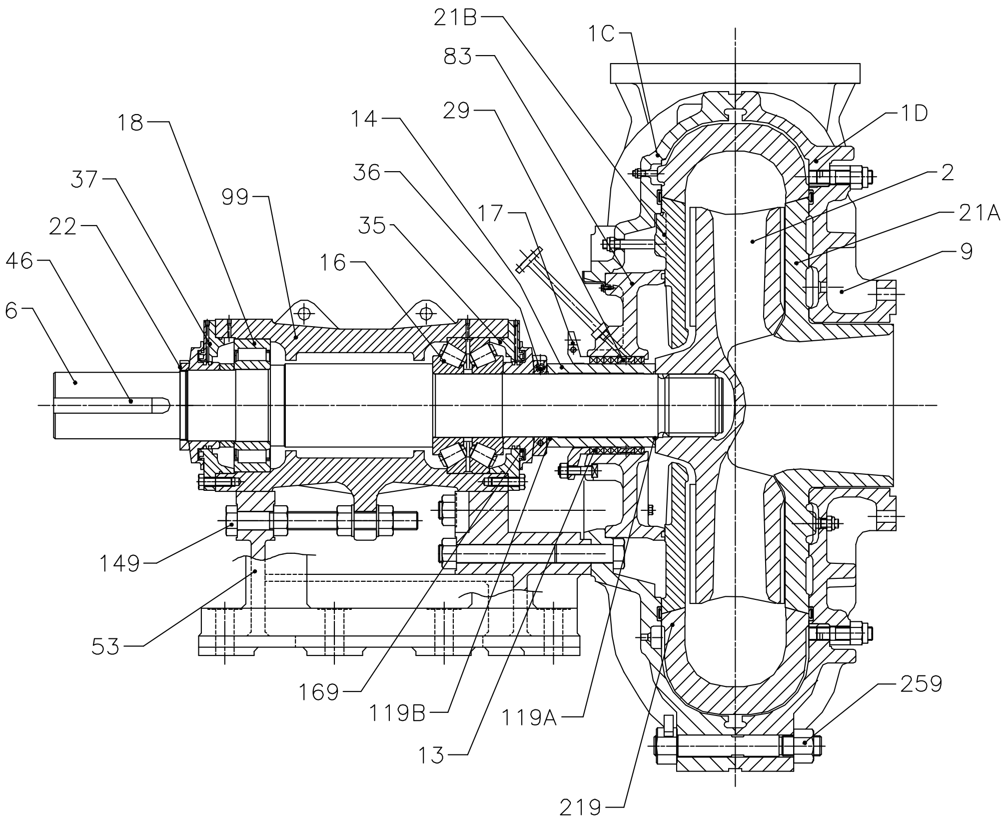
A conventional, radially split, overhung design with a single wall (unlined) casing. It may include the addition of wear plates or sideliners.

12.1.11 Construction drawings

The construction drawings on the following pages are intended to provide a means for identifying the various pump types covered by the ANSI/HI Standards and to serve as the basis for a common language between the purchaser, manufacturer, and specification writer.

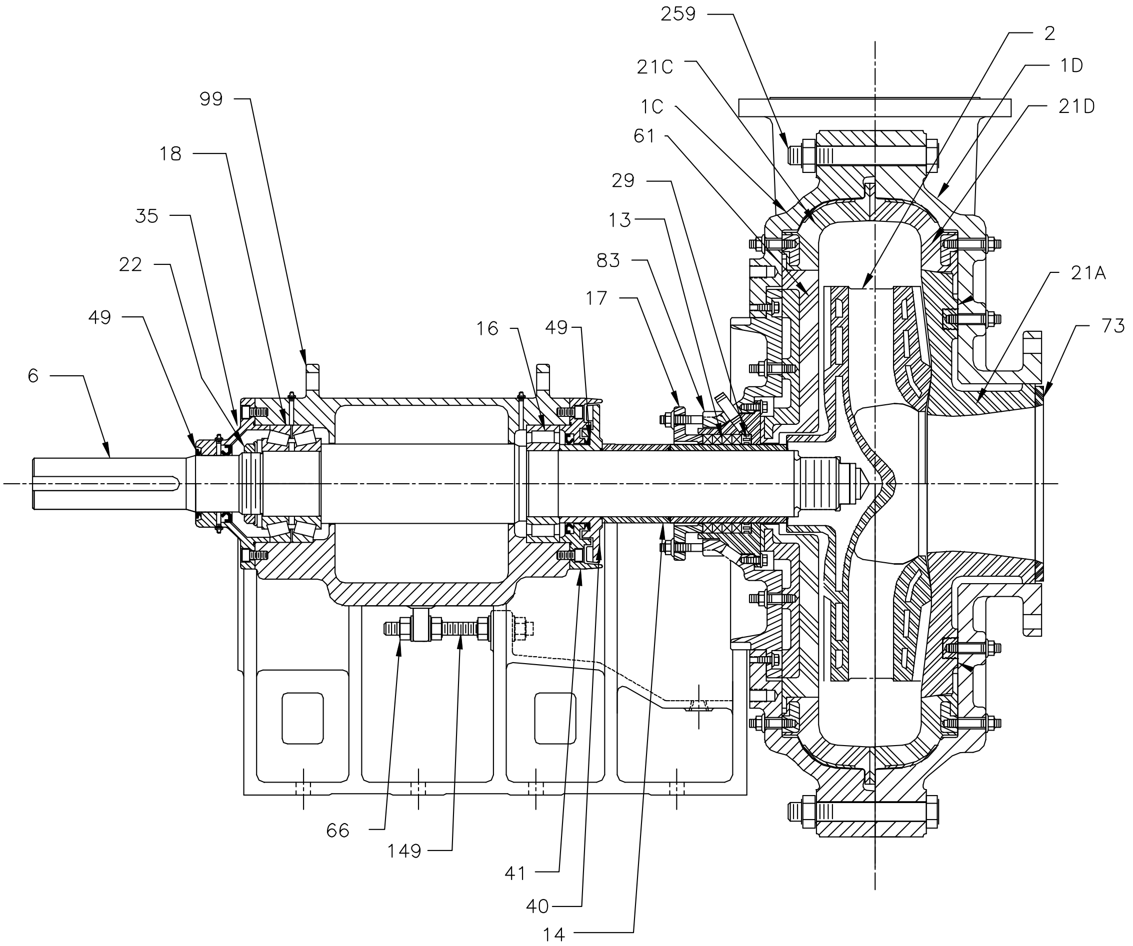
With a few exceptions, the individual part names on these drawings are numbered such that rotating parts have been assigned even numbers while non-rotating parts have been assigned odd numbers. It should be noted that the part names used are the most common industry names, but other manufacturing names are allowable.

In cases where a pump may use two or more parts of the same generic type but different geometries, such as gaskets or bearings, this difference is indicated by the addition of a letter suffix to the item number (e.g., 73A, 73B, etc.).



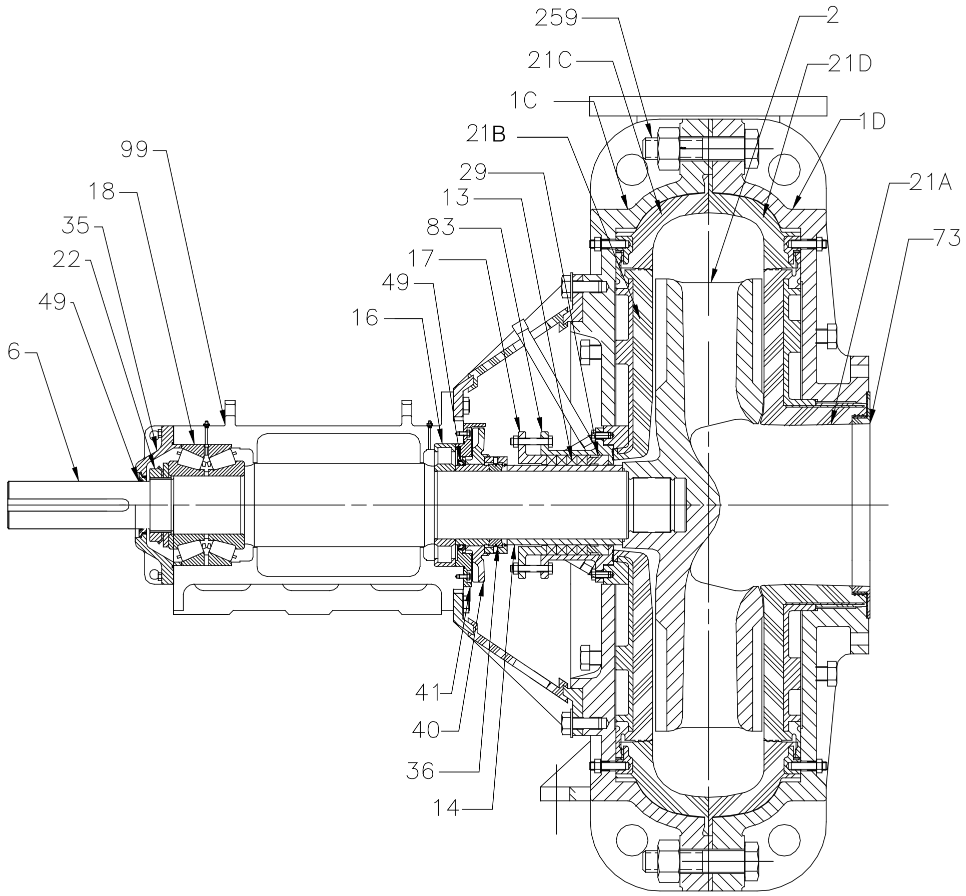
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
9	Cover, suction	13	Packing
14	Sleeve, shaft	16	Bearing, inboard
17	Gland	18	Bearing, outboard
21A	Liner, suction cover	21B	Liner, stuffing box cover
22	Locknut, bearing	29	Ring, lantern
35	Cover, bearing, inboard	36	Collar, release
37	Cover, bearing, outboard	46	Key, coupling
53	Base	67	Shim
83	Stuffing box	99	Housing, bearing
119A	O-ring	119B	O-ring
149	Screw, impeller adjusting	169	Seal, bearing housing
219	Liner, casing		

Figure 12.3 — Overhung impeller, separately coupled, single-stage, frame-mounted, metal-lined pump



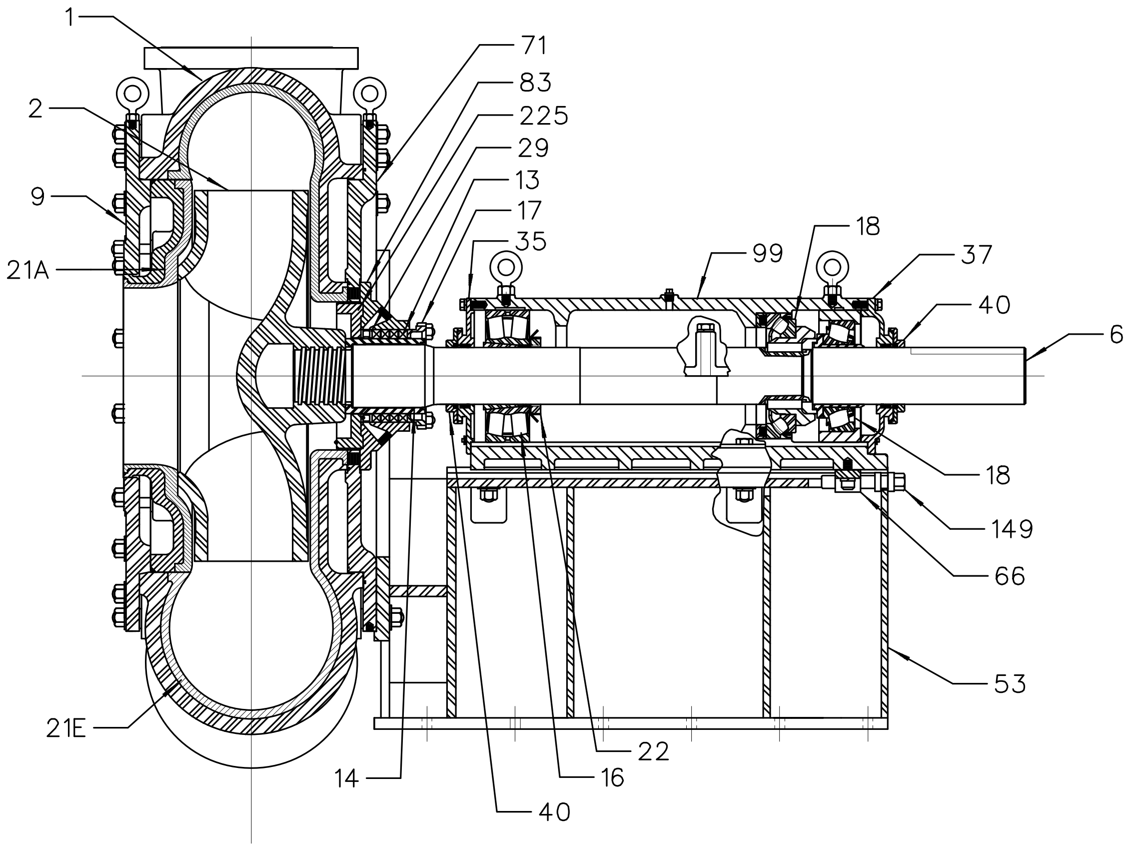
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21C	Liner, gland half	21D	Liner, suction half
22	Locknut, bearing	29	Ring, lantern
35	Cover, bearing, inboard	40	Deflector
41	Cap, bearing, inboard	49	Seal, bearing cover, outboard
21B	Liner, stuffing box cover	66	Nut, shaft adjustment
73	Gasket	83	Stuffing box
99	Housing, bearing	149	Screw, impeller adjusting
259	Bolt, casing		

Figure 12.4A — Overhung impeller, separately coupled, single-stage, frame-mounted, elastomer-lined pump



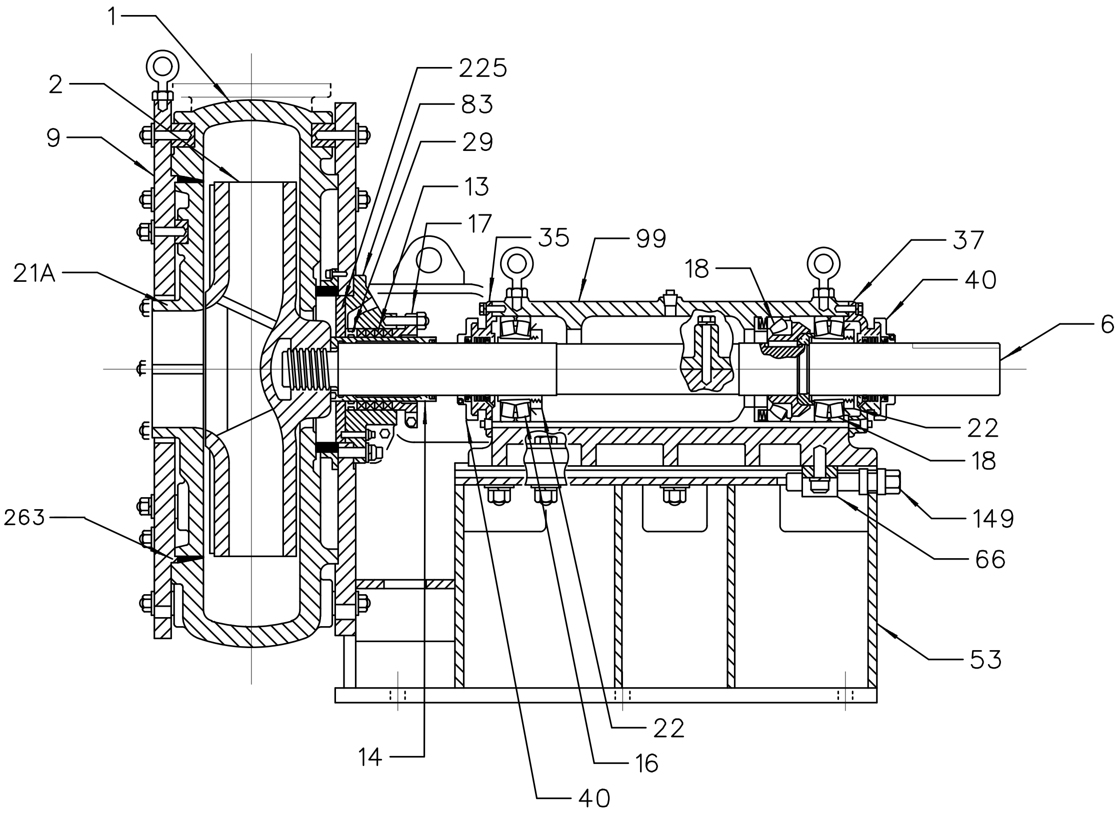
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21B	Liner, stuffing box cover	21C	Liner, gland half
21D	Liner, suction half	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
36	Collar, release	40	Deflector
41	Cap, bearing, inboard	49	Seal, bearing cover, outboard
73	Gasket	83	Stuffing box
99	Housing, bearing	259	Bolt, casing

Figure 12.4B — Overhung impeller, separately coupled, single-stage, frame-mounted, elastomer-lined pump, adjustable sideliners



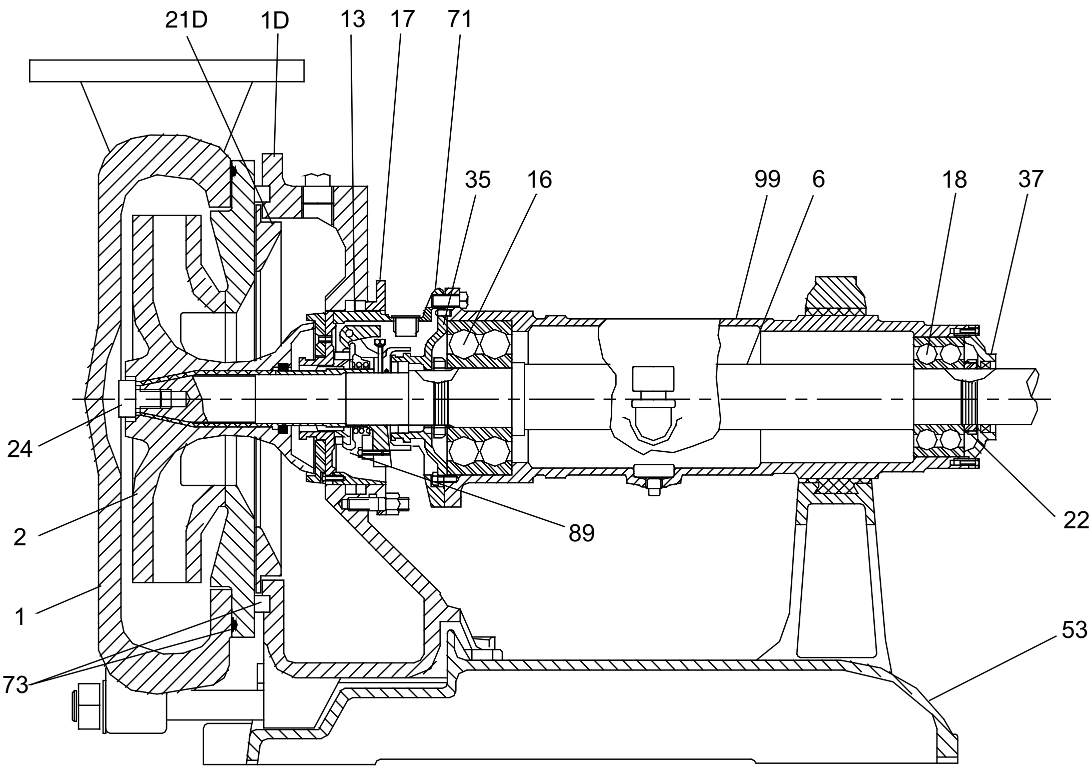
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21E	Liner, vulcanized	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
37	Cover, bearing, outboard	40	Deflector
53	Base	66	Nut, shaft adjusting
71	Adapter	83	Stuffing box
99	Housing, bearing	149	Screw, impeller adjusting
225	Plate, wear		

Figure 12.4C — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, vulcanized-elastomer-lined pump



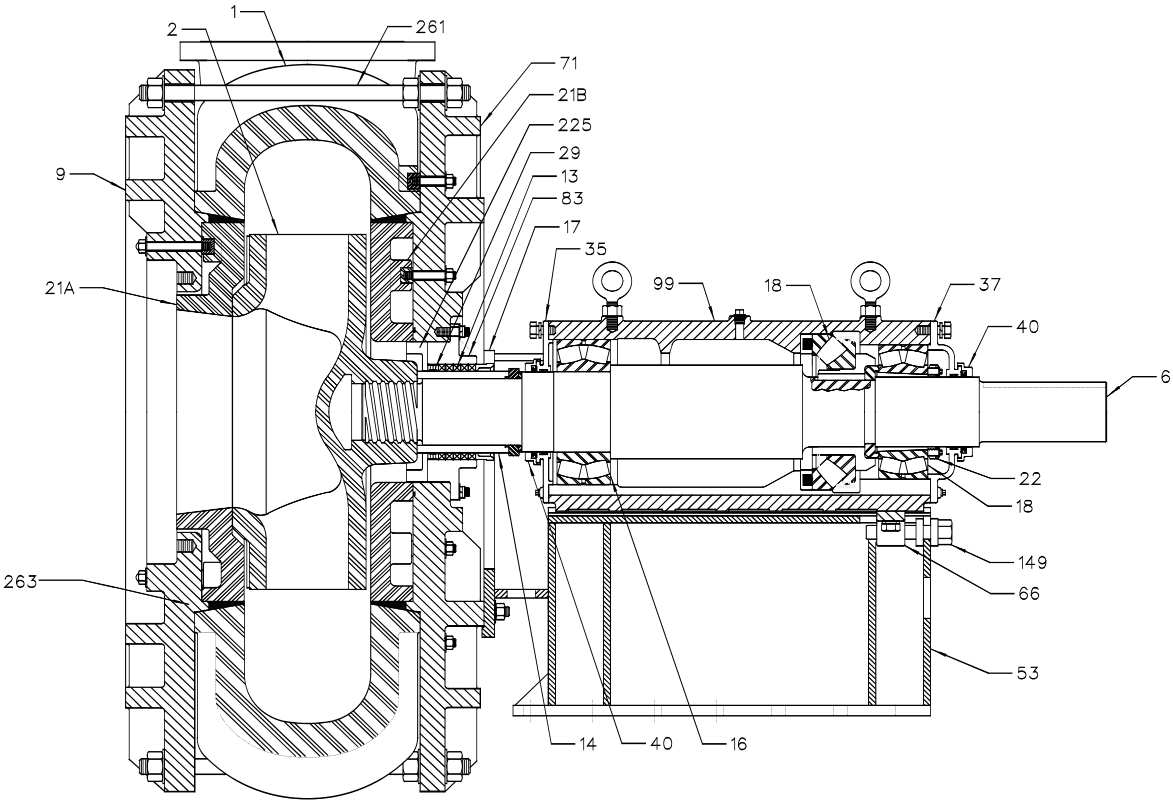
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
22	Locknut, bearing	29	Ring, lantern
35	Cover, bearing, inboard	37	Cover, bearing, outboard
40	Deflector	53	Base
66	Nut, shaft adjusting	83	Stuffing box
99	Housing, bearing	149	Screw, impeller adjusting
225	Plate, wear	263	Gasket, snap ring

Figure 12.5A — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, metal, unlined casing pump



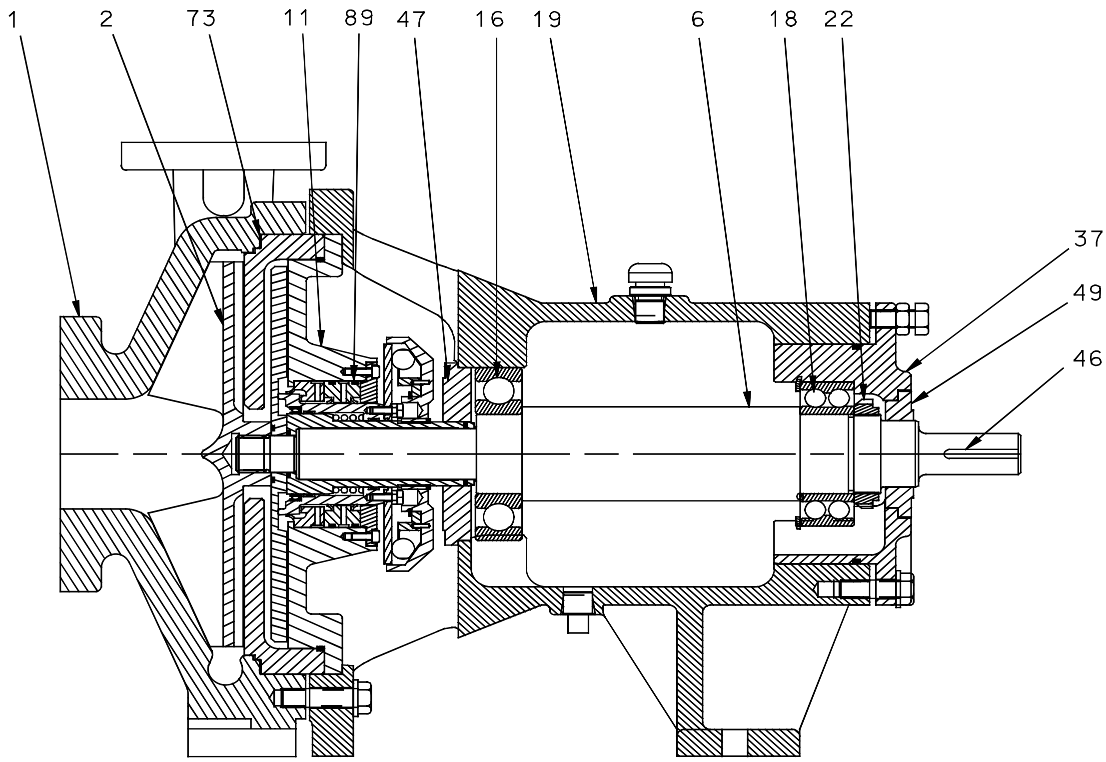
1	Casing	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	16	Bearing, inboard
17	Gland	18	Bearing, outboard
21D	Liner, suction half	22	Locknut, bearing
24	Nut, impeller	35	Cover, bearing, inboard
37	Cover, bearing, outboard	53	Base
71	Adapter	73	Gasket
89	Seal	99	Housing, bearing

Figure 12.5B — Overhung impeller, separately coupled, single-stage, frame-mounted, side inlet, metal, unlined casing pump



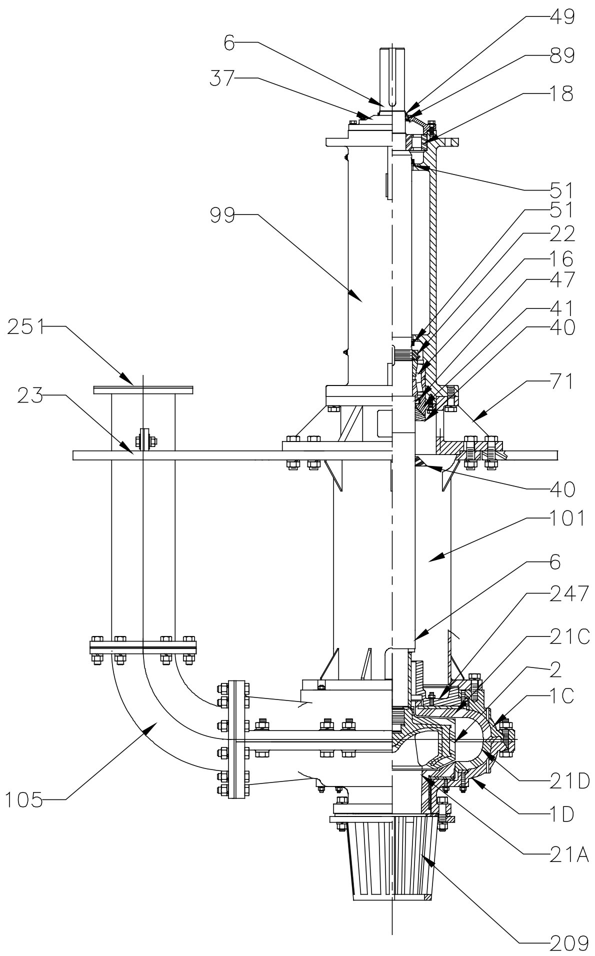
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21B	Liner, stuffing box cover	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
37	Cover, bearing, outboard	40	Deflector
53	Base	66	Nut, shaft adjusting
71	Adapter	83	Stuffing box
99	Housing, bearing	149	Screw, impeller adjusting
225	Plate, wear	261	Tie bolt
263	Gasket, snap ring		

Figure 12.5C — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, metal, tie bolt plate construction pump



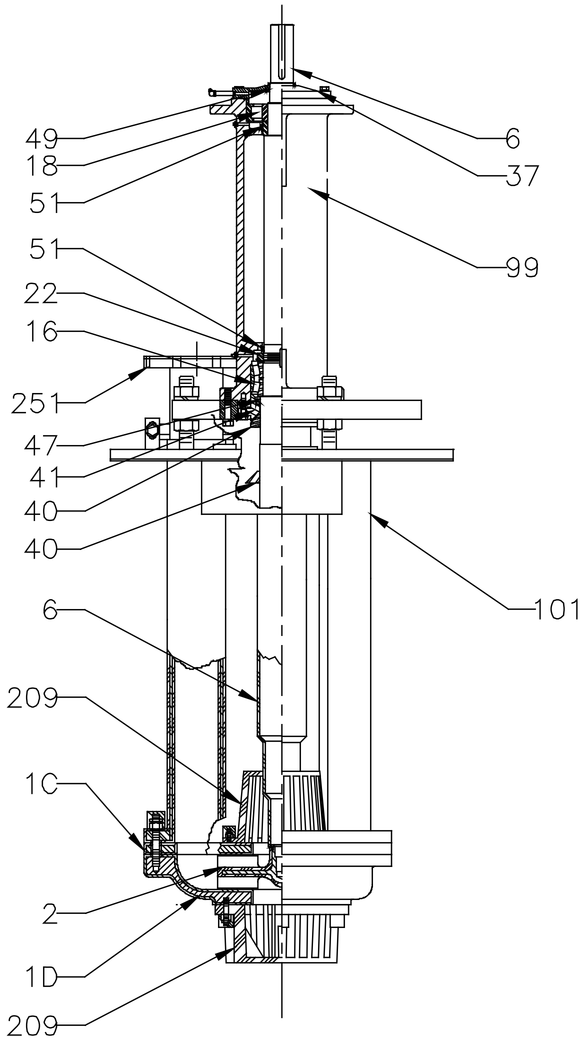
1	Casing	2	Impeller
6	Shaft	11	Cover, stuffing box or seal chamber
16	Bearing, inboard	18	Bearing, outboard
19	Frame	22	Locknut, bearing
37	Cover, bearing, outboard	46	Key, coupling
47	Seal, bearing cover, inboard	49	Seal, bearing cover, outboard
21B	Liner, stuffing box cover	73	Gasket
89	Seal		

Figure 12.6 — Overhung, open impeller, separately coupled, single-stage, frame-mounted, metal, ANSI B73.1 type pump



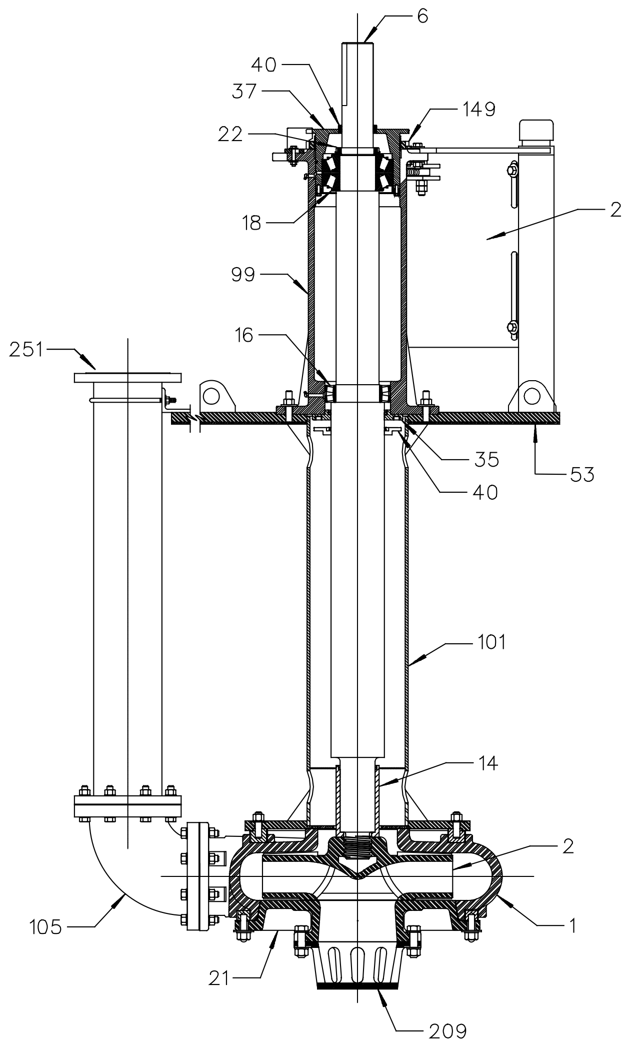
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
16	Bearing, inboard	18	Bearing, outboard
21A	Liner, suction cover	21C	Liner, gland half
21D	Liner, suction half	22	Locknut, bearing
23	Base plate	37	Cover, bearing, outboard
40	Deflector	41	Cap, bearing, inboard
47	Seal, bearing cover, inboard	49	Seal, bearing cover, outboard
51	Retainer, grease	71	Adapter
89	Seal	99	Housing, bearing
101	Pipe, column	105	Elbow, discharge
209	Strainer	247	Adaptor, casing
251	Flange, discharge		

Figure 12.7A — Overhung impeller, separately coupled, single-stage, wet pit cantilever, elastomer-lined, single suction pump



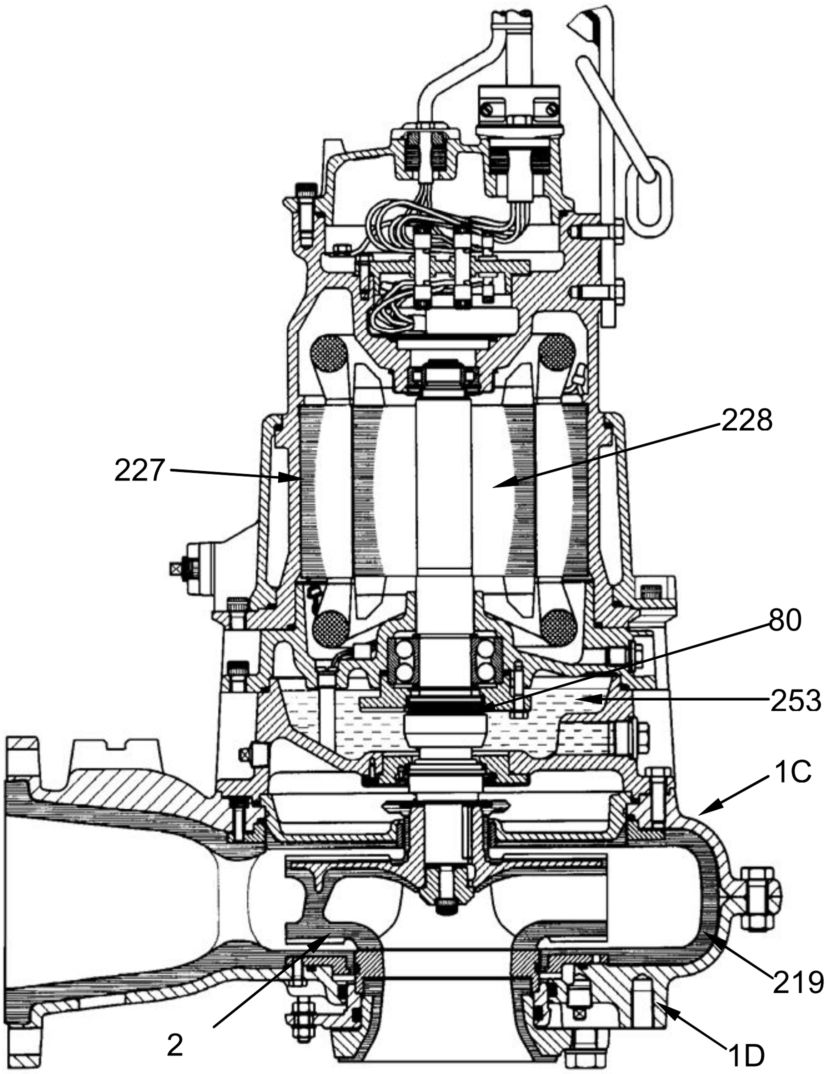
1C	Casing, upper	1D	Casing, lower
2	Impeller	6	Shaft
16	Bearing, inboard	18	Bearing, outboard
22	Locknut, bearing	37	Cover, bearing, outboard
40	Deflector	41	Cap, bearing, inboard
47	Seal, bearing cover, inboard	49	Seal, bearing cover, outboard
51	Retainer, grease	99	Housing, bearing
101	Pipe, column	209	Strainer
251	Flange, discharge		

Figure 12.7B — Overhung impeller, separately coupled, single-stage, wet pit cantilever, elastomer, vulcanized-lined, double suction pump



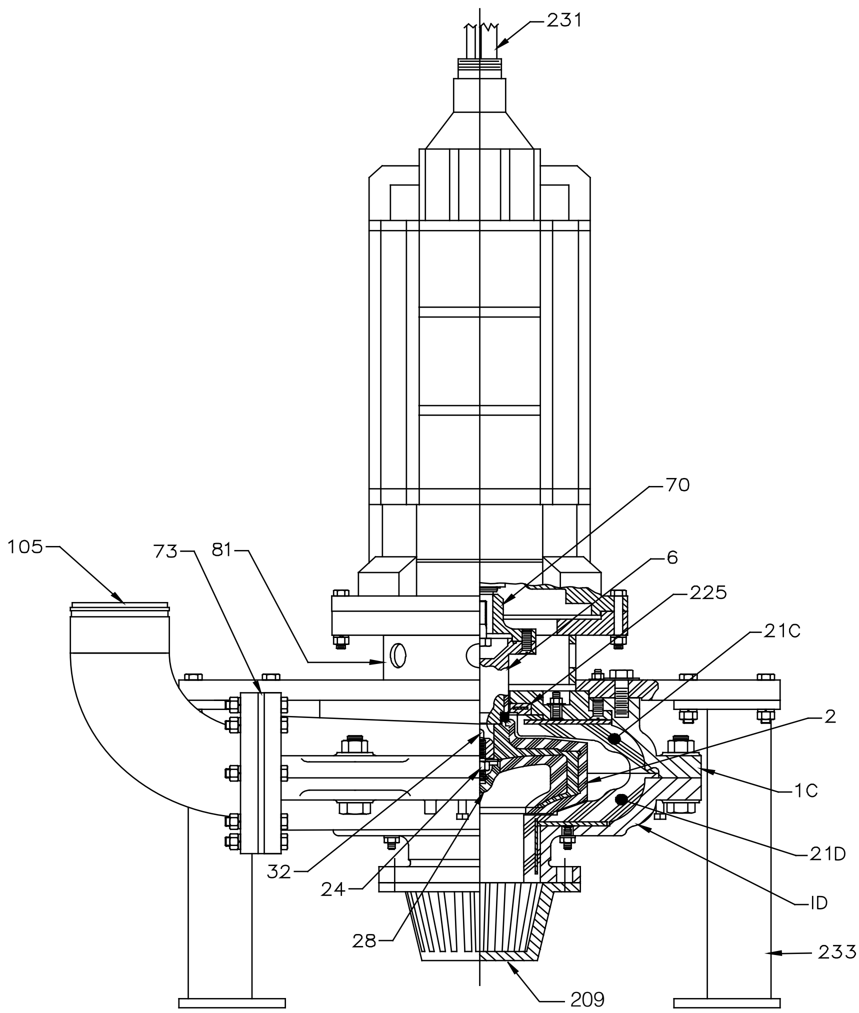
1	Casing	2	Impeller
6	Shaft	14	Sleeve, shaft
16	Bearing, inboard	18	Bearing, outboard
21	Liner, frame	22	Locknut, bearing
35	Cover, bearing, inboard	37	Cover, bearing, outboard
40	Deflector	53	Base
99	Housing, bearing	101	Pipe, column
105	Elbow, discharge	149	Screw, impeller adjusting
209	Strainer	247	Adaptor, casing
251	Flange, discharge		

Figure 12.8 — Overhung impeller, separately coupled, single-stage, wet pit cantilever, unlined, metal, single suction pump



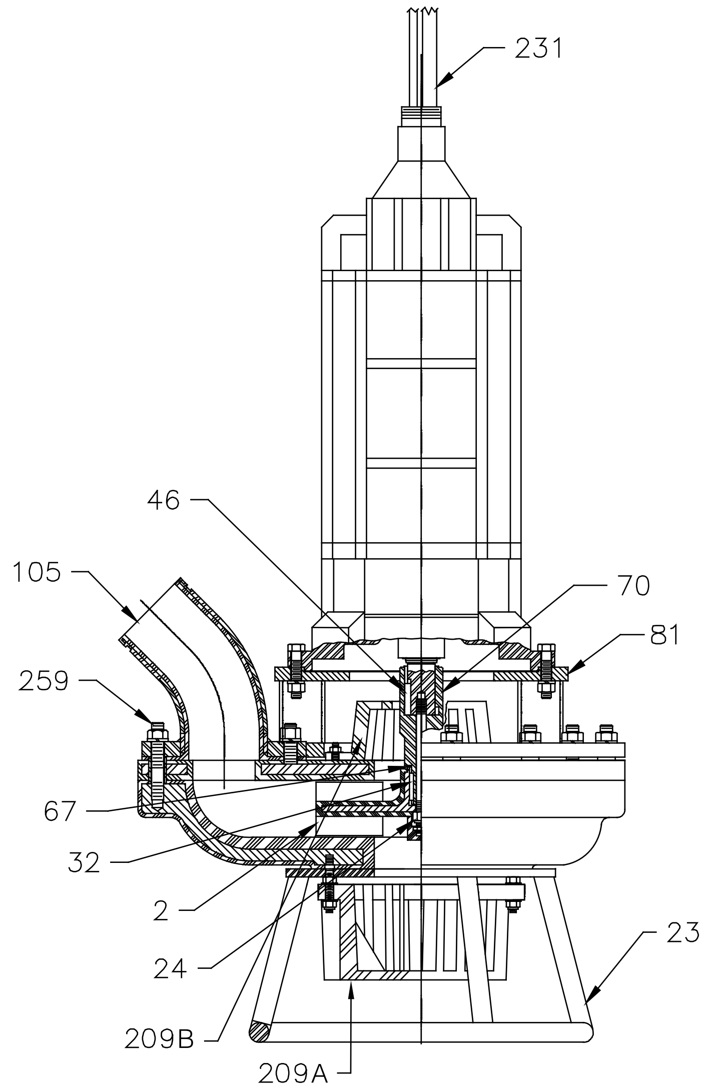
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	80	Seal, mechanical, rotating element
219	Liner, casing	227	Motor, stator
228	Motor, rotor	253	Chamber, barrier liquid, submersible

Figure 12.9A — Overhung impeller, close-coupled, single-stage, submersible, metal-lined, single suction pump



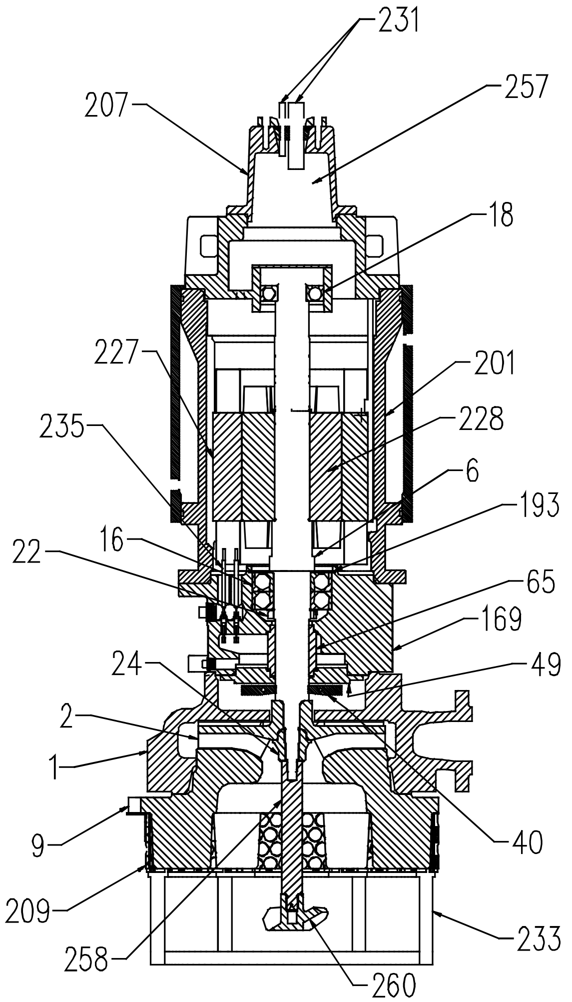
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
21C	Liner, gland half	21D	Liner, suction half
24	Nut, impeller	28	Gasket, impeller screw
32	Key, impeller	70	Coupling, shaft
73	Gasket	81	Pedestal, driver
105	Elbow, discharge	209	Strainer
225	Plate, wear	231	Cable, electric power supply or control
233	Stand, pump		

Figure 12.9B — Overhung impeller, close-coupled, single-stage, submersible, elastomer-lined, single suction pump



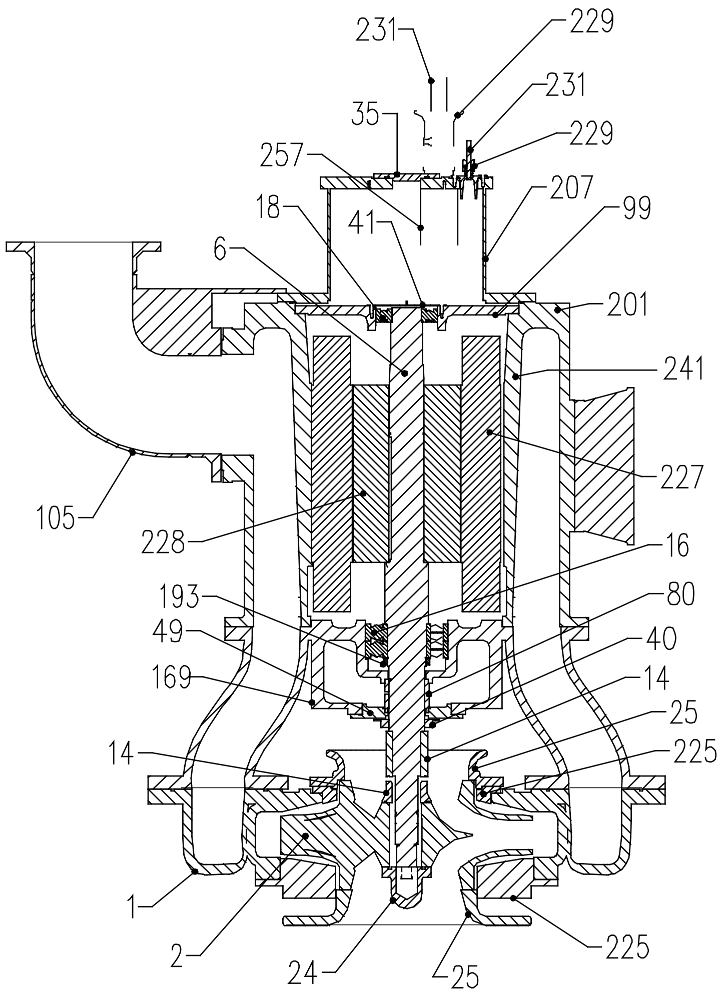
2	Impeller	24	Nut, impeller
32	Key, impeller	46	Key, coupling
67	Shim	70	Coupling, shaft
81	Pedestal, driver	105	Elbow, discharge
209A	Strainer, lower	209B	Strainer, upper
231	Cable, electric power supply or control	233	Stand, pump
259	Bolt, casing		

Figure 12.9C — Overhung impeller, close-coupled, single-stage, submersible, elastomer-lined, double suction pump



1	Casing	2	Impeller
6	Shaft	9	Cover, suction
16	Bearing, inboard	18	Bearing, outboard
22	Locknut, bearing	24	Nut, impeller
40	Deflector	49	Seal, bearing cover, outboard
65	Seal, mechanical, stationary element	169	Seal, bearing housing
193	Retainer, bearing	201	Housing, stator
207	Cover, motor end	209	Strainer
227	Motor, stator	228	Motor, rotor
231	Cable, electric power supply or control	233	Stand, pump
235	Probe, moisture detection	257	Seal, cable, epoxy
258	Extender, shaft	260	Agitator, mechanical

Figure 12.10A — Overhung impeller, close-coupled, single-stage, end suction, metal, submersible pump with agitator



1	Casing	2	Impeller
6	Shaft	14	Sleeve, shaft
16	Bearing, inboard	18	Bearing, outboard
24	Nut, impeller	25	Ring, suction cover
35	Cover, bearing, inboard	40	Deflector
41	Cap, bearing, inboard	49	Seal, bearing cover, outboard
80	Seal, mechanical, rotating element	99	Housing, bearing
105	Elbow, discharge	169	Seal, bearing housing
193	Retainer, bearing	201	Housing, stator
207	Cover, motor end	225	Plate, wear
227	Motor, stator	228	Motor, rotor
229	Clamp, cable	231	Cable, electric power supply or control
241	Jacket, submersible motor	257	Seal, cable, epoxy

Figure 12.10B — Overhung impeller, close-coupled, single-stage, submersible, metal, double suction pump

12.1.12 Part names

Tables 12.1 and 12.2 list the names of most parts that are included in the construction of slurry pumps. The reference numbers are the same as those in Figures 12.3 through 12.10B. The part numbers used are for illustration purposes only and may vary with different manufacturers.

Table 12.1 — Slurry pump nomenclature - alphabetical listing

Part name	Item No.	Abbreviation	Definition
Adapter	71	Adpt	A machined part used to permit assembly of two other parts or as a spacer
Adapter, casing	247	Adpt csg	Part used to mount the pump casing to drive structure
Agitator, mechanical	260	Agtr mech	A device attached to the pump shaft that fluidizes settled solids
Base	53	Base	A pedestal to support a pump
Base plate	23	Base pl	Member on which the pump and driver are mounted
Bearing, inboard	16	Brg inbd	Bearing farthest from the coupling of an end suction pump
Bearing, outboard	18	Brg outbd	Bearing nearest to the coupling of an end suction pump
Bolt, casing	259	Blt csng	A threaded bar used to fasten casing halves together, lined pump
Bolt, tie	261	Blt tie	Threaded bar connecting hub and suction plates
Bushing, bearing	39	Bush brg	Removable portion of a sleeve bearing, in contact with the journal
Bushing, pressure-reducing	117	Bush press red	Replaceable piece used to reduce liquid pressure at the stuffing box by throttling the flow
Bushing, stuffing-box	63	Bush stfg box	Replaceable sleeve or ring in the end of the stuffing box opposite the gland
Bushing, throttle, auxiliary	171	Bush throt aux	Stationary ring or sleeve in the gland of a mechanical seal subassembly to restrict leakage in the event of seal failure
Cable, electric control	231	Cbl elec ctrl	Conductor for motor instrumentation
Cable, electric power supply	231	Cbl elec pwr	Conductor for motor power supply
Cap, bearing, inboard	41	Cap brg inbd	Removable portion of the inboard bearing housing
Cap, bearing, outboard	43	Cap brg outbd	Removable portion of the outboard bearing housing
Casing	1	Csg	Portion of the pump that includes the impeller chamber and volute
Casing, gland half	1 C	Csg gld half	The gland half of a radially split casing
Casing, suction half	1 D	Csg suc half	The suction half of a radially split casing

Table 12.1 — Slurry pump nomenclature - alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Chamber, barrier liquid, submersible	253	Chmbr bar liq subm	Volume located between two seals that, when filled with liquid, acts as a barrier between the liquid being pumped and the motor cavity, submersible pump
Clamp, cable	229	Clmp, cbl	A device to fix the position of a cable, submersible pump
Collar, release	36	Clr rel	Split ring device to ease removal of the impeller
Collar, shaft	68	Clr sft	A ring used to establish a shoulder on a shaft
Column, discharge		Col disch	See pipe, discharge
Coupling half, driver	42	Cplg half drvr	The coupling half mounted on driver shaft
Coupling half, pump	44	Cplg half pump	The coupling half mounted on pump shaft
Coupling, oil pump	120	Cplg oil pump	The coupling for the oil pump
Coupling, shaft	70	Cplg sft	Mechanism used to transmit power from the driver shaft to the driven shaft
Cover, bearing end	123	Cov brg end	Enclosing plate for the end on the bearing housing
Cover, bearing, inboard	35	Cov brg inbd	Enclosing plate for the impeller end of the bearing housing of end suction pumps
Cover, bearing, outboard	37	Cov brg outbd	Enclosing plate for the coupling end of the bearing housing of end suction pumps
Cover, motor end	207	Cov mot end	Removable piece that encloses end(s) of a motor stator housing
Cover, oil bearing cap	45	Cov oil brg cap	A lid or plate over an oil filler hole or inspection hole in a bearing cap
Cover, stuffing box or seal chamber	11	Cov stfg box	A removable piece of an end suction pump casing used to enclose the outboard side of the impeller and includes a stuffing box
Cover, suction	9	Cov suct	A removable piece of an end suction pump casing used to enclose the suction side of the impeller. The suction nozzle may be integral
Deflector	40	Defl	A flange or collar around a shaft and rotating with it to inhibit passage of liquid, grease, oil, or heat along the shaft
Elbow, discharge	105	Ell disch	An elbow in wet pit cantilever or submersible pump by which the liquid leaves the pump
Elbow, suction	57	Ell suct	A curved water passage attached to the pump inlet
Expeller		Explr	A secondary impeller fitted with vanes used to reduce or eliminate pressure at the stuffing box of a slurry pump

Table 12.1 — Slurry pump nomenclature - alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Expelling vanes		Exp Vane	Vanes on the front, back, or both shrouds of a slurry pump impeller used to limit recirculation, to reduce the concentration of solids between the impeller and casing sides, and to reduce the pressure at the stuffing box
Extender, shaft	258	Extdr sft	A part that extends the pump drive shaft outboard of the impeller such that a mechanical agitator can be driven from the pump shaft
Flange, discharge	251	Flg disch	A pipe connection at the pump liquid outlet
Frame	19	Fr	A member of an end suction pump to which are assembled the liquid end and rotating element
Gasket	73	Gskt	Resilient material used to seal joints between parts to prevent leakage
Gasket, impeller screw	28	Gskt imp scr	Resilient material used to seal the joint between the hub of the impeller and the impeller screw
Gasket, shaft sleeve	38	Gskt sft slv	Resilient material used to provide a seal between the shaft sleeve and the impeller
Gasket, snap ring	263	Gskt snp ring	Trapezoidal section shaped resilient material used to provide a seal between liner and casing
Gauge, sight, oil	143	Ga sight oil	Device for visual determination of oil level
Gland	17	Gld	A follower that compresses packing in a stuffing box or retains a stationary element of a mechanical seal
Gland, stuffing box, auxiliary	133	Gld stfg box aux	A follower for compression of packing in an auxiliary stuffing box
Guard, coupling	131	Gld cplg	A protective shield over a shaft coupling
Housing, bearing	99	Hsg brg	A body in which a bearing or bearing set is mounted
Housing, bearing, inboard	31	Hsg brg inbd	See bearing (inboard) and bearing housing
Housing, bearing, outboard	33	Hsg brg outbd	See bearing (outboard) and bearing housing
Housing, seal	237	Hsg seal	A body in which the shaft seals are mounted
Housing, stator	201	Hst sttr	A body in which a stator core assembly is mounted
Impeller	2	Imp	The bladed member of the rotating assembly of a pump that imparts the principal energy to the liquid pumped
Jacket, submersible motor	241	Jkt, sub mtr	A chamber located in close proximity to the submersible pump motor windings, in which coolant is available for maintaining acceptable motor temperature
Journal, thrust bearing	74	Jnl thr brg	Removable cylindrical piece mounted on the shaft that turns in the bearing. It may have an integral thrust collar
Key, bearing journal	76	Key brg jnl	A parallel-sided piece used for preventing the bearing journal from rotating relative to the shaft

Table 12.1 — Slurry pump nomenclature - alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Key, coupling	46	Key cplg	A parallel-sided piece used to prevent the shaft from turning in a coupling half
Key, impeller	32	Key imp	A parallel-sided piece used to prevent the impeller from rotating relative to the shaft
Liner, casing	219	Lnr csg	A replaceable metal or elastomer insert that provides a renewable waterway in the casing of slurry pumps
Liner, frame	21	Lnr fr	A part within the frame carrying one or more of the bearings
Liner, gland half	21 C	Lnr gld half	A part within the casing, gland half
Liner, stuffing box cover	21 B	Lnr stfg box cov	A part within the stuffing-box cover
Liner, suction cover	21 A	Lnr suct cov	A part within the suction cover
Liner, suction half	21 D	Lnr suct half	A part within the casing, suction half
Liner, vulcanized	21 E	Lnr vul	An elastomer liner within the casing
Locknut, bearing	22	Lknut brg	Fastening that positions an antifriction bearing on a shaft
Locknut, coupling	50	Lknut cplg	A fastener holding a coupling half in position on a tapered shaft
Lockwasher	69	Lkwash	A device to prevent loosening of a nut
Motor, rotor	228	Mtr rotr	The rotating part of an electric motor, submersible pump
Motor, stator	227	Mtr statr	The stationary part of an electric motor, submersible pump
Nut, impeller	24	Nut imp	A threaded piece used to fasten the impeller on the shaft
Nut, shaft adjusting	66	Nut sft adj	A threaded piece for altering the axial position of the rotating assembly
Nut, shaft sleeve	20	Nut sft slv	A threaded piece used to locate the shaft sleeve on the shaft
O-ring	119	Ring O	A radial or axial elastomer seal
Packing	13	Pkg	A pliable lubricated material used to provide a seal around the portion of the shaft located in the stuffing box
Pipe, column	101	Pipe col	A vertical pipe by which the pumping element is suspended
Pipe, discharge		Disch Pipe	Pipe used to provide a convenient discharge connection at or above the floor mounting plate of vertical sump pumps

Table 12.1 — Slurry pump nomenclature - alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Plate, floor mounting		Mtg Pl	Plate used to suspend a vertical sump pump over the sump that it draws from
Plate, side	61	Pl side	A replaceable piece in the casing or cover of a pump to maintain a close clearance along the impeller face
Plate, wear	225	Wp pl	A removable, axial clearance part used to protect the casing, stuffing box, or suction cover from wear
Probe, moisture detection	235	Prob moist detct	Conductivity probe to allow the detection of water leakage past the primary seals, submersible pump
Pump, oil	121	Pump oil	A device for supplying pressurized lubricating oil
Retainer, bearing	193	Ret brg	A device used to support the shaft bearing
Retainer, grease	51	Ret grs	A contact seal or cover to keep grease in place
Ring, lantern	29	Ring ltrn	An annular piece used in the stuffing box to establish a path for lubricating or flushing liquid around the shaft sleeve
Ring, oil	60	Ring oil	A rotating ring used to carry oil from the reservoir to the bearings
Ring, suction cover	25	Ring suct cov	A stationary ring to protect the suction cover at the running fit with the impeller ring or impeller
Screw, impeller	26	Scr imp	A special screw to fasten the impeller to the shaft
Screw, impeller, adjusting	149	Scr imp adj	A special screw to adjust the axial movement of shaft/ impeller or sideliner to control front seal face clearance
Seal	89	Seal	A device to prevent the flow of a liquid or gas into or from a cavity
Seal, bearing cover, inboard	47	Seal brg cov inbd	A labyrinth seal, bearing isolator, or lip seal for the bearing cover (inboard)
Seal, bearing cover, outboard	49	Seal brg cov outbd	A labyrinth seal, bearing isolator, or lip seal for the bearing cover (outboard)
Seal, bearing housing	169	Seal brg hsg	A contact seal for a bearing housing on the stuffing-box end having a smooth, flat seal face lined against the rotating element
Seal, cable, epoxy	257	Seal cbl epxy	Where an insulating resin is used to seal the electric supply cable entry to the motor housing, submersible pump
Seal, cable jacket	255	Seal cbl jkt	A resilient component that stops the passage of liquid between the jacket of an electric cable and a component enclosure, submersible pump
Seal, mechanical, rotating element	80	Seal mech rot elem	A subassembly consisting of multiple parts mounted to the pump shaft within the stuffing box and having a smooth flat sealing face.

Table 12.1 — Slurry pump nomenclature - alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Seal, mechanical, stationary element	65	Seal mech sta elem	A subassembly consisting of one or more parts mounted in or on a stuffing box and having a smooth flat sealing face
Shaft	6	Sft	The cylindrical member on which the impeller is mounted and through which power is transmitted to the impeller
Shield, oil retaining	107	Shld oil retg	A device to prevent oil leaking from the bearing housing
Shim	67	Shim	A piece of material placed between two members to adjust their position
Sleeve, impeller hub	34	Slv imp hub	A replaceable, cylindrical wear part mounted on the extended pump impeller hub
Sleeve, shaft	14	Slv sft	A cylindrical piece fitted over the shaft to protect the shaft through the stuffing box and that may also serve to locate the impeller on the shaft
Spacer, bearing	78	Spcr brg	Sleeve that fits over a shaft to locate antifriction bearings
Spacer, coupling	88	Spcr cplg	A cylindrical piece used to provide axial space for the removal of the rotating assembly without removing the driver
Strainer	209	Str	Device used to prevent oversized objects from entering pump
Strainer, lower	209 A	Str, lwr	A device used to prevent large objects from entering the pump, located above the impeller centerline
Strainer, upper	209 B	Str, upr	A device used to prevent large objects from entering the pump, located below the impeller centerline
Stand, pump	233	Stnd pmp	A part that will support the pump weight
Stuffing box	83	Stfg box	A portion of the casing through which packing and a gland or a mechanical seal is placed to prevent leakage
Stuffing box, auxiliary	75	Stfg box aux	A recessed portion of the gland and cover of a mechanical seal subassembly designed to accommodate one or more rings of packing
Suction extension		Suct Ext	See tailpipe
Support, discharge pipe	249	Supt disch pipe	A device to support the discharge pipe
Tailpipe		Tailpipe	A length of pipe used to extend the suction inlet of vertical slurry pumps
Thrower (oil or grease)	62	Thwr (oil or grs)	A disc rotating with the pump shaft to carry the lubricant from the reservoir to the bearing

Table 12.2 — Slurry pump nomenclature - numerical listing

1	Casing
1 C	Casing, gland half
1 D	Casing, suction half
2	Impeller
6	Shaft
9	Cover, suction
11	Cover, stuffing box or seal chamber
13	Packing
14	Sleeve, shaft
16	Bearing, inboard
17	Gland
18	Bearing, outboard
19	Frame
20	Nut, shaft sleeve
21	Liner, frame
21A	Liner, suction cover
21B	Liner, stuffing box cover
21C	Liner, gland half
21D	Liner, suction half
21E	Liner, vulcanized
22	Locknut, bearing
23	Base plate
24	Nut, impeller
25	Ring, suction cover
26	Screw, impeller
28	Gasket, impeller screw
29	Ring, lantern
31	Housing, bearing, inboard
32	Key, impeller
33	Housing, bearing, outboard
34	Sleeve, impeller hub
35	Cover, bearing, inboard
36	Collar, release
37	Cover, bearing, outboard
38	Gasket, shaft sleeve
39	Bushing, bearing
40	Deflector
41	Cap, bearing, inboard
42	Coupling half, driver
43	Cap, bearing outboard
44	Coupling half, pump
45	Cover, oil bearing cap
46	Key, coupling
47	Seal, bearing cover, inboard
49	Seal, bearing cover, outboard
50	Locknut, coupling
51	Retainer, grease
53	Base
57	Elbow, suction
60	Ring, oil
61	Plate, side
62	Thrower (oil or grease)
63	Bushing, stuffing-box
65	Seal, mechanical, stationary element
66	Nut, shaft adjusting
67	Shim

68	Collar, shaft
69	Lockwasher
70	Coupling, shaft
71	Adapter
73	Gasket
74	Journal, thrust bearing
75	Stuffing box, auxiliary
76	Key, bearing journal
78	Spacer, bearing
80	Seal, mechanical, rotating element
83	Stuffing box
88	Spacer, coupling
89	Seal
99	Housing, bearing
101	Pipe, column
105	Elbow, discharge
107	Shield, oil retaining
117	Bushing, pressure reducing
119	O-ring
120	Coupling, oil pump
121	Pump, oil
123	Cover, bearing end
131	Guard, coupling
133	Gland, stuffing box, auxiliary
143	Gauge, sight, oil
149	Screw, impeller, adjusting
169	Seal, bearing housing
171	Bushing, throttle, auxiliary
193	Retainer, bearing
201	Housing, stator
207	Cover, motor end
209	Strainer
209A	Strainer, lower
209B	Strainer, upper
219	Liner, casing
225	Plate, wear
227	Motor, stator
228	Motor, rotor
229	Clamp, cable
231	Cable, electric control
231	Cable, electric power supply
233	Stand, pump
235	Probe, moisture detection
237	Housing, seal
241	Jacket, submersible motor
247	Adapter, casing
249	Support, discharge pipe
251	Flange, discharge
253	Chamber, barrier liquid, submersible
255	Seal, cable jacket
257	Seal, cable, epoxy
258	Extender, shaft
259	Bolt, casing
260	Agitator, mechanical
261	Bolt, tie
263	Gasket, snap ring

12.1.13 Letter dimensional designations

The letter designations used in Figures 12.11 through 12.13 provide a common means of identifying various pump dimensions and serve as a common language that will be mutually understandable to the purchaser, manufacturers, and to anyone writing specifications for pumps and pumping equipment.

12.2 Definitions

This section defines terms used in slurry pump applications. Symbols, terms, and units are described in Appendix C.1 and subscripts in Table 12.3.

12.2.1 Rate of flow (Q)

The rate of flow of a pump is the total volume throughput per unit of time at suction conditions. It assumes no entrained gases at the stated operating conditions. The term *capacity* is also used. Preferred units are cubic meters/hour (m^3/h) and US gallons/minute (US gpm).

12.2.2 Speed (n)

The number of revolutions of the pump or driver shaft in a given unit of time. Speed is expressed as revolutions per minute (rpm).

12.2.3 Head (H)

Head is a measure of the energy content of the liquid referred to any arbitrary datum. It is expressed in units of energy per unit mass of liquid, divided by acceleration due to gravity. Common measuring units are meters or feet of liquid column.

12.2.3.1 Gauge head (h_g)

The energy of the liquid due to its pressure as determined by a pressure gauge or other pressure measuring device. The measuring units for gauge head are kilopascals (kPa) and pounds per square inch (psi). Negative pressure or vacuum readings can also be expressed in millimeters of mercury (mm H_g) and inches of mercury (in. H_g).

12.2.3.2 Velocity head (h_v)

The kinetic energy of the liquid at a given cross section. Velocity head is expressed by the following equation:$_{oa}$

$$h_v = \frac{v^2}{2g}$$

Where v is obtained by dividing the flow by the cross-section area of the pipe at the point of gauge connection: v = velocity, m/s or ft/s, and g = gravitational acceleration, m/s^2 or ft/s^2.

(Metric)

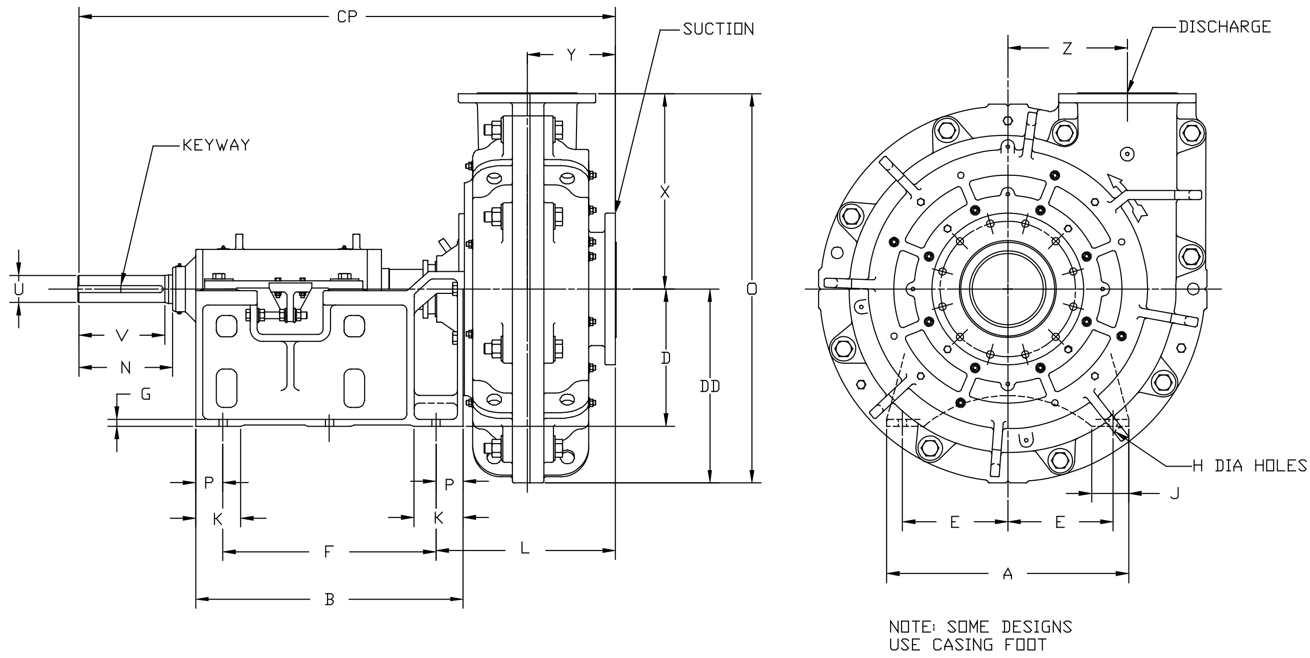
$$v = \frac{278 \times Q}{A} \text{m/s} \quad Q = m^3/h \quad A = mm^2$$

(US customary [USCS] units)

$$v = \frac{0.3205 \times Q}{A} \text{ft/s} \quad Q = \text{US gpm} \quad A = in^2$$

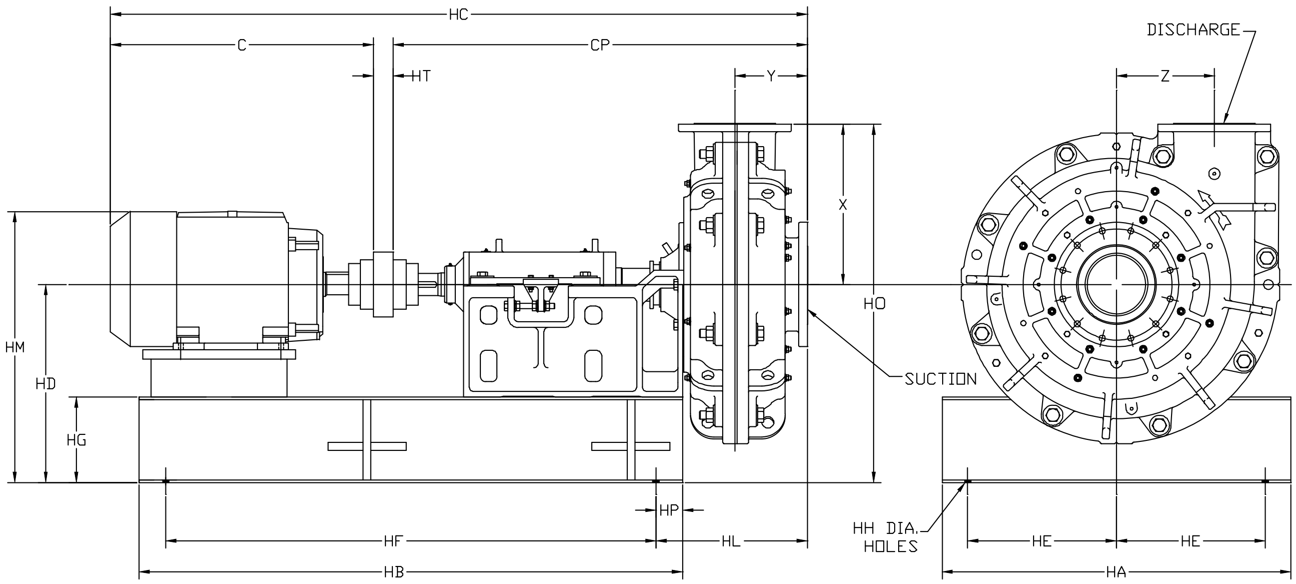
Table 12.3 — Subscripts

Subscript	Term	Subscript	Term	Subscript	Term
1	Test condition or model	g	Gauge	s	Suction
2	Specific condition or prototype	max	Maximum	sm	Slurry concentration corresponding to highest deposit velocity
a	Absolute	min	Minimum	t	Theoretical
atm	Atmospheric	mot	Motor	v	Velocity
b	Barometric	ot	Operating temperature	vp	Vapor pressure
d	Discharge	OA	Overall unit	w	Water



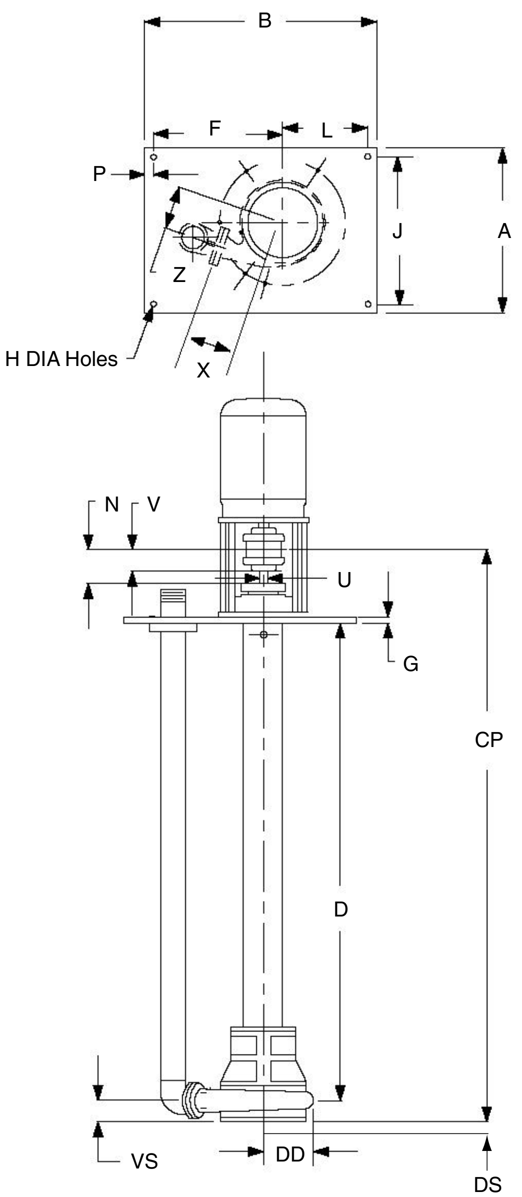
A	Width of base support	K	Length of support pads for hold-down bolts
B	Length of base support	L	Horizontal distance from suction nozzle face to centerline nearest hold-down bolt holes
CP	Length of pump	N	Distance – end of bearing housing to end of shaft
D	Vertical height - bottom of base support to centerline of pump	O	Vertical distance – bottom of casing to discharge nozzle face
DD	Distance - pump centerline to bottom casing	P	Length from edge of support or base plate to centerline of bolt holes
E	Distance from centerline pump to centerline hold-down bolts	U	Diameter of straight shaft – coupling end
F	Distance from centerline to centerline of outer most hold-down bolts	V	Length of shaft available for coupling or pulley
G	Thickness of pads on support or height of base plate	X	Distance from discharge face to centerline of pump
H	Diameter of hold-down bolt holes	Y	Horizontal distance – centerline discharge nozzle to suction nozzle face
J	Width of pads for hold-down bolts	Z	Centerline discharge nozzle to centerline of pump

Figure 12.11 — Horizontal pump dimensions



C	Length of driver	HH	Diameter of hold-down bolt holes
CP	Length of pump	HL	Horizontal distance from suction nozzle face to centerline nearest hold-down bolt holes
HA	Width of base support	HM	Height of unit – bottom of base to top of driver
HB	Length of base support	HO	Vertical distance – bottom of support to discharge nozzle face
HC	Overall length of combined pump and driver when on base	HP	Length from edge of support, or base plate, to centerline of bolt holes
HD	Vertical height – bottom of base support to centerline of pump	HT	Horizontal distance – between pump and driving shaft
HE	Distance from centerline pump to centerline hold-down bolts	X	Distance from discharge face to centerline of pump
HF	Distance from centerline to centerline of hold-down bolt holes	Y	Horizontal distance – centerline discharge nozzle to suction nozzle face
HG	Thickness of pads on support or heights of base plate	Z	Centerline discharge nozzle to centerline of pump

Figure 12.12 — Direct drive pump and motor assembly dimensions



A Width of base support

B Length of base support

CP Length of pump

D Vertical height – bottom of base support to centerline of pump

DD Distance from centerline pump to casing

DS Minimum distance from suction to floor

F Distance from centerline pump to centerline of furthest hold-down bolts

G Thickness of pads on support or height of base plate

H Diameter of hold-down bolt holes

J Distance between hold-down bolts on the short side

L Horizontal distance from suction nozzle face to centerline nearest hold-down bolt holes

N Distance – end of bearing housing to end of shaft

P Length from edge of support or base plate to centerline of bolt holes

U Diameter of straight shaft-coupling end

X Distance from discharge face to centerline of pump

VS Vertical distance – centerline discharge nozzle to suction nozzle face

Z Centerline discharge nozzle to centerline of pump

Figure 12.13 — Vertical pump dimensions

12.2.3.3 Elevation head (Z)

The potential energy of the liquid due to its elevation relative to a datum level, measured to the center of the pressure gauge or liquid surface, expressed in meters or feet of liquid column.

12.2.3.4 Datum

The pump's datum is a horizontal plane that serves as the reference for head measurements. For horizontal pumps this datum is considered by convention to be the centerline of the impeller. The datum for single suction vertical pumps is the eye of the impeller and for double suction pumps the middle plane, as shown in Figure 12.14.

Irrespective of pump mounting, the pump's datum is maintained at the eye of the first-stage impeller.

12.2.3.5 Total suction head (h_s), open suction

For open suction wet pit installations, the impeller is submerged in a pit. The total suction head (h_s) at the datum is the submergence (Z_w).

If the average velocity head of the flow in the pit is small enough to be neglected, then:

$$h_s = Z_w$$

Where:

Z_w = Vertical distance in meters or feet from free water surface to the pump datum.

12.2.3.6 Total suction head (h_s)

Total suction head (h_s), relative to the eye of the first-stage impeller, is the algebraic sum of the suction gauge head (h_{gs}) plus the velocity head (h_{vs}) at point of gauge attachment, plus the elevation head (Z_s) from the suction gauge centerline (or manometer zero) to the pump datum:

$$h_s = h_{gs} + h_{vs} + Z_s$$

The suction head (h_s) is positive when the suction gauge reading is above atmospheric pressure, and negative when the reading is below atmospheric pressure, by the amount exceeding the sum of the elevation head and the velocity head.

12.2.3.7 Total discharge head (h_d)

Total discharge head (h_d) is the sum of the discharge gauge head (h_{gd}) plus the velocity head (h_{vd}) at the gauge attachment point, plus the elevation head (Z_d) from the discharge gauge centerline to the pump datum, where Z_d is positive if the gauge is above the pump datum.

$$h_d = h_{gd} + h_{vd} + Z_d$$

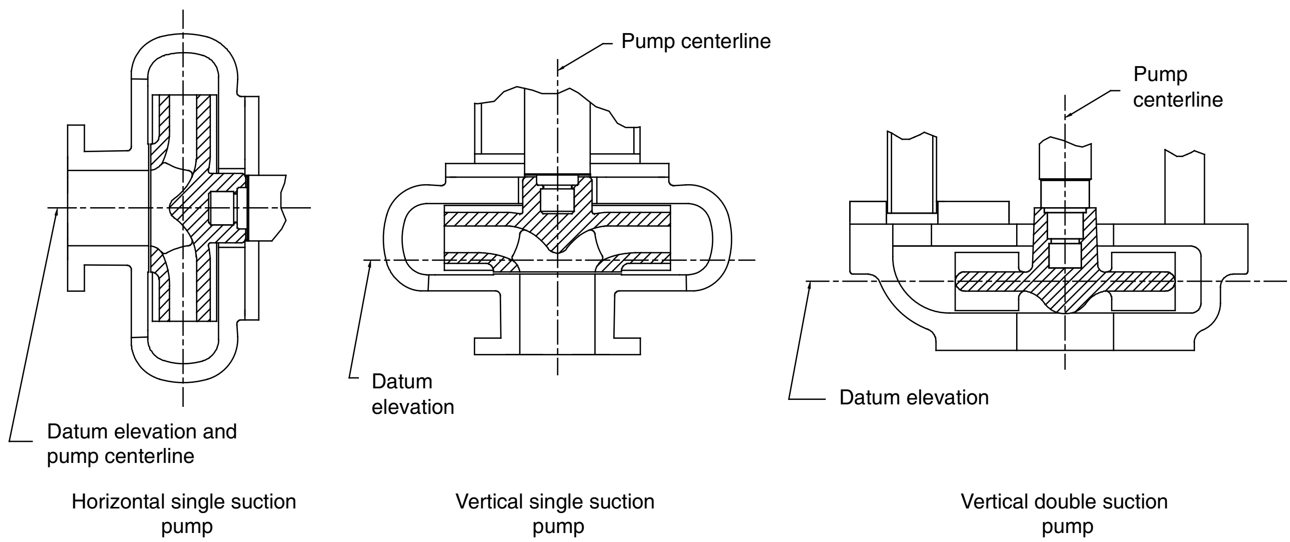


Figure 12.14 — Datum elevations for various slurry pump designs

12.2.3.8 Total head (H_w)

Total head (H_w) is the difference between the total discharge head (h_d) and the total suction head (h_s). This is the head normally specified for pumping applications because the complete characteristics of a system determine the total head required.

$$H_w = h_d - h_s$$

12.2.3.9 Atmospheric head (h_{atm})

Local atmospheric pressure expressed in meters or feet of liquid column.

12.2.3.10 Friction head (h_f)

Friction head is the hydraulic energy required to overcome frictional resistance of a piping system to liquid flow, expressed in meters or feet of liquid column.

12.2.4 Condition points

12.2.4.1 Rated condition point

Rated condition point applies to the capacity, head, net positive suction head, and speed of the pump, as specified by the order.

12.2.4.2 Specified condition point

Specified condition point is the same as rated condition point.

12.2.4.3 Normal condition point

Applies to the point on the rating curve at which the pump will normally operate. It may be the same as the rated condition point.

12.2.4.4 Best efficiency point (BEP)

The flow rate and head at which the pump efficiency is the maximum at a given speed and impeller diameter.

12.2.4.5 Shut off

The condition of zero flow where no liquid is flowing through the pump, but the pump is primed and running.

12.2.4.6 Allowable operating range

The flow range of a pump at the specified speeds, as limited by wear, cavitation, heating, vibration, noise, shaft deflection, fatigue, and other similar criteria. This range shall be defined by the manufacturer.

12.2.5 Suction conditions

12.2.5.1 Submerged suction

A submerged suction exists when the centerline of the pump inlet is below the level of the liquid in the supply tank.

12.2.5.2 Net positive suction head available (NPSHA)

Net positive suction head available is the absolute suction head of liquid, determined at the first-stage impeller datum, minus the absolute vapor pressure of the liquid at inlet process conditions, expressed in meters or feet:

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

Where:

$$\text{Absolute suction head} = h_{sa} = h_{atm} + h_s$$

$$\text{or NPSHA} = h_{atm} + h_s - h_{vp}$$

12.2.5.3 Net positive suction head required (NPSHR [NPSH3])

The amount of suction head required to prevent more than 3% loss in total differential head from the first stage of the pump at a specific capacity, expressed in meters or feet.

12.2.5.4 Maximum suction pressure

This is the highest suction pressure (or lowest absolute pressure) to which the pump will be subjected during operation.

12.2.6 Power

12.2.6.1 Electric motor input power (P_{mot})

The electrical input power to the motor, normally defined by reading voltage and current and expressed by calculating kilowatts.

12.2.6.2 Pump input power (P_p)

The power delivered to the pump shaft from the electric motor or other driver. It is also called *brake horsepower* and is expressed in kilowatts or horsepower.

12.2.6.3 Pump output power (P_w)

The power imparted to the liquid by the pump. It is also called *water power.*

(Metric)

$$P_w = \frac{Q \times H \times S_m}{366}$$

(US units)

$$P_w = \frac{Q \times H \times S_m}{3960}$$

12.2.6.4 Pump efficiency (η_p)

This is the ratio of the energy imparted to the liquid by the pump (P_w) to the energy delivered to the pump shaft (P_p), expressed in percent.

$$\eta_p = \frac{P_w}{P_p}$$

12.2.6.5 Overall efficiency (η_{OA})

This is the ratio of the energy imparted to the liquid (P_w) by the pump, to the energy supplied to the motor (P_{mot}), or the ratio of the water horsepower to the power input to the primary driver, expressed in percent.

$$\eta_{OA} = \frac{P_w}{P_{mot}}$$

12.2.7 Pump pressures

12.2.7.1 Working pressure (p_d)

The maximum discharge pressure in the pump when it is operated at rated speed and suction pressure for the given application.

12.2.7.2 Maximum allowable casing working pressure

The highest pressure, at a specified pumping temperature, for which the pump casing is designed. This maximum pressure shall be equal to or greater than the maximum discharge pressure.

12.2.7.3 Test pressure

The hydrostatic pressure applied to demonstrate that the pump, when subjected to hydrostatic pressures, will not leak or fail structurally as defined in ANSI/HI 1.6.

12.2.8 Mechanical seal terms

12.2.8.1 Pusher seal

Seal design where the secondary seal in the axially flexible assembly slides on the pump shaft or cartridge seal sleeve to compensate for wear and misalignment. The most common pusher seal secondary seal is an elastomeric O-ring.

12.2.8.2 Nonpusher seal

Seal design where the secondary seal in the axially flexible assembly is not required to slide in contact with the pump shaft or cartridge seal sleeve to compensate for wear and misalignment. Elastomeric bellows and metal bellows seals are examples of nonpusher type seals.

12.2.8.3 Dual seal

Seal design using two or more axially flexible assemblies. The inboard seal of a dual seal arrangement seals the product, and the outboard seals a buffer/barrier fluid.

12.2.8.4 Dual pressurized seal

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

12.2.8.5 Dual unpressurized seal

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

12.2.8.6 Buffer fluid

An externally supplied fluid at a pressure lower than the pump seal chamber to lubricate the outer seal in a dual seal arrangement. The buffer fluid creates a buffer between the product pumped and atmosphere.

12.2.8.7 Barrier fluid

An externally supplied fluid used in the outer seal cavity of a dual seal arrangement at a pressure greater than that in the pump seal chamber creating a barrier between the product pumped and atmosphere to eliminate product leakage to atmosphere.

12.2.8.8 Back-to-back seals

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

12.2.8.9 External flush fluid

A fluid from an external source, not pumped fluid, which is introduced into the stuffing box or seal chamber to cool and lubricate the seal faces. Sometimes termed an *external injection* and designated by ANSI Plan No. 7332.

12.2.8.10 Secondary seal

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

12.2.8.11 Single seal

A seal arrangement with only one mechanical seal per stuffing box or seal chamber.

12.2.9 Slurry terminology

12.2.9.1 Slurry

A mixture consisting of solid particles dispersed in a liquid.

12.2.9.2 Apparent viscosity

The viscosity of a non-Newtonian slurry at a particular rate of shear, expressed in terms applicable to Newtonian fluids.

12.2.9.3 Minimum carrying velocity

The velocity of the specific slurry in a particular conduit, above which the solids remain in suspension and below which solid–liquid separation occurs.

12.2.9.4 Mean effective particle diameter, or average particle size (d50)

The single particle size used to represent certain behavior of a mixture of various sizes of particles in slurry. This particle size is where 50% by weight passes through a designated size screen. The d50 size is normally specified in micrometers but may also be in other units, such as the Tyler Mesh as shown in Figure 12.17. This d50 designation is used by some engineers to calculate system requirements and pump performance. Figure 12.17 shows how it may be used to classify slurries and provides different d50 size equivalents.

12.2.9.5 Solids mixture d85 size

The particle size, where 85% by weight passes through a designated size screen. The d85 size is normally expressed in micrometers, but may also be in other units, such as the as the Tyler Mesh, as shown in Figure 12.17.

12.2.9.6 Maximum particle size

The maximum particle size expected in the slurry, under normal conditions, that has to pass though the pump.

12.2.9.7 Friction characteristic

A term used to describe the resistance to flow, which is exhibited by solid–liquid mixtures moving at various rates of flow in pipes or conduits.

12.2.9.8 Heterogeneous mixture

A mixture of solids and a liquid in which the solids are not distributed uniformly and tend to be more concentrated in the bottom of the pipe.

12.2.9.9 Homogeneous mixture

A mixture of solids and a liquid in which the solids are distributed uniformly.

12.2.9.10 Homogeneous flow (fully suspended solids)

A type of slurry flow in which the solids are thoroughly mixed in the flowing stream and a negligible amount of the solids are sliding along the conduit wall.

12.2.9.11 Nonhomogeneous flow (partially suspended solids)

A type of slurry flow in which the solids are stratified, with a portion of the solids sliding along the conduit wall. Sometimes called *heterogeneous flow* or *flow with partially suspended solids*.

12.2.9.12 Non-settling slurry

A slurry in which the solids do not settle to the bottom of the containment vessel or conduit but remain in suspension, without agitation, for long periods of time.

12.2.9.13 Concentration of solids by volume (C_v)

The actual volume of the solid material in a given volume of slurry, divided by the given volume of slurry, multiplied by 100, and expressed in percent.

12.2.9.14 Concentration of solids by mass or weight (C_w)

The mass (or weight) of dry solids in a given volume of slurry, divided by the total mass (or weight) of that volume of slurry, multiplied by 100, and expressed in percent.

12.2.9.15 Saltation

A condition that exists in a moving stream of slurry when solids settle in the bottom of the stream in random agglomerations that build up and wash away with irregular frequency.

12.2.9.16 Settling slurry

A slurry in that the solids move to the bottom of the containment vessel or conduit at a discernible rate but that remain in suspension if the slurry is agitated constantly.

12.2.9.17 Settling velocity

The rate at which the solids in slurry fall to the bottom of a container of liquid that is not in motion. (Not to be confused with the deposit velocity of slurry.)

12.2.9.18 Deposit velocity (V_s)

Deposit velocity, or velocity at the limit of stationary deposition, is the velocity at which particles form a stationary bed in a moving slurry mixture at a given concentration.

12.2.9.19 Maximum value of deposit velocity (V_{sm})

The deposit velocity, at the slurry concentration, that results in the highest value of V_s.

12.2.9.20 Specific gravity of solids (S_s)

The relative density of solids to that of water. The specific gravity of various materials is shown in Appendix C.3.

12.2.9.21 Specific gravity of slurry (S_m)

The relative density of the slurry mixture to that of water at the same temperature.

12.2.9.22 Slurry head (H_m)

Pumped slurry head, expressed in meters or feet of slurry column.

12.2.9.23 Head ratio (H_r)

Head ratio is the ratio of the head produced in meters or feet of slurry compared with the head produced on water.

$$H_r = \frac{H_m}{H_w}$$

12.2.9.24 Efficiency ratio (η_r)

Ratio of the efficiency realized while pumping a given slurry mixture divided by the efficiency achieved while pumping water, $\eta_r = \eta_m/\eta_w$.

12.2.9.25 Head reduction factor (R_h)

Decimal expression of the value of 1 minus the head ratio (H_r), as $R_h = 1 - H_r$.

12.2.9.26 Efficiency reduction factor (R_η)

Equal to the value of 1 less the efficiency ratio η_r, expressed as $R_\eta = 1 - \eta_r$.

12.2.9.27 Water efficiency (η_w)

Efficiency achieved when pumping clear water, expressed in percent.

12.2.9.28 Slurry efficiency (η_m)

Efficiency realized when pumping a given slurry, expressed in percent.

12.2.9.29 Slurry power (P_m)

Power required to pump a slurry mixture, expressed in kilowatts or horsepower (analogous to P_w).

12.2.9.30 Specific gravity correction factor (C_s)

Correction applied to head reduction factor (R_h) to correct for slurries with solids of specific gravity other than 2.65 ($S_s \neq 2.65$).

$$C_s = \left(\frac{S_S - 1}{1.65}\right)^{0.65}$$

12.2.9.31 Fine-particle correction factor (C_{fp})

Correction applied to head reduction factor (R_h) to correct for slurries containing particles of less than 75 micrometers.

$$C_{fp} = (1 - \text{fractional content of particles by weight} < 75 \text{ micrometers})^2$$

12.2.9.32 Concentration correction factor (C_{cv})

Correction applied to head reduction factor (R_h) to correct for slurries with concentration correction factor (C_v) other than 15%.

$$C_{cv} = \frac{C_v\%}{15}$$

12.2.9.33 Stationary bed

A non-moving bed of stationary solids particles, on the bottom of a flowing pipeline, with carrier liquid passing over the top of the bed.

12.2.9.34 Laminar region

The region of flow and mean velocity inside a pipe, where the internal viscous fluid forces predominate over the inertial fluid forces.

12.2.9.35 Turbulent region

The region of flow and mean velocity inside a pipe, where the inertial fluid forces predominate over the viscous fluid forces.

12.2.9.36 Transition region

The region of flow and mean velocity inside a pipe, where the inertial and viscous fluid forces are approximately equal.

12.2.9.37 Slurry service classes

Class 1 light, Class 2 medium, Class 3 heavy, Class 4 very heavy. These are detailed in *12.3.4.2 Pump wear* and *12.3.5 Hydraulic design and application considerations*.

12.2.9.38 Impeller seal face

A part of the impeller, typically near the suction eye, designed to create a close clearance with a stationary part of the pump, such as the suction liner, to limit internal recirculation.

12.2.9.39 Slurry abrasivity

The tendency of a particular moving slurry to produce abrasive and erosive wear.

12.2.9.40 Abrasive wear

Wear due to hard particles or hard, small surface protrusions forced against and moving along a material surface.

12.2.9.41 Abrasion–corrosion

A synergistic process involving both abrasive wear and corrosion. Each of these processes is affected by the simultaneous action of the other, which may accelerate the overall wear rate.

12.2.9.42 Corrosion

Loss of material created by chemical or electrochemical reaction within the pump environment.

12.2.9.43 Erosion

Progressive loss of material from a material surface due to mechanical interaction between that surface and a fluid, a multi-component fluid, or impinging liquid or material particles.

12.2.9.44 Erosion–corrosion

A loss of material due to both erosion and corrosion, in which each of these processes is affected by the simultaneous action of the other, and in many cases is thereby accelerated.

12.2.9.45 Miller number

A measure of slurry abrasivity as related to the instantaneous rate of mass loss of a standard metal wear block at a specific time on the cumulative abrasion–corrosion time curve as defined in *ASTM Standard G75-01*.

12.2.9.46 SAR number

A measure of the relative abrasion response of any material in any slurry, as related to the instantaneous rate of mass loss of a specimen at a specific time on the cumulative abrasion–corrosion time curve, converted to volume or thickness loss rate as defined in *ASTM Standard G75-01*.

12.2.9.47 Specific energy (E_{sp})

The erosive energy of particles in joules/m^3 (lb-ft/in^3) required to remove a unit volume of the target wear material.

12.2.9.48 Wear coefficient (W_c)

The volume of the target wear material removed for a given unit particle energy in m^3/joules (in^3/lb-ft).

12.3 Design and application

12.3.1 Scope

The purpose of this section is to outline the minimum design requirements for slurry pumps and to provide guidelines for their application.

Slurry pumps are similar to other rotodynamic pumps. Refer to *ANSI/HI Standard 1.1-1.2 Centrifugal Pump Standards*, for general definitions, nomenclature, design application, installation, operation, maintenance, and testing requirements.

Slurry pump design and application differences determine the wear performance and the ability to effectively pump solids. This section concentrates on design and application considerations unique to slurry pumps, and presumes that background knowledge exists relative to rotodynamic pumps in general.

Equipment data sheets, suitable for use by suppliers and users, are referenced in Appendix A and Appendix B.2, reference 25. These sheets should be used to specify the user's duty and equipment needs.

12.3.2 Slurry services

12.3.2.1 Slurry applications

Pumps are used to move mixtures of liquids and solids in many industries. Typical uses for slurry pumps include cleaning, processing, drilling, and transport.

In cleaning applications such as flue gas scrubbers, large volumes of water are used to capture chemicals and some solids in an exhaust stream. These slurries are generally of fairly low concentration and the challenges focus on material selection and shaft sealing.

Fertilizer production processing applications involve solids as a contaminant that must be removed from the final product. These mixtures can contain highly corrosive carriers and higher concentrations of solids complicating material selection and resulting in additional problems, such as settling and plugging pipes.

Drilling applications involve injection of slurries into wells and the removal of earth and rock cuttings generated at the drill face. These slurries are typically not corrosive, but proper material must be selected for erosion resistance.

The purpose of transport service is to move the maximum volume of solids economically. These services include dredging, the movement of ore, rock, oil sands, or matrix into a mine, mill, separation or wash plant; the transport of the product and waste through the plant; and final disposal of the tails and fine waste. In general, pumping slurries can be an economical method to transport solid particles in the volumetric concentration range of 15–40%.

The most suitable transport concentration depends on the characteristics of the slurry and the application in which it is used. Due to the large size and high concentrations of solids, settling, pipe plugging, increased pipe friction, and component wear become major concerns.

Figure 12.15 provides a convenient tool to correlate transport rate and pipe velocity to assist the designer in sizing the pump/piping system to optimum cost-effectiveness. Here, a horizontal marker line from the transport rate should be drawn to the solids specific gravity S_s (normally 2.65) graph line, then vertically to the concentration by volume graph line. Then the marker line should be drawn horizontally to the

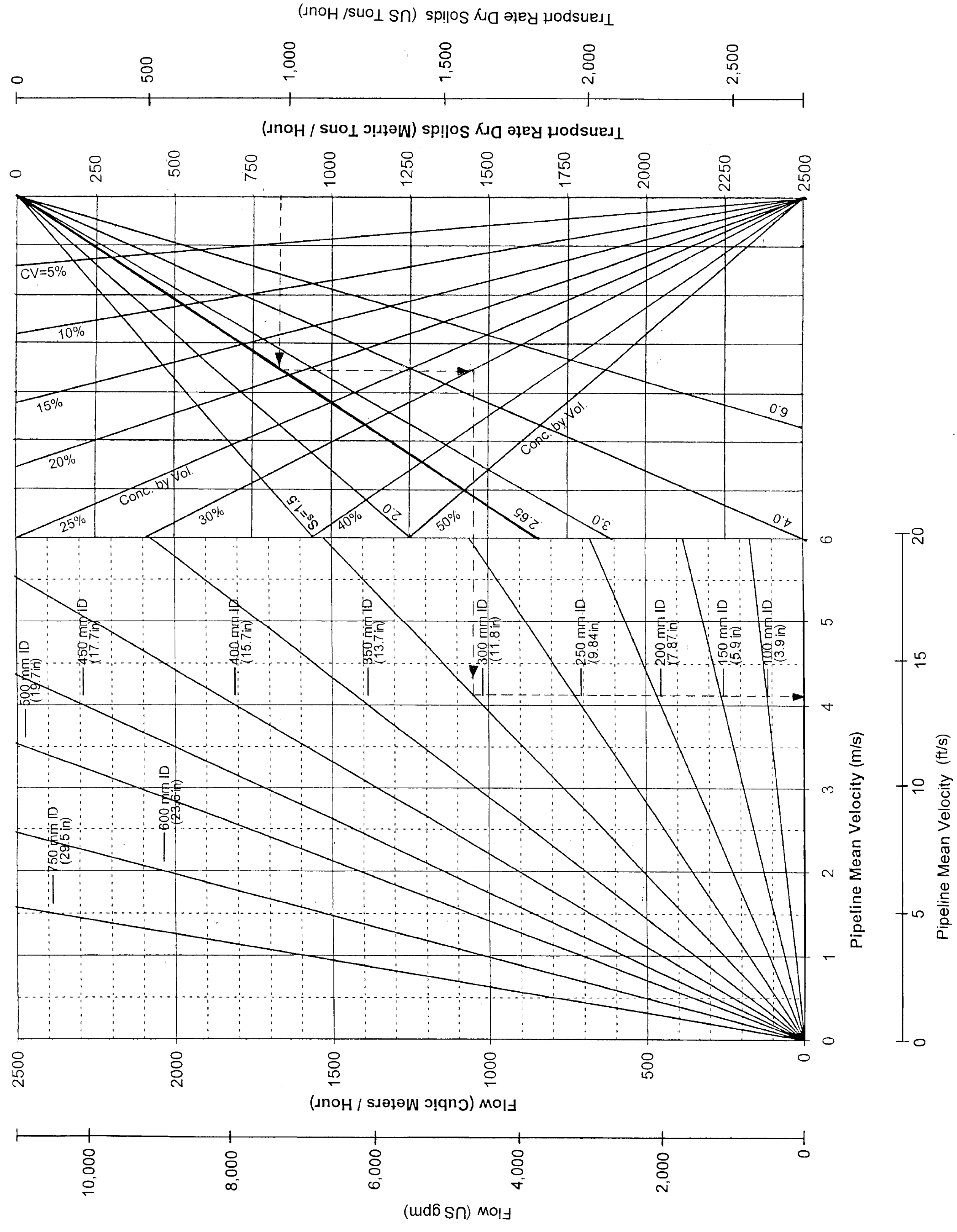


Figure 12.15 — Solids transport rate

appropriate pipe diameter graph line and then vertically, to read the mean pipeline velocity, or horizontally, to get the mean flow rate.

12.3.2.2 Characteristics of slurries

Usually slurry concentrations are discussed in volumetric terms. This alleviates the variables particular to a given slurry. However, within a given industry or field, slurry concentrations are often discussed in terms of the concentration by weight or the specific gravity of the mixture. The relationship between these different measurements is shown in Figure 12.16.

12.3.2.3 Slurry types

Depending mostly on the size of the particles, slurries tend to be classified as settling or non-settling. Non-settling slurries act in a homogeneous manner, but with non-Newtonian characteristics. Settling slurries, depending mostly on the size of the particles, can form a stationary bed and flow in a stratified heterogeneous manner. Depending on the specific gravity and the size of solids, there are also slurries that may be either heterogeneous or homogeneous forming an intermediate type depending on the actual concentration and the presence of any fine-sized clays. Slurries composed of mostly large-sized particles may even move as a sliding bed. Figure 12.17 gives a guide to the slurry types and flow mechanism.

12.3.2.4 Settling slurries

For every settling slurry, there is a deposit velocity (V_s) at which solids will drop out of suspension and form a bed on the bottom of the pipe. A pumping system (pump and piping) must be sized and operated so that the velocity in the pipe exceeds V_s or the pipe will plug. Therefore, a system must be designed for the lowest acceptable value of settling velocity.

V_s is dependent on pipe size, particle size, concentration, and specific gravity of the solids. If pipe size, particle size, and specific gravity are assumed constant, V_s varies with concentration. It is lowest at high concentrations and increases as concentration decreases. The point where V_s reaches a maximum is defined as the maximum value of deposit velocity (V_{sm}) or maximum velocity at limit of stationary deposition. The nomograph in Figure 12.18 can be used to determine V_{sm}. Since concentration cannot usually be controlled, the pumping system should preferably be designed so that the velocity in the piping always exceeds V_{sm} by, at least, a 10% margin.

To obtain a value for V_{sm}, draw a straight line on the left-hand half of Figure 12.18 from the appropriate pipe diameter value over the solids particle diameter line. Where that line intersects, the middle axis gives the value of V_{sm}. In the cases of solids specific gravities other than 2.65, a second line from the center axis V_{sm} value to the right across the relative density value in mind gives a corrected value at the point where that line intersects the far right-hand axis.

Figure 12.18 can also be used to evaluate the sensitivity of the system to changes in particle size. For example, V_{sm} in a 0.3-m (11.8-in.) pipe is virtually unaffected by a variation of particle diameter between 0.4 and 1.0 mm (0.016 and 0.039 in.), however, a reduction in particle diameter to 0.15 mm (0.006 in.) or an increase to 15 mm (0.59 in.) reduces V_{sm} by more than 40%.

If the particle size is not closely controlled, the worst-case particle size (that giving the highest V_{sm}), should be used to determine V_{sm} for design. This will generally be a particle size of 0.4 to 0.6 mm (0.016 to 0.024 in.) depending on pipe size. This yields conservatively high values of V_{sm}, especially in large pipe diameters.

12.3.2.5 Effect of slurry on performance

The performance of a centrifugal pump on slurries will differ from the performance on water, which is the basis for most published curves. Head (H) and rate of flow (Q) will normally decrease as solids size and concentration increases. Power (P) will increase and starting torque may also be affected. This “solids effect” is shown schematically in Figure 12.19 along with the head and efficiency derating terms used.

Net positive suction head required by the pump in order not to exceed 3% head drop (NPSHR) will increase, in most circumstances. Effects of solids on NPSHR are dependent on the slurry type and the pump design and can be highly variable.

For settling slurries of low to medium concentration, a modest increase in NPSHR can be expected. For a particular application this increase can be conservatively estimated by dividing the value of NPSHR on water by the head derating factor discussed below.

For viscous and non-settling slurries or slurries with entrained air, the effect on NPSHR can be significantly greater. The pump manufacturer should be consulted for guidance regarding slurry effects on NPSHR.

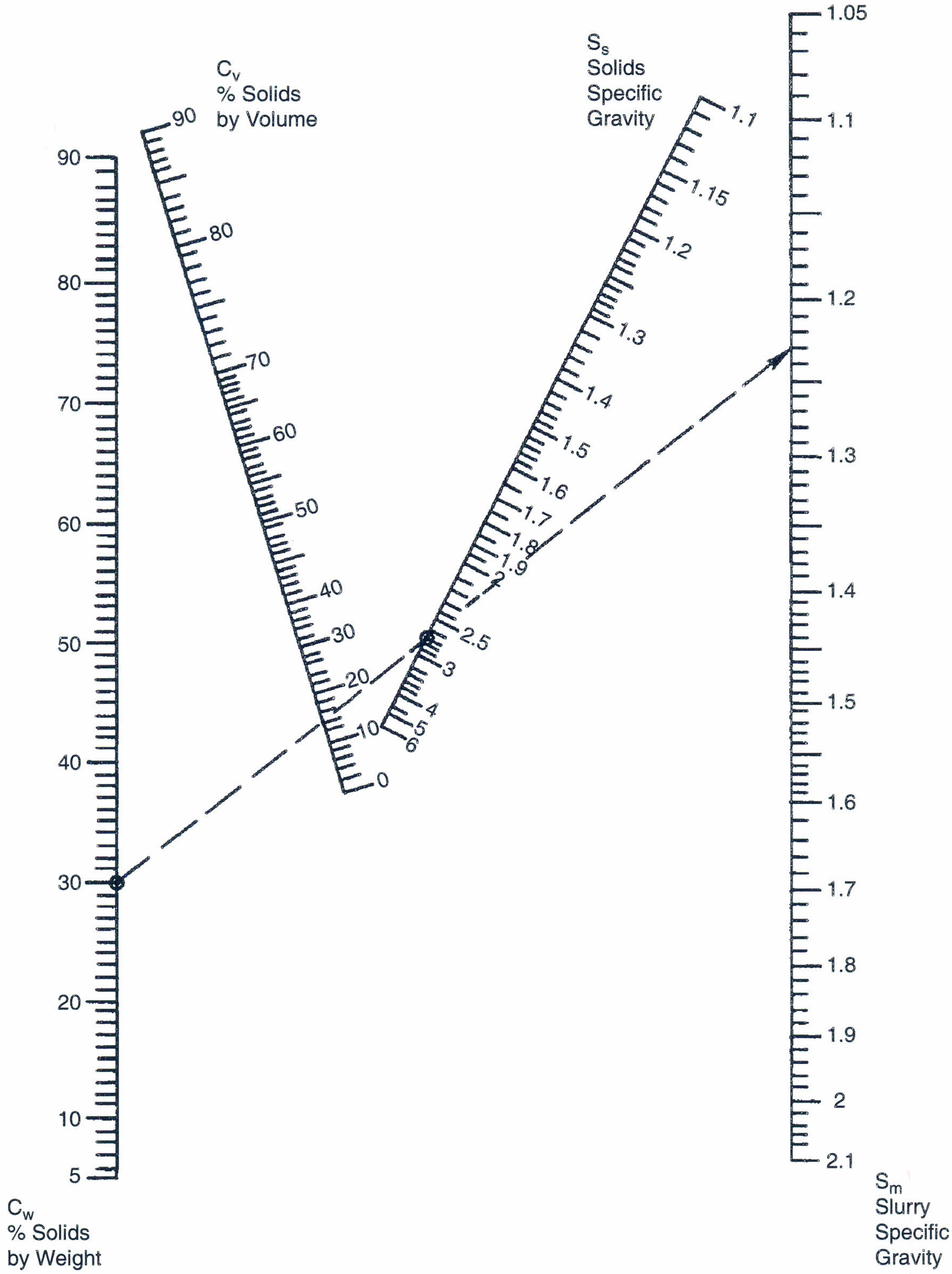


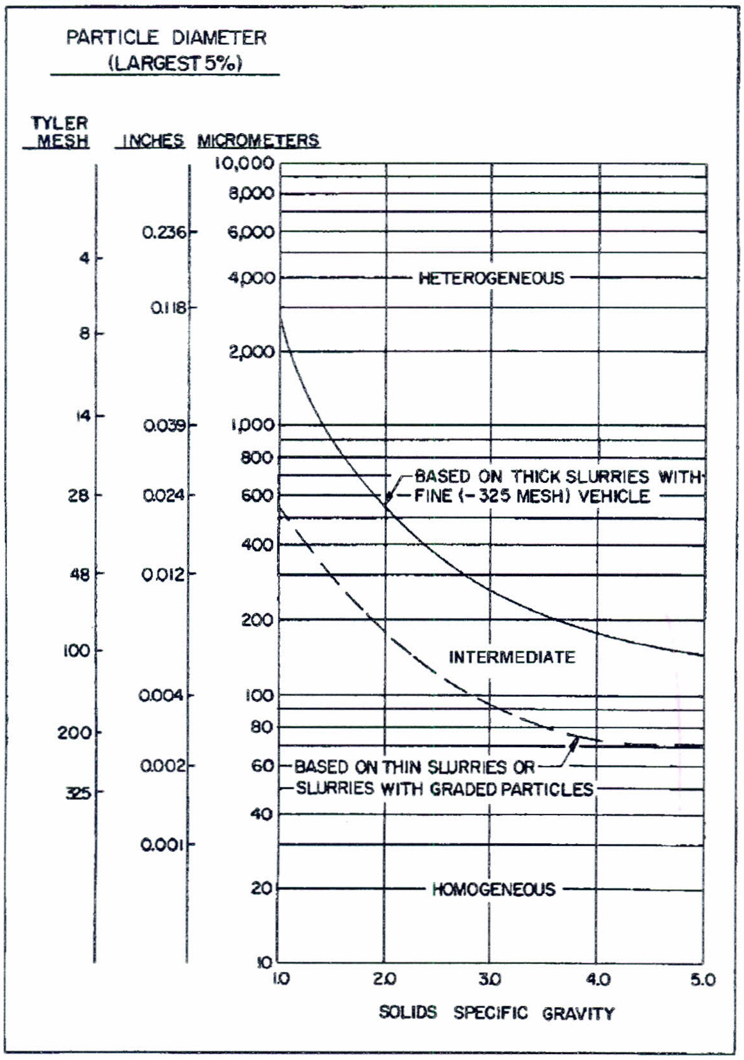
Figure 12.16 — Nomograph for the relationship of concentration to specific gravity in aqueous slurries

Different approaches can be used for predicting the centrifugal pump performance change from water to slurry, dependent on the slurry type.

When the solids-fluid mixture as shown on Figure 12.17 is considered homogeneous and exhibits Newtonian behavior that can be characterized by an apparent viscosity, the ANSI/HI method for pump performance viscosity correction can be applied as outlined in Section 12.3.2.6. The second method is for heterogeneous slurry using the solids size and pump impeller diameter as outlined in Section 12.3.2.7. Pumping of frothy slurries is discussed separately in Section 12.3.3.

These are empirical methods based on the best test data available from sources throughout the world. There are many factors for a particular pump geometry and flow conditions that are not taken into account. However, the methods provide for dependable approximations when limited data on the application is available.

Pump users should consult with pump manufacturers for more accurate predictions of performance for a particular pump and particular slurry.



SLURRY FLOW REGIME (HETEROGENEOUS, HOMOGENEOUS) IS A FUNCTION OF SOLIDS SIZE AND SPECIFIC GRAVITY.

Figure 12.17 — Schematic classification of slurries in industrial pipeline applications

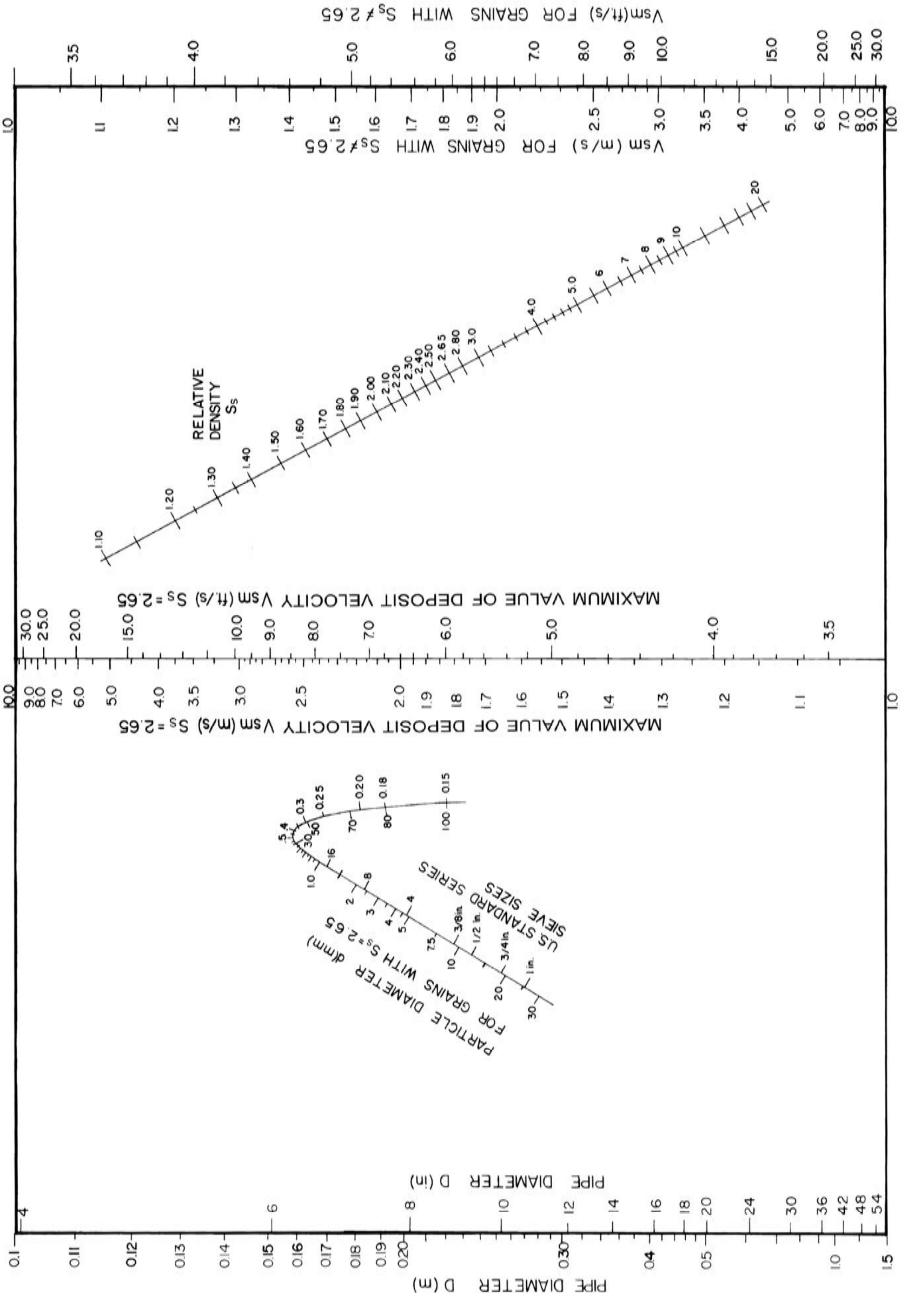


Figure 12.18 — Nomograph for maximum velocity at limit of stationary deposition of solids

12.3.2.6 Performance derating based on apparent viscosity

Standard viscosity correction procedures may be used for non-settling homogeneous slurries provided the apparent viscosity of the slurry is known. Most slurries with high concentrations of very fine particles are, for practical purposes, non-settling. The apparent viscosity of the slurry must be determined, then standard viscosity corrections may be applied to find the head and efficiency derates. (Refer to *ANSI/HI Standard 9.6.7* for viscosity corrections).

Non-settling slurries usually behave as non-Newtonian fluids, so pump performance effects can vary widely. In the reference section simplified ways of expressing the non-Newtonian behavior in the form of a suitable viscosity can be found. With typical industrial slurries, viscosity is generally 50 to 100 times that of water (at 20°C [68°F]) resulting in reductions of 5 to 10% in head and efficiency, dependent on the flow rate and head according to the HI viscosity correction charts. Effects may be greater when operating at flows considerably lower than the BEP flow. This can result in an unstable head curve. The pump manufacturer should be consulted for guidance regarding non-Newtonian effects on pump performance.

Theoretical methods based on loss analysis may provide more accurate predictions of the effects of fluid viscosity on pump performance when the geometry of a particular pump is known in more detail.

12.3.2.7 Performance derating based on solids size and content

Where the slurry is heterogeneous, Figure 12.20 can be used to determine the head and efficiency reductions from the original water performance for different sizes of pumps for a slurry mixture concentration by volume of 15% and with negligible portions of less than 75-micrometer fines.

For solids of S_s other than 2.65, for concentrations other than 15% by volume, and with significant amounts of fines present, the values of R_h shall be modified by multiplying them by the correction factors C_s, C_{fp}, and C_{cv} noted below, applied concurrently.

Specific gravity correction factor (C_s) where $C_s = [(S_s-1)/1.65]^{0.65}$ or using Table 12.4.

Fine-particle correction factor (C_{fp}) where C_{fp} = (1 – fractional content of particles by weight < 75 micrometers)^2.

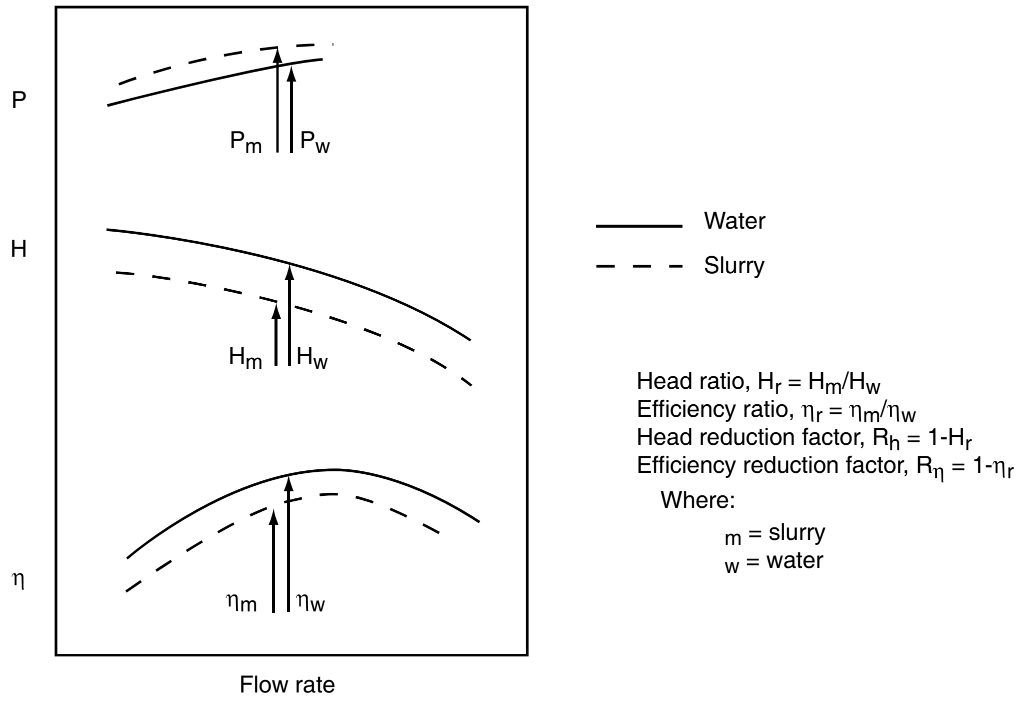


Figure 12.19 — Effect of slurry on pump characteristics (schematic)

Concentration correction factor (C_{cv}) where C_{cv} = (C_v %/15).

For power, it is assumed that the efficiency reduction factor follows the head reduction factor ($R_h = R_\eta$) so that power consumption increases directly with the slurry specific gravity ($P_m = S_m \times P_w$). This assumption is usually conservative on large, heavy-duty slurry pumps, but is adequate to safely size motors. With small pumps and slurries of well over 20% volumetric concentration, the power may be up to 1.5 times larger than the power on water, dependent on the individual properties of the solids.

A pump with a 914-mm (36-in.) diameter impeller that produces 61 m (200 ft) of head at 80% efficiency while pumping water, when pumping a 2.65 solids specific gravity slurry of average particle size of 1 mm (0.040 in.) at a concentration by volume of 15% will, according to Figure 12.20, have a R_h of 8% and an H_r of 0.92. The head produced while pumping slurry will therefore be 61 m × 0.92 = 56.1 m or 184 ft of mixture. $R_\eta = R_h$ = 8%, so η_r = 0.92 and the efficiency produced while pumping slurry will therefore be 80% × 0.92 = 73.6%.

If the concentration by volume was 30%, then the R_h value would double to 16% and the head produced would now be 61 × 0.84 = 51.2 m or 168 ft of mixture.

If 20% by weight of the above slurry is less than 75 micrometers, then the R_h value above would become

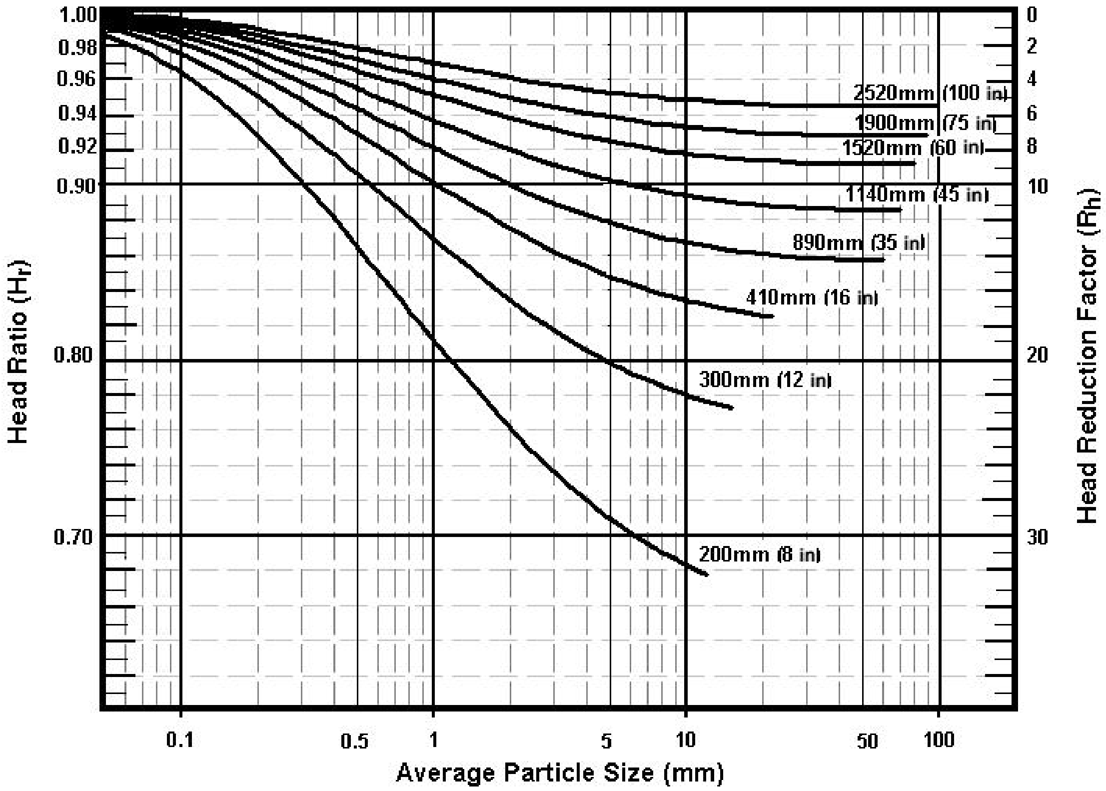


Figure 12.20 — Effect of average particle size and impeller diameter on H_r and R_h (For solids concentration by volume, C_v = 15% with solids S_s = 2.65 and a negligible amount of fine particles. Impeller diameters are given in mm and inches.)

Table 12.4 — Specific gravity correction factor

S_s	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0	3.2	3.4	3.6	3.8	4.0
C_s	0.25	0.40	0.52	0.62	0.72	0.81	0.90	0.98	1.06	1.13	1.21	1.28	1.34	1.41	1.47

$R_h = 16 \times (1 - 0.20)^2 = 10.2$ and the head produced = $61 \times 0.898 = 54.8$ m or 179 ft of mixture.

If the above slurry (at $C_v = 30$ and 20% fines) had a solids specific gravity of 3.0, then the R_h value would be $10.2 \times 1.13 = 11.5$ and the head produced = $61 \times 0.885 = 54$ m or 177 ft of mixture.

It should be noted that slurry mixture characteristics may vary significantly depending on the size of the solids, their shape, solids specific gravity, concentration, and carrier liquid that cannot always be predicted by generalized formulations such as those above. Whenever possible, actual field comparisons should be made and the more sophisticated methods of the different manufacturers, noted in Appendix B, Source material and references, utilized.

12.3.3 Pumping froth

Froth is an aerated liquid medium (slurry) that occurs naturally or is created intentionally. Natural occurrence may be due to the nature of the ore processed in the mineral industries and can create a general nuisance. Froth is created for the purpose of separating minerals, floating the product from the waste or vice versa. Froth is created by the aeration of the slurry through air injection during agitation. Polymers are added to increase the surface tension creating bubbles to which the product or waste adheres. This allows for the separation and collection of the sought after mineral for further refining.

The transfer of froths with slurry pumps is a special-purpose application commonly encountered in the launders of flotation circuits. The very large proportion of air in the froth being handled upsets the normal relationships used to predict pumping performance and requires a unique approach in selecting and applying pumps for this service.

It is well known that the presence of air at the suction inlet will decrease the head, flow, and efficiency of a pump and, with increasing amounts of air, the losses will also increase. The NPSH required increases with increasing air content and at a certain critical level the pump loses prime and stops pumping. Studies done with air injected into the suction of the pump have confirmed this but the data cannot be applied directly to froth pumping.

Depending on the process, type of slurry, or frothing agents used, a certain amount of air or gas will separate from the froth and can lead to problems with pump performance. The change in performance due to the presence of this air or gas could be quantified based on various factors, such as pump geometry, specific speed, and suction pressure. However, it is practically impossible to determine with reasonable accuracy what amount of free air or gas will separate from the froth at the impeller inlet. This problem requires a special approach in selecting a pump to successfully handle the froth application.

The general approach is to oversize the pump for the application by use of a “froth factor.” The froth factor is a multiplier that increases the process design capacity to allow for the increased passing volume caused by the gas in the froth. The factored volume usually causes the pump to be at least one pipe size larger than would normally be selected. This over-sizing helps in handling froth by increasing the impeller eye diameter and by allowing the pump to run at a lower speed due to the larger impeller diameter. There is no sound engineering science to support the technique, but experience has proven that this empirical method works well.

Double suction vertical pumps with semiopen impellers are often applied in this duty. This type of pump is effective because it has over twice the inlet eye area of a conventional pump design and the arrangement allows some air to vent out the top suction inlet. Some installations are better suited to horizontal pumps with open, semiopen, or closed impellers. Open and semiopen impeller designs will handle higher concentrations of air or gas than closed-type impellers, but the closed type can be successfully applied in some services. Horizontal pumps with closed impellers generally require a higher froth factor.

Froth applications normally fall into the low head system range and heads are seldom over 20 m (65 ft). The pumps are typically in the size range of 50 to 200 mm (1.97 to 7.87 in.) discharge diameter and are either elastomer-lined or constructed of hard metals to provide abrasion and corrosion resistance.

Froth collection systems should permit as much gas as possible to escape from the froth before it enters the pump, and some applications will require water sprays to “knock down” the froth to prevent air locking the pump.

The froth factor is normally specified by the pump buyer and is based on previous plant experience. The factors are usually in the range of 1.5 to 4 but can be as high as 8. Many factors influence the size of the froth factor and these may include the viscosity of the liquid, the size of grind of the mineral, and the chemistry used in the process. The type of pump selected will also have an effect on the froth factor used. Some typical vertical pump froth factors for common processes are given in Table 12.5. These are only approximate values; the most reliable factors will come from the users.

Figure 12.21 shows how a froth factor is used to deal with the process design capacity and select a pump for the service. For example, for molybdenum, rougher concentrate a design capacity is given at 182 m^3/h (800 US gpm) and the required head is 10 m (33 ft). A 150-mm (6-in.) discharge size pump would normally be selected for this flow, but this example process has a froth factor of 2.0. The factored capacity will be 364 m^3/h (1600 US gpm). This flow rate is better suited to a 200 mm (8-in.) pump. The original condition point has been plotted on the curve along with the factored condition point. The required head does not change and is simply transferred to the new flow rate. The pump speed will increase from 580 rpm to 640 rpm, as shown in the figure.

The pump drive must be selected to have sufficient power to handle the factored capacity (including froth) at the pipeline (zero froth) specific gravity. This is necessary because there will be times that the system is operated without froth. These conditions will exist at system start-up, after power outages or system shutdowns allow the froth to collapse or deaerate over time. It is important to plan for these upset conditions for a successful installation.

1) Obtain the desired operating condition point and the froth factor from the user.

Table 12.5 — Approximate froth factors

Application	Pump Froth Factor
Copper rougher concentrates	1.5
Copper cleaner concentrates	3.0
Molybdenum rougher concentrates	2.0
Molybdenum concentrates	3.0
Potash	2.0
Iron concentrates	4.0 to 6.0
Coal	6.0

2) Multiply the process design capacity by the froth factor and select an appropriately sized pump for that flow rate.

3) If more than one size impeller is available for the selected pump model, select the largest impeller available to keep the operating rpm at the lowest possible value.

4) Select the pump operating speed at the factored flow rate and the original required head.

5) Size the motor to have enough power to operate at the factored flow rate at the highest expected specific gravity.

The sump design will impact the performance of the pump. Where froth or air entrainment is a minor annoyance, the sump size should be at least equivalent to one minute of flow (capacity) of the pump in question at its design flow to ensure the sump is not too active. Additional approaches can be used for difficult froths using the following arrangements:

a) A gentle rain-like spray to pierce the froth bubbles;

b) A partly submerged box to control the entrance of slurry into a mostly submerged box much smaller than the sump surface area that has no bottom or top to control the air and allow it a chance to escape;

c) Inclined plates to create a circuitous flow path from the slurry entrance point to the suction around and beneath an inclined plate to allow maximum time for the air to escape;

d) A vent on the suction line to allow air/froth to escape;

e) A tall sump to use the compression of the sump static to reduce the size of the air/froth bubble through the relationship of $P_1V_1 = P_2V_2$ or the pressure times the volume of the atmosphere (1) equals the pressure times the volume of the inlet pressure (2) reducing the volume of the air/froth ingested by the pump and;

f) An eccentric reducer installed with the belly side up to allow for a very wide inlet at the sump reducing to the inlet size of the pump to allow for air to escape back to the sump.

12.3.4 Wear in centrifugal pumps

12.3.4.1 Wear considerations

Slurry pumps are usually designed for specific applications. When this involves transporting large solids and/or high concentrations, component wear will be a major factor and must be considered in the selection of the pump and the configuration of the pump installation.

The major slurry erosive mechanisms inside a pump are sliding abrasion and particle impact. Sliding abrasion typically involves a bed of particles bearing against a surface and moving tangentially to it. Impact wear occurs where particles strike the wearing surface at an angle.

Abrasive wear varies with the number of particles or volume concentration of the solids, the velocity of the eroding particles to a power of 2.5 – 3, the abrasivity of the eroding solids, and the wear resistance of the surface being impacted.

The specific energy E_{sp} in J/m^3 (lb-ft/in^3) is defined as the erosive energy of particles required to remove a unit volume of the target wear material. The reciprocal of E_{sp} is referred to as the *wear coefficient* (W_c). The

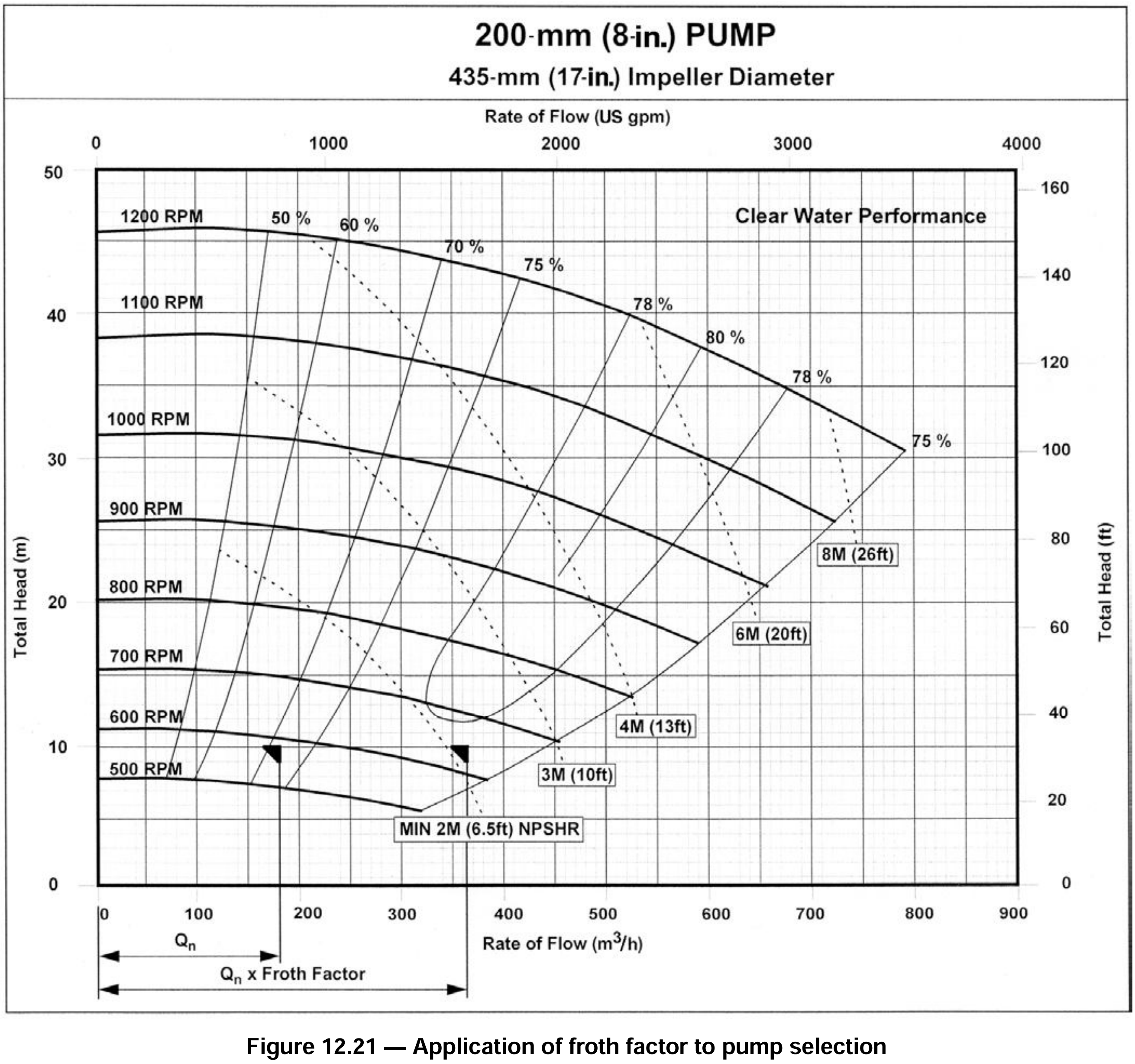


Figure 12.21 — Application of froth factor to pump selection

larger the value of E_{sp}, the lower the expected wear rate for identical slurry flow conditions. A specific energy may be empirically determined for either sliding wear or impact wear.

In computing a sliding wear rate, the friction power of the slurry layer adjacent to the wear surface is estimated from the solid–liquid flow field. The friction power is simply the product of wall shear stress and the velocity tangential to wear surface. A computational fluid dynamics calculation may be essential for determining these in pump components. The friction power (at any location on the wetted surface) divided by the sliding wear specific energy E_{sp} yields the local wear rate at that position. Note that the friction power intrinsically includes the effect of local concentration and particle size.

The wall shear stress may be approximated in terms of the local volumetric concentration of particles, material density of the particles, and the solids tangential velocity. In such a case, the friction power is computed as the product of the local concentration, particle density, the third power of the tangential velocity of the solids, and a multiplicative factor. The multiplicative factor is, for convenience, absorbed into the definition of a modified specific energy E_{sp} (or its reciprocal, W_c). In many practical computations, this simpler approach is found to yield quite reliable results.

In the case of impact wear, the specific energy E_i is a function of the angle of impact α. For cast-iron alloys, normal impact at right angles to the surface gives the lowest E_i (highest wear rate). A number of impact wear specimens with different angles of the wear test wedge pieces are tested to characterize E_i as a function of α. Similar to the friction power, the impact power carried by the particles is proportional to the particle density, concentration, and approximately the cube of the particle velocity. Dividing the impact power by E_i (for the specific angle of impact), the impact wear rate is computed.

Theoretically, for a given slurry/wear surface combination, E_{sp} and E_i (α) are expected to be constant. Experiments indicate that particle size plays an important role in determining E_{sp} and E_i (α).

Figure 12.22 shows examples for different sand particle sizes against different resisting materials for sliding abrasion in a neutral pH medium. The sliding wear coefficient (W_c) values shown in Figure 12.22 are used on the modified definition (including the approximation to the wall shear stress and the attendant multiplicative factor) as outlined in the foregoing. The values shown in Figure 12.22 will vary with different types of solids, hardness, solids specific gravity, and sharpness of particles. The abrasivity of a particular slurry will be used as a measure of this difference.

The *ASTM Standard G75-01* presents details of characterizing the abrasivity of slurries using the Miller test. The Miller number helps rank the abrasivity of the slurries in terms of the wear of a standard reference material. The higher the Miller number, the greater is the wear on the standard Miller test specimen, and hence the greater is the slurry abrasivity. The Miller test apparatus consists of a reciprocating arm with the wear specimen attached to it. The wear specimen thus reciprocates in the slurry of known concentration. The test is run in three two-hour stages with different orientations of the wear specimens. In the typical test, two wear samples reciprocating in separate slurry trays (made of plastic) are used to ensure reliable test data. The cumulative mass loss on each wear sample is recorded after each two-hour duration of testing. The data points are fitted to a power-law curve to obtain the erosion rate.

In practice, slurries of 50% concentration (by weight) are used. It is found that slurries of higher concentration yield essentially the same Miller number. Apart from slurry concentration, the hardness, size, and shape of particles are important. In addition, the corrosivity of the slurry can significantly affect the Miller test.

The effect of corrosion can be isolated by neutralizing the slurry carrier, rerunning the test, and comparing the results. If the Miller number drops significantly, the corrosion effect is dominant. The pump materials may then be appropriately chosen to minimize corrosion.

Slurries with a Miller number of 50 or lower can usually be pumped with minor erosive damage to the pumping system. Such slurries may be classified as “light duty.”

For impact wear, the erosion rate varies for different materials and solids impact angles. Figure 12.23 below shows how ductile (elastomer) and brittle (white iron) materials respond.

It is possible to model the slurry particle velocities within the main components and, using the wear coefficients noted, calculate the wear. Modeled values for wear are for specific geometries and conditions. The uncertainties associated with the modeling are such that it is not suitable for use in a standard.

The foregoing is provided as an overview of the factors associated with slurry pump internal component wear inside of pumps and an insight into the relative material and other effects. For slurry pump selection, a wear service class method along with various limiting velocities (described later) is used in this standard, in conjunction with field experience with specific slurries.

Slurries may be corrosive in chemical composition, which creates an "erosion–corrosion" wear condition. This can be much more aggressive than either erosion or corrosion, so standard wear predictions become highly uncertain. Proper application and material selection (refer to Section 12.3.7) are needed to maximize life. This can involve trial-and-error evaluation due to the large variety of slurry solids and fluid chemicals being pumped.

12.3.4.2 Pump wear

Pump wear depends on the pump design, the abrasivity of the slurry, the specifics of the application or duty conditions, the way in which the pump is applied or selected for the duty, and the actual conditions of service.

Wear inside the pump varies significantly depending on the velocity, concentration, and impact angle of the particles. It is normally most severe in the impeller seal face area of the suction liner, followed by the vane inlet and exit. The casing wear amount and location also varies with the shape of the collector, and as a percentage of the actual operating conditions compared to the best efficiency point flow.

Many slurry pump wear parts may last for years with only routine maintenance. Services such as transportation of high concentrations and very abrasive or large solids can sometimes reduce part life to several months. Larger pumps with thicker sections, more wear material, and slower operating speeds can improve life in all applications, although the significant associated product cost increase may not be warranted in each particular case. There are analytical

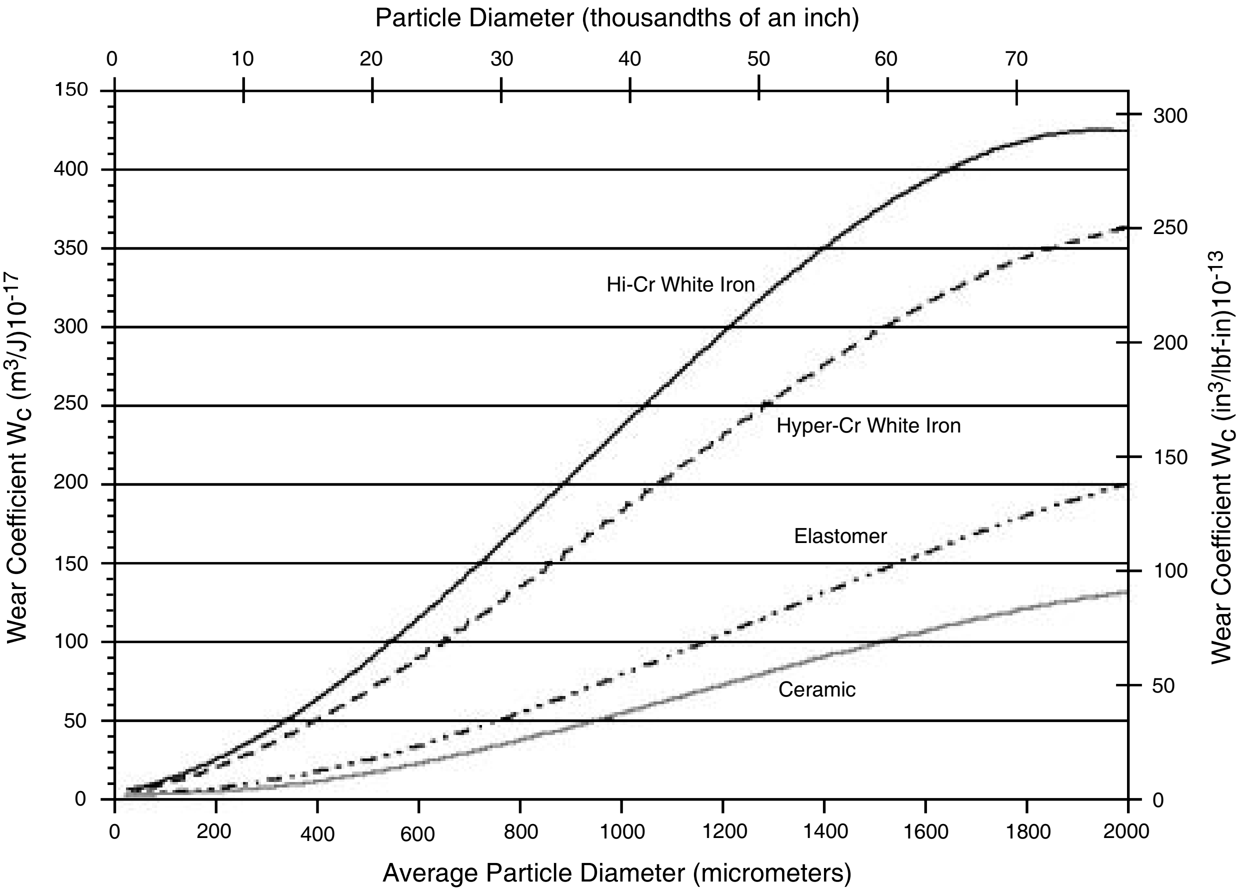


Figure 12.22 — Wear coefficient W_c for different resisting materials in a neutral pH media for different average-sized abrading particles

and numerical models for making qualitative predictions of wear. Their limitations and the variability of slurry service are such that wetted component life prediction is still only good for estimation and should not be used for guarantees. These estimates are normally based on the specified operating condition of the pump and may vary greatly if the pump is operated at significantly different conditions. Using such an analysis, a life-cycle cost (LCC) evaluation of the capital, power, wear, and other costs associated with the pump operation can be used to estimate the best balance between different pump designs. Such analysis is largely theoretical, however, as wear can be unpredictable in actual service.

Ranking the slurry into light (class 1), medium (class 2), heavy (class 3), and very heavy (class 4) services, as shown in Figure 12.24, provides a practical tool for pump selection. Application limitations specific to each service class are found in Section 12.3.6.

Figure 12.24 is based on aqueous slurries of silica-based solids pumping (S_s = 2.65). It can also be used to provide guidance for mineral slurries if an equivalent specific gravity for the mineral slurry is used to determine the service class.

The chart is used by drawing a horizontal line for the concentration by volume of solids (or specific gravity when the solids specific gravity = 2.65), and a vertical line for the average size of the solids. Where the lines intersect determines the class of service. The equivalent specific gravity of the mineral slurry is computed by multiplying the actual slurry specific gravity by the ratio of the Miller number for the mineral slurry to the Miller number of the equivalent silica slurry. This simplified approach can be adjusted based on field experience.

12.3.5 Hydraulic design and application considerations

Slurries can be very abrasive and contain large particles making wear life and the ability to pass these solids major considerations in the design and application of slurry pumps. Compromises in design must be

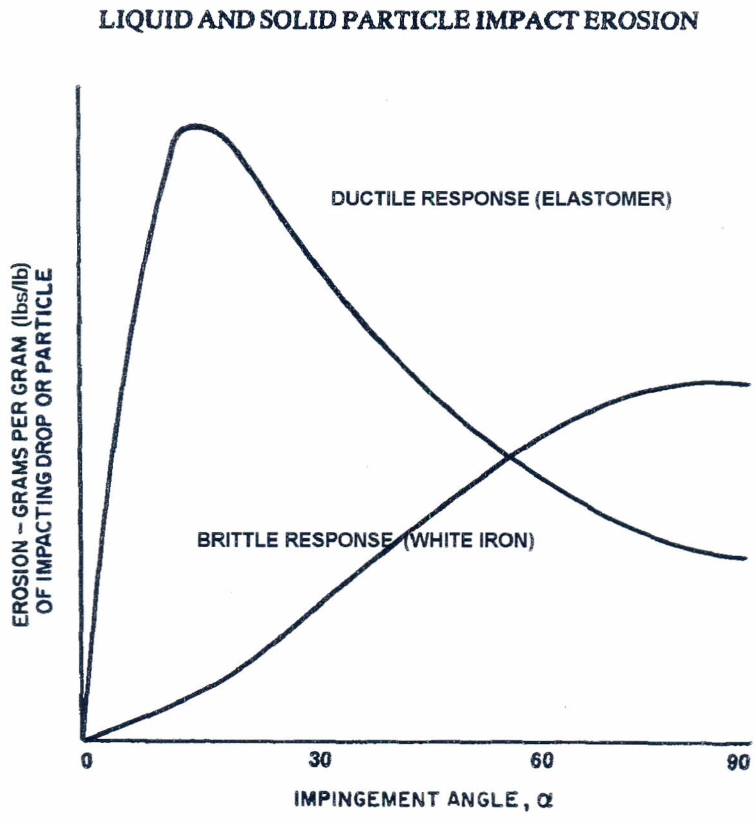


Figure 12.23 — Erosion response for different impingement angles and materials

made to provide large hydraulic passages, and obtain satisfactory wear life in these erosive conditions. As a result, slurry pumps tend to be larger, have flatter head capacity curves, and require more power than their clear-liquid counterparts.

The pump designer maximizes part life by avoiding sharp edges and adding more material to key wear locations. Blunt edges do not wear as fast as sharp ones and more material can be worn from a thicker section before it becomes unusable. The presence of large solids will require wider-than-normal impellers to pass the solids and fewer and thicker vanes to withstand impact loads. This type of design usually compromises optimum hydraulic geometry to some degree, depending on the size of the pump and the intended service. Thus efficiencies of slurry pumps are lower than that of clear-liquid pumps, but wear life is greatly increased. Extra material is needed to combat wear as the size and severity of service increase, so large class 4 pumps will have a casing thickness up to four times greater than that would be used for water pumps. Dimensional compromises to accommodate large solids are usually more pronounced on smaller pumps. As a result, small slurry pumps on class 1 services with solids no larger than 50 micrometers (0.002 in.) will approximate the size and performance of water pumps, but small pumps handling large solids and large pumps in severe services will be larger and heavier and performance will be significantly affected. The pump supplier must be advised of the properties of the slurry, including the maximum size solids, so the proper design can be furnished.

Wear must also be controlled by proper pump application. Wear is related to the relative velocity between the pumped liquid and the pump parts. Liquid velocities must be reduced for more severe services to obtain satisfactory life. The different suction liner, impeller, and casing components and their wetted surfaces are exposed to different velocities, slurry concentrations, and impingement angles making it difficult to provide limits that cover all cases and all components.

Table 12.6 provides recommended service limitations for different service classes that, when coupled with proper design and material selection, has resulted in acceptable wear life.

In Class 4 service high rotational speed with large pump suction diameters over 750 mm (30 in.) will lead to high wear in the impeller front sealing area, while life of the suction liner can be three months or less.

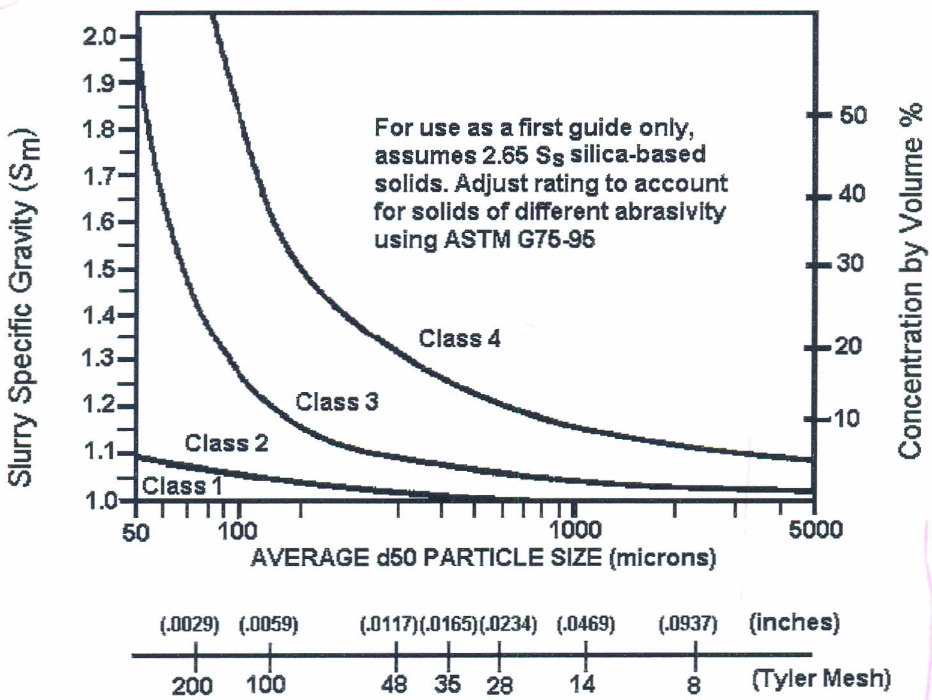


Figure 12.24 — Service class chart for slurry pump erosive wear

Where downtime or other considerations require longer life, the impeller peripheral speeds should be lowered, noting that a 20% drop in pump rotational speed will roughly double the wear life.

Even the largest, most robust pump running at slow speeds will still have increased wear and gouging. Where the pump is operated relative to its best efficiency design point is extremely important. Acceptable gouge-free wear depends on the radial shape of the discharge casing, its width, and the actual percentage of BEP flow rates experienced during operation.

Here an annular-shaped casing is one where the radial distance above the impeller remains constant, while the near volute radial distance increases (approximately) linearly from a cutwater in an angular manner around the periphery of the impeller. The semivolute lies between the two. These are the most commonly accepted types.

While it should be noted that the width and the actual shape of the casing cross sections can modify the wear and its location around the casing, Table 12.7 shows the generally acceptable range of flow rates for the different designs.

12.3.6 System design

The characteristics of slurries require special considerations in the design of pumping systems. In general, flow velocity must be kept within an optimal range. Higher velocities result in high energy requirements and wear. Lower velocities cause instability and plugging of pipes. It is especially important in a slurry pipeline to keep velocities low and at the same time above a certain minimum. Figure 12.25 gives a generalized depiction of slurry effects in piping systems.

System head and velocity requirements for non-settling slurries can be treated very similarly to clear viscous liquids. Systems should be designed to operate near the transition point (refer to Figure 12.25) to obtain energy-efficient and stable operation.

Table 12.6 — Suggested maximum operating values for acceptable wear

	Service class			
	1	2	3	4
Maximum head per stage: meter	123	66	52	40
feet	400	225	168	130
Maximum impeller peripheral speed: All-metal pump (m/s)	43	38	33	28
(ft/min)	8500	7500	6500	5500
Rubber-lined pump (m/s)	31	28	26	23
(ft/min)	6000	5500	5000	4500

Table 12.7 — Optimum wear relative to BEP flow rate for slurry pump casings

Operating Limits	Casing Type	Service Class			
		1	2	3	4
Recommended percent range of BEP flow rate	Annular	20 – 120%	30 - 110%	40 - 100%	50 - 90%
	Semivolute	30 – 130%	40 - 120%	50 - 110%	60 - 100%
	Near volute	50 – 140%	60 - 130%	70 - 120%	80 - 110%

System head and velocity requirements for settling slurries can be determined by similar means, but the deposit velocity (Figure 12.18) must also be taken into account to avoid plugging pipes. For safe operation, the flow velocity should be the larger of the minimum head loss velocity (Figure 12.25) or 110% of the V_{sm}.

Information is available in references to calculate the friction loss of slurry flows. Different manufacturer's methods and experience are also valuable sources of knowledge. They must be used with care as the friction loss can be significant and vary drastically for different slurries so it is necessary to know the characteristics of the specific slurry being pumped. Actual experience with the specific slurry being pumped is the best source of information. If the slurry concentration by volume is over 5% and the piping system is more than a few hundred meters, published literature may not be adequate. In such cases, tests on the actual slurry should be considered if experience is insufficient.

There will always be some inaccuracy in calculations due to the variations in slurry, piping geometry, etc. that will occur. These small variations can have a significant effect on system head requirements; therefore, provisions should be designed into the system to adjust the pump output to match the actual system needs. This can be accomplished by changing the pump speed using a variable-frequency drive, sheave changes, or by changing the impeller diameter. In either case, excess driver power must be available to accommodate the new pump requirements.

12.3.7 Wetted materials of construction

12.3.7.1 General usage selections

A large variety of metals and elastomers are used for slurry pumps due to the diverse range of applications. Slurries can be erosive, corrosive, or erosive/corrosive. Proper material selection depends on the properties of the mixture to be pumped and the pump design. Figure 12.24, introduced earlier, may be used to rank purely erosive wear. Table 12.8 is a selection guide for various materials commonly used in these services along with their appropriate erosive wear service class. Appendix C.2 lists the various international standards for materials typically used in slurry pump construction.

Metals resist erosion through a combination of proper hardness and toughness. Toughness is defined as the ability of a material to absorb energy and even deform plastically before fracturing. Hardness provides

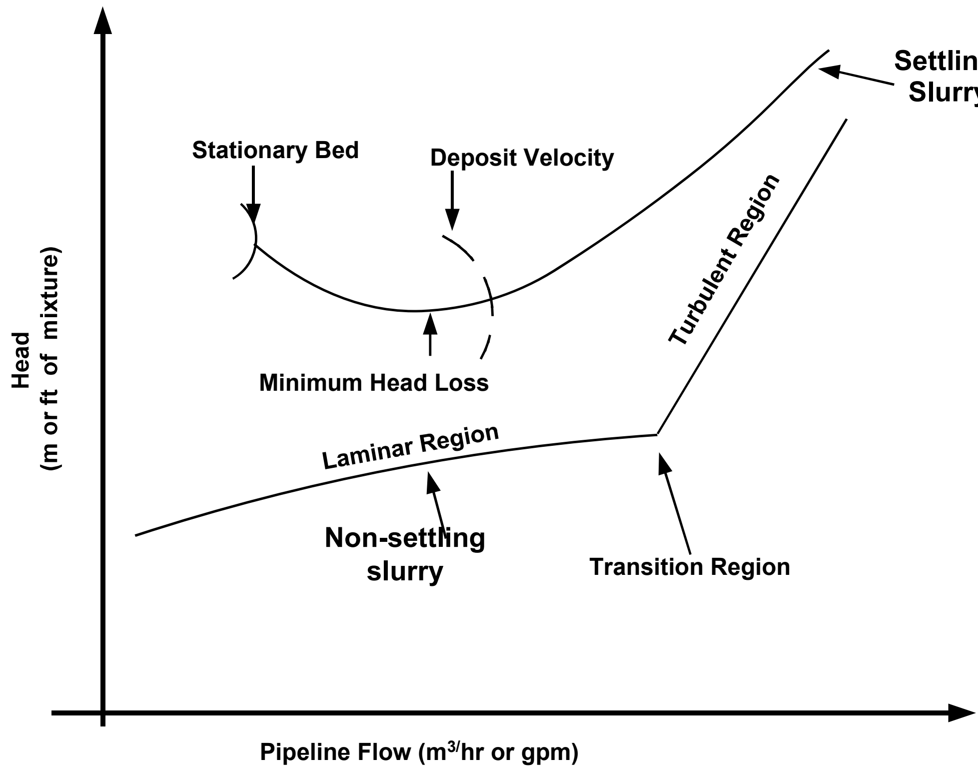


Figure 12.25 — Typical constant concentration slurry pipeline friction loss characteristics

resistance to sliding wear. Toughness diminishes crack formation and propagation encountered in impacting wear situations, providing resistance to impact fracture. Harder materials are better choices for sliding wear services. A very hard, brittle material that fractures easily may not perform as well as a softer metal that resists brittle fracture. Erosion resistance should not be judged only on the hardness of the material.

Material selection is further complicated when corrosive carrier liquids are involved. Materials that are highly resistant to erosion are usually not highly resistant to aggressive corrosion. Material selection is a compromise between erosion and corrosion resistance properties to achieve optimum wear life for any specific installation.

Metals resist corrosion by forming a passivated surface layer that protects against further corrosion. Effectiveness is determined by how tough the passive layer is and how fast it forms. In slurry services, the passive layer is continually being worn away and reformed so corrosive attack is accelerated.

Elastomers resist erosion through resilience and tear resistance. These are soft and the solid particles rebound without damaging the elastomer by abrasion or fracture. Large or sharp particles may tear the elastomer, so material selection must be carefully matched to the slurry.

Elastomers do not depend on a passivated layer to resist corrosion. The basic chemical resistance is a function of proper material selection, and is not significantly changed by exposure to erosive environments. Slurry pumps usually have thicker liners than other elastomer-lined pumps. Experience has shown that when lining thickness is increased, wear life increases by a factor of approximately two to one, within the limits of a practical liner thickness.

Elastomers can be easily bonded to metals to combine the strength and rigidity of the metal with the elasticity of the elastomer. They can also be bonded to materials such as fiberglass reinforced plastic (FRP) or thermosetting phenolic/nylon cloth to stiffen liners to prevent collapsing during process disruptions such as cavitation or surges. They can be bonded to ceramics to take advantage of the best of both materials. Process and environmental temperatures must be considered, as elastomers do not perform well above 80°C (180°F).

Erosive and erosive/corrosive wear may occur under different mechanisms. Because of the complex nature, the wear results may vary substantially from case to case. Experience with similar applications is always the best guide to selecting materials. If there is inadequate experience, wear testing can be performed to help evaluate the level and characteristics of wear factors. Some typical wet wear tests include the *ASTM G75-01* Miller procedure, slurry-jet wear testing, and Coriolis wear testing. Corrosion, erosion, and corrosion/erosion testing may be required to analyze erosive/corrosive applications where experience is not available.

12.3.7.2 Irons

Gray cast iron is relatively soft and brittle so it has limited application in slurry services. It is normally only used on very low concentrations of fine soft particles in a noncorrosive carrier, or for clear-water service.

Table 12.8 — General suitability of wetted materials

Wetted Material	Abrasive characteristics of pumpage	Applicable wear service class	Corrosive characteristics of pumpage
Gray cast iron	Very mild, fine particles	1	Noncorrosive
Ductile iron	Moderate	2	Noncorrosive
White irons	Severe	4	Mildly corrosive
Martensitic stainless steel	Moderate	3	Mildly corrosive
Austenitic stainless steel	Mild	1	Highly corrosive
Duplex stainless steel	Moderate	2	Corrosive
Elastomers	Severe, fine particles	3	Mildly corrosive

Ductile iron is significantly tougher than gray or other irons due to its microstructure of nodular graphite in an iron matrix. Elongation can be up to 18%. This makes it suitable for pumping large particles where impact can be a problem. It is still only moderately abrasion resistant and is limited to moderate concentrations in noncorrosive services.

Hard irons are used for the most erosive slurries and those with weak acid or caustic carriers. These are classified as chromium-nickel (NiHard), chromium-molybdenum, and high-chromium white irons. These irons can normally be used for slurries with pH values between 3.5 and 10.0, depending on the chloride level. At zero chlorides, for example, a pH of 4.5 is satisfactory, whereas at 20,000 ppm chlorides, the pH is limited to 6.5.

These irons gain erosion and corrosion resistance primarily by the addition of chromium, which promotes the formation of chromium carbides in a softer matrix of ferrite, martensite, or austenite. These chromium carbides are three to four times harder than the matrix and provide excellent wear resistance, while the softer matrix maintains strength and some ductility. Higher chromium levels generally increase the corrosion resistance, making them more suitable for solutions with pH farther from a neutral.

Machining hard irons is relatively difficult and grinding will be required in some cases to obtain desired dimensions. One of the advantages of a hard chrome iron is that it can be annealed, machined, and rehardened.

12.3.7.3 Stainless steels

Stainless steels must be used for more severe corrosive applications. Their erosion resistance is much less than the hard irons, but is offset by increased corrosion resistance. Martensitic, ferritic, austenitic, and duplex grades can be used, depending on the application. Corrosion-resistant properties are achieved by large additions of chromium, nickel, molybdenum, and copper.

Martensitic stainless steels (400 series ASTM A487 CA15) are used for mildly corrosive applications. Martensite is quite hard, but is not highly corrosion resistant. These are the most wear-resistant stainless steels. They are suitable for moderately abrasive slurries, but are limited to relatively mild corrosive services.

Austenitic stainless steels and other high-nickel alloys (300 series ASTM A744 CN7MCu) are used for highly corrosive applications. Austenite granules are very tough and corrosion resistant, but the matrix is quite soft. These are the softest steels commonly used in slurry services. They are very corrosion resistant, but are limited to very light slurry applications.

Duplex stainless steels (ASTM A744 CD4MCu) are two-phase alloys, which contain both ferrite and austenite in the microstructure. This provides better corrosion resistance than martensitic steels and better erosion resistance than austenitic steels. These steels are used for light slurries with aggressive carrier liquids.

12.3.7.4 Elastomers

The most commonly used polymer is natural rubber (NR). The common form is “pure gum” natural rubber, which is usually defined by having a specific gravity of < 1.0, a hardness of approximately 40 Shore A, and very high resilience. This high resilience gives maximum abrasion resistance, providing the slurry particles are not too large (< 6 mm [0.24 in.] for impellers) or too sharp, causing excessive cutting and tearing. Natural rubber components are chemically resistant to most slurries, which are mildly acidic or basic, and at temperatures less than about 80°C (180°F).

Natural rubber can be compounded with fillers such as carbon black of extremely fine carbon particles and/or silica to increase hardness and stiffness. This increases resistance of rubber to cutting and tearing and allows it to handle larger particles (< 13 mm [0.50 in.]) and higher tip speeds. Resilience is decreased, so abrasion resistance is decreased.

The following nonnatural rubber elastomers are used, mainly for improved chemical and/or heat resistance.

Polychloroprene (CR) is commonly known as *neoprene*. It is used for increased heat resistance (up to approximately 100°C [212°F]), resistance to certain hot acids, and moderate oil resistance. It is not as abrasion resistant as natural rubber, but is better than most others.

Polyurethane is often used for fine-particle slurries (Minus 65 mesh), as it is much harder than other elastomers, with the same resilience, so increased impeller tip speeds are possible. Oil and solvent resistance is good. Care must be taken to select the proper type of urethane to prevent problems with hydrolysis in hot (80°C [180°F] maximum) service.

Butyl (IIR), chlorobutyl (CIIR), or bromobutyl (BIIR) are sometimes used for hot acid service but abrasion resistance is generally poor. Ethylene-propylene-diene-monomer (EPDM) has much the same chemical resistances as the butyl's, generally better heat resistance, and considerably better abrasion resistance. It has often replaced butyl in hot acid service.

Nitrile (NBR) is used where maximum resistance to nonpolar oils (total petroleum hydrocarbon oils that do not have a charge at the end of the molecule) and solvents is required. Abrasion resistance is only fair. Carboxylated nitrile (XNBR) has better abrasion resistance. Hydrogenated nitrile (HNBR) has better abrasion resistance along with much better heat resistance. It is also resistant to hot water (up to approximately 175°C [347°F]). It is extremely expensive, so use is usually restricted to small parts.

12.3.8 General arrangement details

12.3.8.1 Impellers

Both semi-open and closed impellers are used in slurry services. The control of leakage back into suction is usually accomplished with a combination of clearing or expelling vanes on the impeller and close axial clearances. Because these axial clearances increase with wear, pumps should be arranged to allow simple clearance adjustments to maintain performance. Close radial clearances wear quickly when solids are present and cannot be conveniently corrected with external adjustment, and should only be used on low concentrations of fine slurries. An axial clearance arrangement between the impeller inlet diameter and liner is common for providing leakage control for high-wear services.

Impeller attachment methods vary by manufacturer and service requirements. Various bolted designs and threaded designs are used successfully. When pumping highly abrasive slurries the impeller attachment should be protected from wear to optimize service life. An internally threaded impeller is typically used in high-wear services to provide this protection.

Balancing requirements for slurry pump impellers are different than those applied to impellers for clear liquids. An impeller balanced for clear-liquid service is expected to remain substantially in balance for most of its operating life. As a slurry pump impeller wears in service, it will naturally begin to change its balance due to the erosion of metal along the wear surfaces. Consequently, the bearings and shafts in a slurry pump must be designed for a large amount of unbalance in the impeller. In general, slurry pump impellers will be balanced to a lesser standard (higher residual unbalance) than a clear-liquid impeller. The levels of residual unbalance allowed are determined by the manufacturer and are based on a number of operational and design factors. As a rule of thumb, slurry pump impeller balance requirements will fall between ISO 1940/1 G40 on the high (large amount of residual unbalance) side and ISO 1940/1 G6.3 on the low (small amount of residual unbalance) side.

Impellers having a diameter to width ratio of > 6 may be statically balanced in a single plane, using balance rails, on a roller shaft arrangement or by using a commercial balancing machine and making a single-plane correction.

12.3.8.2 Bearings

Antifriction ball or roller bearings are used on most pumps. Hydrodynamic bearings may be used on some large units, such as dredge pumps. Bearings may be grease or oil lubricated. Bearing housings must be effectively sealed from leakage and outside contamination. Labyrinth seals, bearing isolators, lip seals (Figure 12.26), and other proprietary seals are commonly used.

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

Labyrinth seals, Figure 12.27, consist of a simple gap seal with labyrinth grooves and possibly a gravity drain to augment performance. Labyrinth grooves provide for a means of retaining splash oil lubrication, but they rely on a simple gap for contaminant exclusion.

Bearing isolators, Figure 12.28, are composed of both a stationary and a rotating component that act in concert to retain lubricant and exclude contaminants from the bearing housing.

Bearings should be sized for the calculated fatigue life that corresponds to the slurry service class shown in Table 12.9. Calculations should be done at the worst acceptable operating point, which in most cases is minimum flow. More severe services require a longer calculated bearing life due to the impact of large solids, possible cavitation, and variable loads. It may be necessary to increase shaft and housing size to accommodate the correct bearings for a given application.

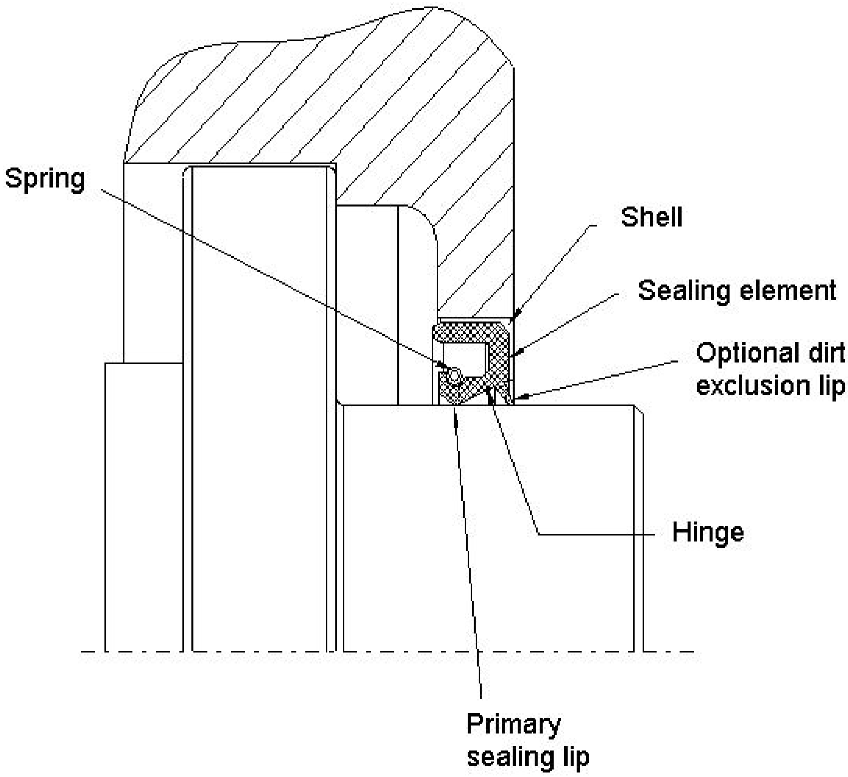


Figure 12.26 — Typical lip seal and its components

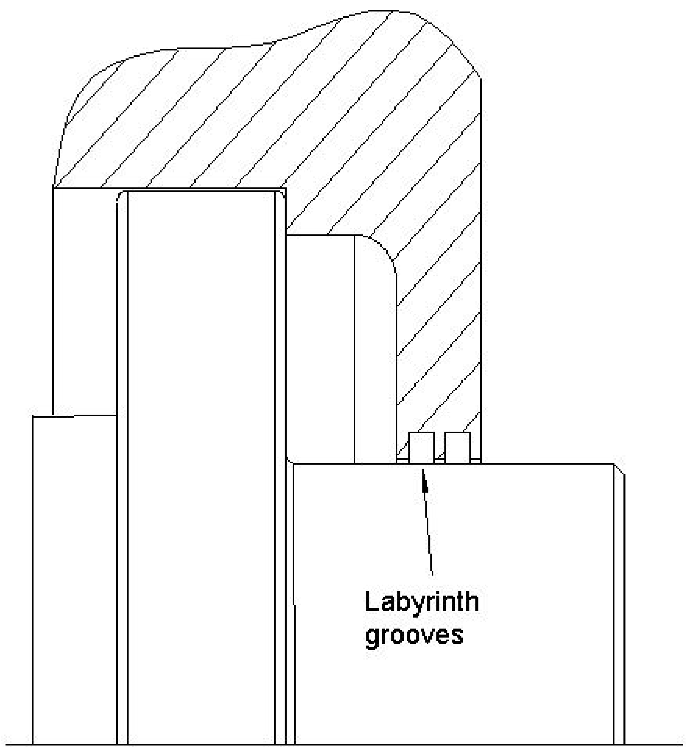


Figure 12.27 — Typical labyrinth seal

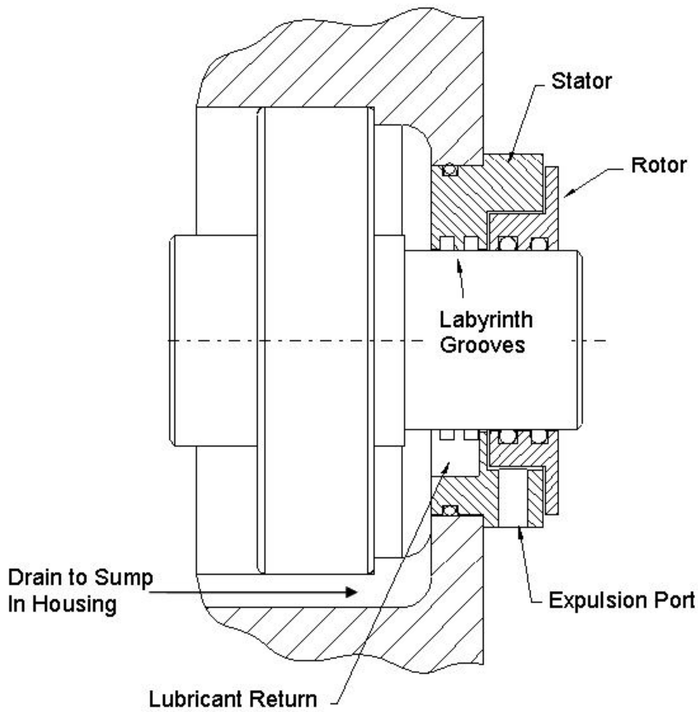


Figure 12.28 — Generic bearing isolator and its major components

Table 12.9 — Calculated fatigue life of bearings by slurry service class

Slurry service class (Ref. Figure 12.24)	Minimum calculated bearing fatigue life (L_{10} life in hrs)
1	17,500
2	35,000
3	50,000
4	50,000 (NOTE 1)

NOTE 1 For large pumps for class 4 service, life should be increased 2000 hours for every inch (25.4 mm) of suction diameter over 12 in. (305 mm).

12.3.8.3 Seals and packing

Sealing slurries around the pump shaft can be very difficult. Manufacturers have developed systems with packing, dynamic seals (expellers), or mechanical seals. These involve adaptations of available products as well as proprietary designs. The user should review the proposed shaft seal with the manufacturer to be sure that its service life and cost-effectiveness are suitable for their service and maintenance practices.

12.3.8.3.1 Compression-type packing

This is the most common method of sealing slurry pumps. It has the advantages of the lowest initial cost and ease of maintenance. The main disadvantages are that periodic attention and (except for the dynamic type) a clean supply of water, which dilutes the product, are required. Severe duty transport service often involves shaft deflections that also exceed the limits of mechanical shaft seals and can exhibit shock loading that may damage typical hard face seal materials. Packing can dampen out shock loads, acting to some extent as an extra bearing, although this might shorten the life of packing.

There are three basic compression packing arrangements. The packing design selection is based on duty conditions, process fluid requirements, and the availability of sealing water.

The "flush-type" arrangement, Figure 12.29, positions the lantern ring in front of the packing rings and a clean liquid is injected at a higher pressure than pump discharge pressure to prevent the slurry from reaching the packing rings. This arrangement is recommended for severely abrasive applications wear service classes 3 and 4 (Ref. Figure 12.24) that can tolerate considerable process fluid dilution.

The “weep-type” arrangement, Figure 12.30, restricts flow into the pump by positioning the lantern ring between the packing rings with a clean-liquid injection at a higher pressure than pump discharge. Product dilution is greatly reduced with this arrangement, but the slurry can penetrate the packing and wear the packing sleeve or shaft, if the packing has not been carefully adjusted. This type of arrangement is recommended for wear service classes 2 and 3.

The “dry-type” packing, Figure 12.31, is used with a primary dynamic seal or expeller when product dilution must be minimized. The dynamic seal prevents abrasives from reaching the stuffing box during operation. The packing acts as a backup to the dynamic seal and seals against the shaft when the pump is not running. This is not viable for constantly wetted applications where slurry can get to the packing and cause undue wear. Packing life can be extended if it is lubricated with grease or flush water through the lantern ring located one or two rings from the inboard end of the box. The other option is to have a true dry packing where no supplemental lubrication is provided. This

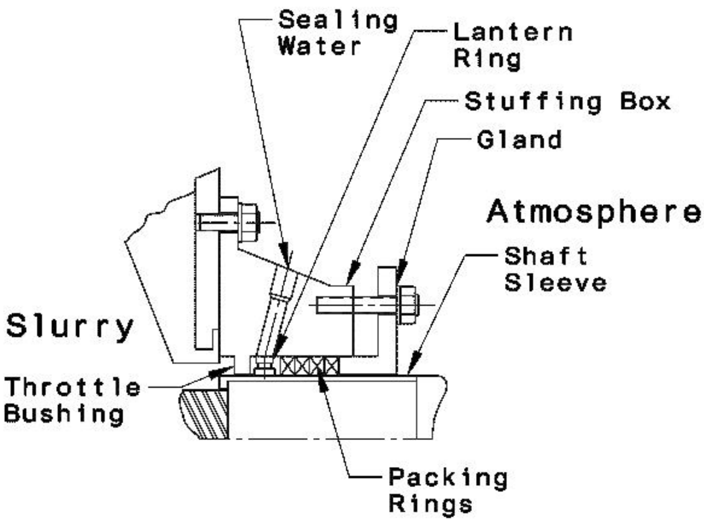


Figure 12.29 — “Flush-type” stuffing box with lantern ring in standard position

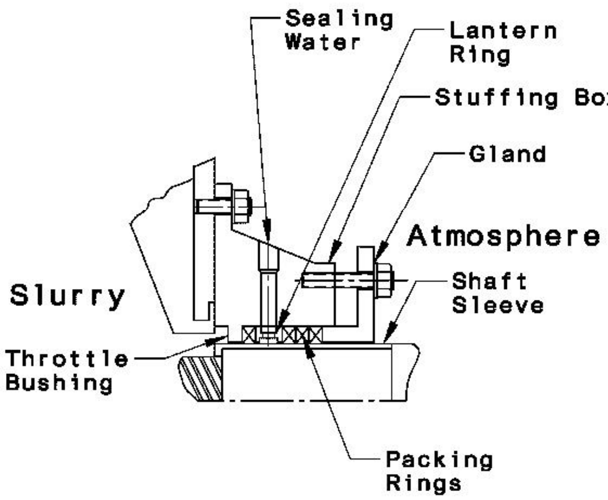


Figure 12.30 — “Weep-type” stuffing box with lantern ring in extended position

requires a packing that contains graphite or another solid lubricant. This type of seal is usually limited to wear service classes 1 and 2.

12.3.8.3.2 Packing installation and leakage

The stuffing box may or may not be packed before the shipment from the factory. If the stuffing box is not packed, it should be carefully cleaned and packed once the motor is mounted and connected to the pump. Instructions are usually provided with the box of packing. If not, the following instructions may be used as a guide.

Each packing ring should be cut so that the ends come together but do not overlap. Succeeding rings of packing should be placed so that the joints are staggered. Packing rings should be tapped down individually, but not too tightly, as this may result in burning the packing and scoring of the shaft or shaft sleeve during pump operation. When the pump is first started, the gland should be left fairly loose. Once the pump is operating normally, the gland may be tightened while the pump is running, if the leakage is excessive. A slight flow of liquid from the stuffing box is necessary to provide lubrication and cooling. Leakage should be in the range of 10 to 20 drops per minute for shaft diameters up to 100 mm (3.94 in.) and can be up to 60 drops per minute for shaft diameters in the range of 200 mm (7.87 in.).

When leakage cannot be controlled by adjusting the gland, all rings of packing should be replaced. The addition of a single ring to restore gland adjustment is not recommended.

12.3.8.3.3 Packing materials

Packing material should be selected based on operating conditions including temperature, process pH, shaft speed, discharge pressure, and flush water availability. Packing should run against a shaft sleeve that is either hardened or coated to resist wear. Sleeves should be hardened to a minimum of 50 Rc. Coated sleeves are higher in hardness with a value usually greater than 60 Rc.

Design and construction of packing varies among manufacturers, but the typical "flush type" uses a treated aramid yarn suitable for handling highly abrasive fluids. The "weep type" is a combination of treated aramid yarn with an interlaced polytetrafluoroethylene (PTFE) multifilament yarn. The "dry type" is a braided configuration containing graphite particles in a PTFE matrix. Injectable packing material is also available for special applications. This requires a modified stuffing box and a pressurized gun to renew the material as it wears.

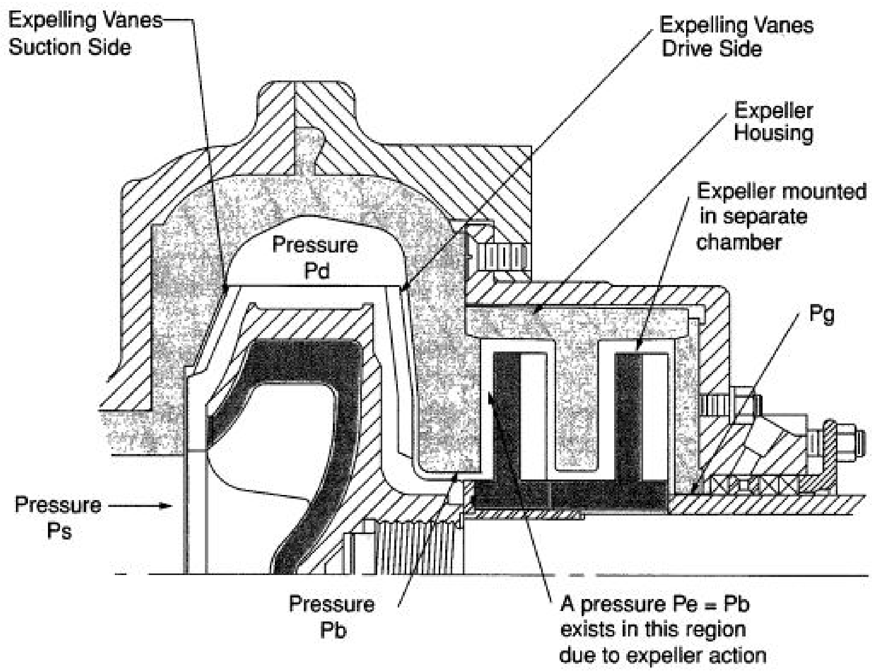


Figure 12.31 — "Dry-type" packing with dynamic seal

12.3.8.3.4 Lantern ring materials

Lantern rings can be manufactured from a variety of materials depending on the application and the nature of the process fluid. Standard-duty pumps, where the pH is near neutral, will have lantern rings made from iron, brass, PTFE, or bronze alloys. Service in acid or caustic environments will require parts made from ni-resist, PTFE, or other nonferrous materials.

12.3.8.3.5 Shaft sleeves

Pumps fitted with packed stuffing boxes shall have replaceable shaft sleeves. For wear service classes 1 and 2, these may be of stainless steel or low-hardness alloy steel.

For wear service classes 3 and 4, shaft sleeves should be of a hardened type, preferably with a chromium carbide or tungsten carbide fused overlay of hardness in the 50 to 60 Rockwell hardness range.

12.3.8.3.6 Dynamic seals

Expelling vanes fitted to the back shroud of the impeller work to keep solids out of the gap behind the rotating impeller and the stationary casing wall or liner. A separate expeller or set of expellers, called a *dynamic seal*, can be fitted to the pump. When the pump is running the dynamic seal generates additional pressure to equalize the pressures as shown in Figure 12.31 so that the pump operates without leakage. Dynamic seals need to be combined with a backup or static seal to prevent leakage when the pump is not running. The general requirements for the backup sealing device is that it must seal statically when the pump is shut down and it must run dry during operation. This can be accomplished by dry-type packing, lip seals, other proprietary devices, or mechanical seals with either dry run capabilities or fitted with a separate flush.

Dynamic seals are usually not effective on the second or higher stages of multiple pump installations, where the pumps are arranged to have the full discharge pressure of the preceding stage applied to the suction of the following stage. If the pumps are arranged at specified intervals and elevations spread out along a slurry transport line, it is possible to use dynamic seals on all stages. The arrangement should be such that the suction pressures on each stage are approximately equal and do not exceed 10-20% of the discharge pressure. An analysis should be made of the dynamic seal performance, based on actual head, flow, and suction pressure, so that proper operation is assured.

12.3.8.3.7 Mechanical seal and seal chamber design

Mechanical shaft seals are used primarily in wear service classes 1 and 2 for non-settling slurries where the solids size is small and concentrations are low. The overhung impeller, close-coupled, single-stage, end suction, metal, submersible type (Ref. Figure 12.10A), commonly employs a mechanical seal. The heavier duty, horizontal types (Ref. Figure 12.5A) applied in wear service classes 1 and 2, such as in the case of flue gas desulphurization pump service, usually utilize these seals. Mechanical seals for wear service classes 3 and 4 are possible, but should be carefully engineered. Table 12.10 may be used as a guide.

The application of seals in slurry pumps depends not only on the type of slurry being pumped, but also the design of the seal chamber. Based on experience, it is

Table 12.10 — Application limits of single mechanical seals

Seal Type	Concentration % (by volume)	Specific Gravity	Average d50 Particle Size micrometers (inches)
Split Nonpusher Design	10	1.2	≤ 1000 / 0.04
Rotating Elastomeric Bellows	10	≤ 1.4	50 – 400 / 0.002 – 0.016
Stationary Elastomeric Bellows	20	≤ 1.4	50 – 400 / 0.002 – 0.016
Heavy-Duty Slurry Design	50	≤ 1.5	100 – 1000 / 0.004 – 0.04
General guidelines for use of various seal types. Assumes a large or open bore sealing chamber, no external injection, and shaft deflection is kept to requirements of the seal manufacturer.			

not recommended to install mechanical shaft seals in pumps with stuffing boxes designed for packing, unless an external injection (ANSI Plan No. 7332) is used. Seal chambers for single seals not using an external flush (ANSI Plan No. 7311) should be large-bore, open-ended cavities. Bell housings, large tapered bore seal chambers, or large tapered bore seal chambers with vortex breakers will improve seal life by preventing a buildup of the slurry around the sealing faces that can either cause excessive erosion or packing up of the slurry around the seal that can lead to the seal running dry. The seal chambers should be self-venting by design. For no-flush applications (ANSI Plan No. 7302), care must be taken to ensure that any impeller back vanes do not cause a vacuum to develop in the seal chamber. Seals for slurries, like most other single mechanical seals, are sensitive to entrained air in the process. Air bubbles will tend to collect at lower diameters in the seal chamber when the shaft is rotating. If the air bubbles surround the seal faces, the seal can run dry leading to seal face damage and potential seal failure.

Table 12.10 provides a selection guide to the applicability of various single seal types to the type of slurry being sealed when no external injection is utilized. The slurry concentration and particle size limits when using an external injection are only limited by the ability of the injection system to exclude the slurry from the seal chamber. The associated product dilution needed to accomplish this task needs to be assessed. Figure 12.40 shows that on large diameter shafts this is normally not practical as the required bushing radial clearances to account for shaft deflections will result in excessive flow rates, or dilution, into the product.

The most common mechanical shaft sealing arrangement, by volume, is the single inside seal where the seal rotates with the shaft. Stationary mounted seal arrangements can accommodate higher shaft deflections, especially with elastomeric bellows designs, but have the disadvantage of allowing the slurry to settle around the seal. Shaft deflections on the order of .25 mm (.010 in.) can be accommodated with rotating elastomeric bellows nonpusher seals on pumps with shaft diameters less than 75 mm (3 in.) depending upon the seal design, shaft speed, and type of slurry.

Heavy duty slurry pumps can have significant shaft movement, shaft deflection, seal chamber face runout, and shaft to housing non-concentricity. Figure 12.32 shows equipment conditions that can exist for heavy duty slurry pump shafts around 152 mm (6 in.) in diameter. Combinations of these conditions in conjunction with the overall operating conditions need to be reviewed on a case-by-case basis with the pump and seal manufacturers to assure the suitability of applying mechanical seals, either rotating or stationary types, to larger slurry pumps.

Quenches can be used for scaling or crystallizing products containing dissolved solids. The quench is introduced on the atmospheric side of the seal as a connection in the gland plate or can be a separate bolt-on component. The quench can be a continuous flow of fluid when some type of sealing device contains the quench fluid or it can be used periodically to clean the atmospheric side of the seal, especially when the pump is shut down.

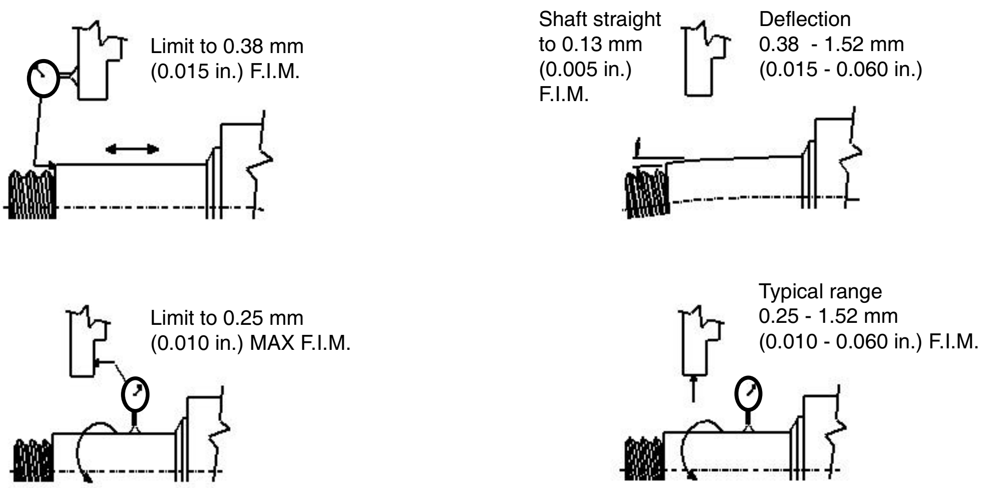


Figure 12.32 — Slurry pump shaft alignment and runout

Where more positive sealing of the quench fluid is required, dual unpressurized seals can be utilized, but are not common. In these installations the quench fluid is called a *buffer fluid*. For very viscous, dry, and/or corrosive slurries, dual pressurized seals can be used. Dual pressurized seals have the advantage of providing enhanced lubrication to the faces with a pressurized barrier fluid. This arrangement prevents process fluid leakage to atmosphere to improve safety. Conventional back-to-back seals using an ANSI Plan No. 7353 or 7354 are prone to hang-up as the slurry can accumulate under the inboard seal, where the slurry is stagnant. Back-to-back seals in conjunction with a properly flushed throat bushing using an ANSI Plan No. 7332, located inboard of the inboard seal, has proven to be an effective solution in some slurry applications. Product dilution with a compatible injection fluid is a requirement when using this arrangement. A more common arrangement is shown in Figure 12.33, where the slurry is on the outer diameter of the sealing faces, where it is not stagnant, and hang-up is not a problem. In this arrangement, product dilution is minimal. Dual pressurized seals are used when the limits in Table 12.10 are exceeded, when there is a potential for entrained air in the slurry, or when large volumes of air can be introduced into the pump.

12.3.8.3.8 Mechanical seal types

Mechanical shaft seal types used in slurry pumps range from conventional process industry designs to seals specifically designed for handling slurries with no auxiliary support. Common designs are typically supplied as cartridges and include the following basic seal component arrangements:

Single-spring elastomeric bellows seal (Figure 12.34 and Figure 12.35) - The nonpusher-type elastomer bellows seal has the advantage of free movement of the front section of the bellows and utilizes a nonclogging single-spring design. Nonpusher refers to the movement of the secondary seal along

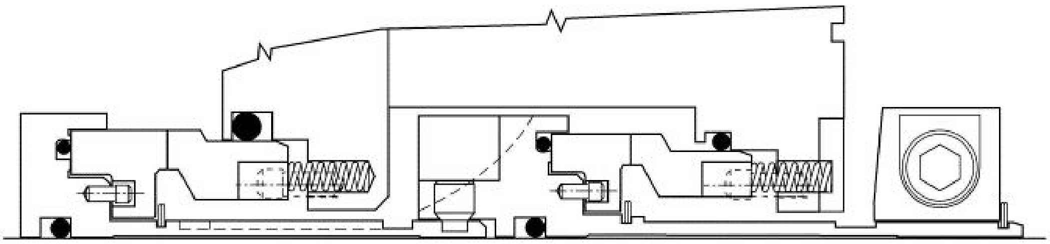


Figure 12.33 — Dual pressurized seal arrangement for slurry applications

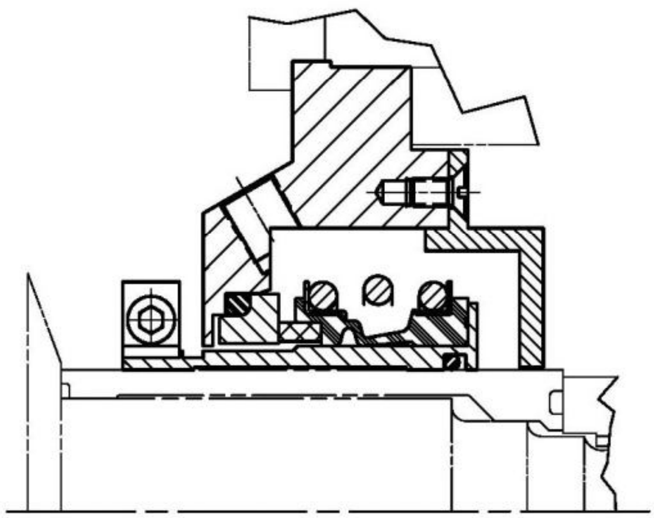


Figure 12.34 — Rotating elastomeric bellows seal

the shaft or sleeve. When mounted as a stationary seal, as shown in Figure 12.35, the probability of seal hang-up from leakage is decreased.

Heavy-duty pusher designs (Figures 12.36 and 12.37) - Typically use an O-ring secondary seal on a stationary primary ring with a stationary spring or springs on the atmospheric side of the seal, so as to be nonclogging. The drive mechanism is also on the atmospheric side to avoid damage or wear from the pumpage. These seals often use elastomeric liners, depending on the type of slurry, to avoid erosion of exposed metal surfaces. This design is typically preferred for application in heavy-duty slurry pumps.

Pusher or nonpusher split seals (Figure 12.38) - Have the advantage of replacement without complete disassembly of the pump. Designs vary from a fully split seal to others that have only split secondary seals and faces with the other nonwearing components being solid. Split pusher seals would normally be limited to runouts less than 0.1 mm (0.004 in.) depending on size, speed, and the type of slurry. Some nonpusher split seals can accommodate runouts up to 1.3 mm (0.05 in.).

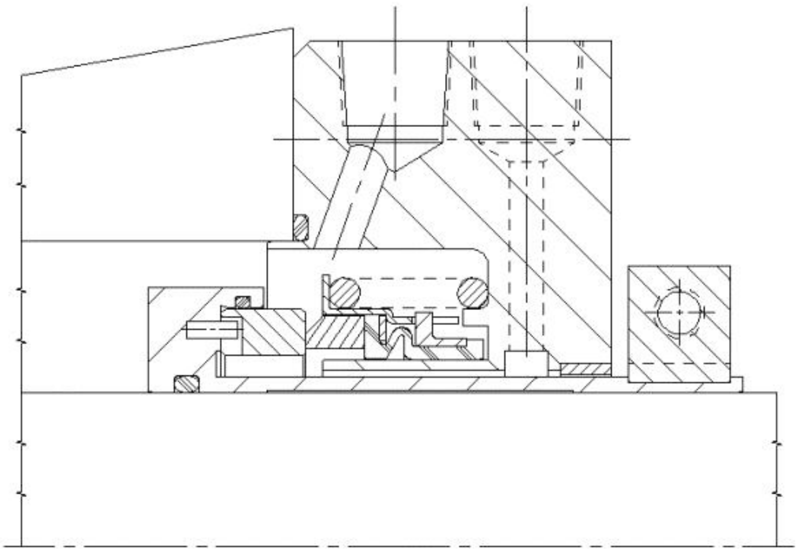


Figure 12.35 — Stationary elastomeric bellows seal

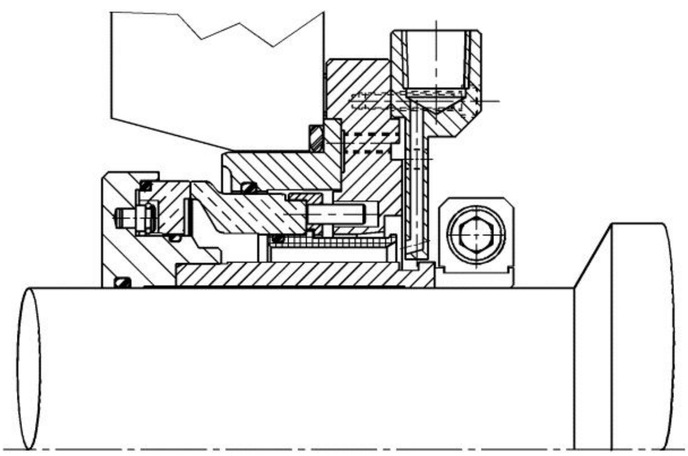


Figure 12.36 — Heavy-duty slurry seal with quench device

Specialty designs (Figure 12.39) are typically nonpusher designs using an elastomeric seal that may also act as the spring or utilize single coil or encapsulated finger-type springs that are nonclogging.

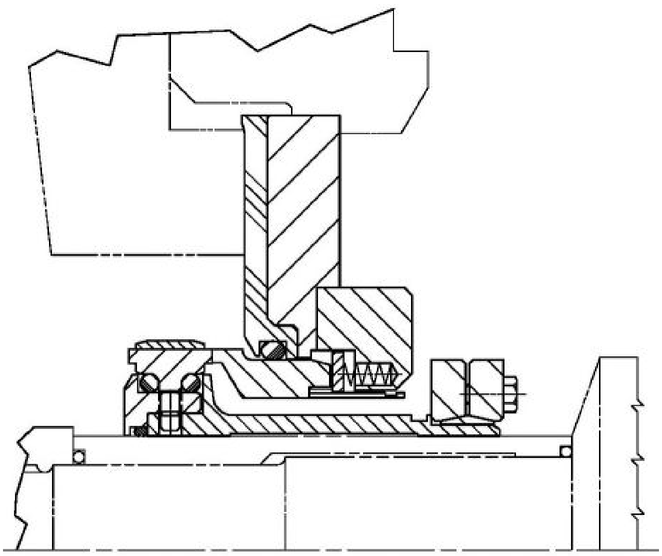


Figure 12.37 — Heavy-duty slurry seal with elastomeric liner

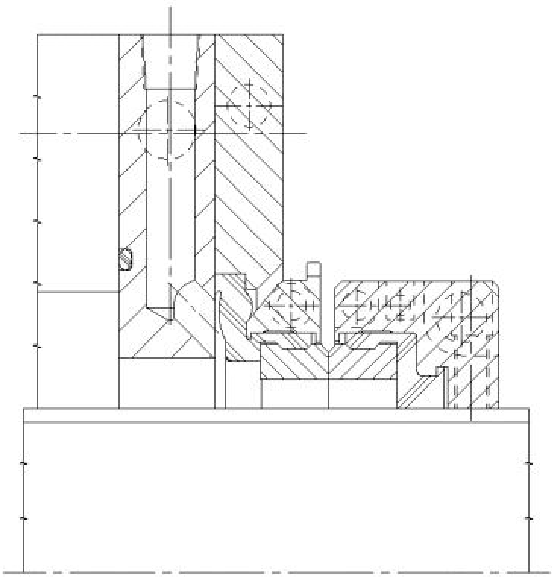


Figure 12.38 — Fully-split nonpusher seal

12.3.8.3.9 Mechanical seal and packing flush arrangements

Flush arrangements used in conjunction with end face mechanical seals vary from no-flush designs (ANSI Plan No. 7302) to elaborate external circulating systems (ANSI Plan No. 7354). This is a brief description of flush plans that can be used:

ANSI Plan No. 7302 - Dead-ended seal chamber with no forced circulation. Typically used where the seal faces are located in the pump casing or with tapered bore seal chambers where the seal faces are exposed to the product so as to avoid clogging. Can be used with jacketed seal chambers or glands for thermally sensitive slurries. The main advantage is no product dilution.

ANSI Plan No. 7311 – Recirculation from pump case through an orifice to the seal chamber. This plan can be used on lighter-duty slurries that would not tend to clog the orifice or the flush hole in the seal chamber or seal gland. It is not commonly used as the velocity impacting the seal could damage or erode the seal components located near the entrance of the flush into the seal chamber.

ANSI Plan No. 7332 - Injection from an external source into the seal chamber or stuffing box. This plan almost always uses a close clearance throat bushing to isolate the pumped product from the seal chamber. The velocity past the bushing to exclude the slurry should be on the order of 4.6 m/s (15 ft/s) as shown in Figure 12.40. The bushing will wear with shaft deflections, opening up the clearance, which increases the flow rate and process fluid dilution with time. Like packing arrangements, the clean flush will dilute the product, but to a lesser degree. This needs a reliable clean external source along with a proper control system.

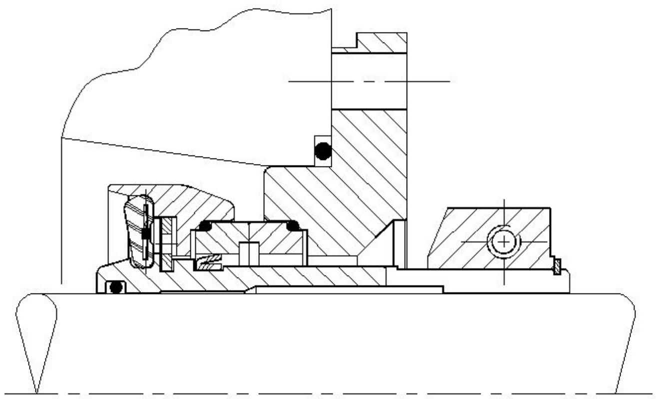


Figure 12.39 — Specialty slurry seal

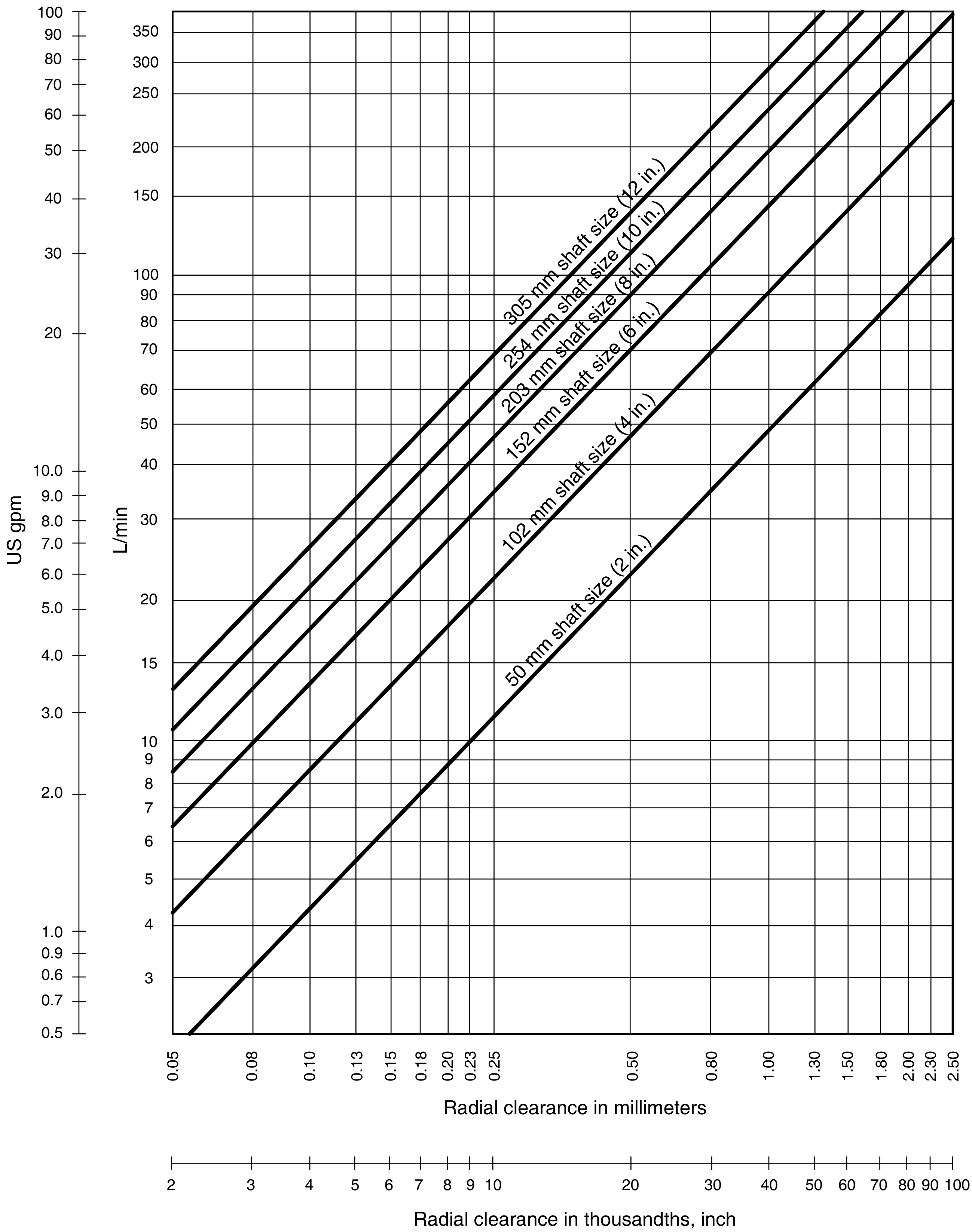


Figure 12.40 — Flow rates required to create 4.6 m/s (15 ft/s) velocity past a bushing

Water quality is an extremely important factor in both mechanical seal and packing operation. The water used should be within the limits listed in Table 12.11. This is attainable with relatively inexpensive filtration treatment equipment available and will provide for good packing life.

ANSI Plan No. 7352 - Uses an unpressurized reservoir to provide a low-pressure buffer fluid for the outer seal of a dual unpressurized seal arrangement. Used to provide a buffer to scaling-type slurries.

ANSI Plan No. 7353 - Pressurized reservoir to provide a barrier fluid at a pressure higher than the pumpage for dual pressurized seals. The advantage is that the seal is lubricated by a clean barrier fluid and there is limited process fluid dilution. The disadvantage is maintenance intervals and increased capital cost.

ANSI Plan No. 7454 - Uses an external source to provide a clean pressurized barrier fluid to a dual pressurized seal. The external source can vary from a relatively simple controlled and instrumented water source to an elaborate closed-loop system with circulation pumps.

ANSI Plan No. 7362 - An external low-pressure quench to the atmospheric side of the seal to prevent crystalline solids from forming. Typically, the quench is water or steam. Depending on the application, the quench can be used continuously during operation or periodically to clean out the atmospheric side of the seal. The quench can also provide limited cooling to the mechanical seal and enhance face lubrication during periods of dry running.

12.3.8.3.10 Mechanical seal materials

The materials used in the mechanical seals are suited to the particular slurry being sealed. Metal hardware is typically 300 series stainless steel, duplex stainless steel alloys, or a nickel alloy for more corrosive applications and high-chrome iron for abrasive slurries. Elastomeric bellows or O-rings are available in any of the common materials ranging from Nitrile to perfluoroelastomers. Seal face materials commonly used are:

Carbon Graphite - used on one of the faces against either tungsten carbide or silicon carbide. Has good chemical resistance, but poor erosion and abrasion resistance. Should only be used with a ANSI Plan No. 7332.

Tungsten Carbide - a high-strength hard face material with typically either a cobalt or nickel binder. Typically run against itself or silicon carbide for enhanced erosion and abrasion resistance.

Silicon Carbide - a family of high-strength hard ceramics. There are a number of variations, including reaction bonded or alpha sintered grades for the most corrosive and abrasive applications. There are also silicon carbides with select porosity or graphite loading to enhance face lubrication during periods of dry running. When using either of the above two select silicon carbides materials, both seal faces may be the same material, or one of the faces can be tungsten carbide or another silicon carbide grade.

12.3.8.4 Flanges

Flanges generally should conform to appropriate ANSI, ISO, JIS, or other recognized flange standards. The unique features of slurry pumps, such as liners and heavy wall sections, sometimes make this impractical because the nozzle wall interferes with the bolting. The next larger size flange is often used to alleviate this problem. This requires adapter pieces or similar special mating flanges on the attached piping. Agreement between the supplier and purchaser is needed in such cases.

12.3.9 Drive train arrangements

Slurry pumps are sized to specific applications by varying the speed, impeller diameter, or both depending on materials, construction, and preferences. The choices determine the drive system. Gray iron, ductile iron, and stainless steel impellers are often trimmed to meet the required service conditions using constant-speed, direct-drive arrangements.

Hard iron and elastomer impellers are difficult to turn down so speed is often changed with either belt drives

Table 12.11 — Recommended water quality limits

pH	Dissolved Solids	Suspended Solids	Filtration
6 – 8	< 1000 mg/L (< 0.013 oz/gal)	< 100 mg/L (< 0.0013 oz/gal)	100% of 60 μm (0.0024 in.) or larger particles removed

or variable-speed drives to meet the required service conditions. The type of driver used for nonsynchronous speeds generally varies by size. On smaller pumps, motors up to 186 kW (250 hp) are often mounted overhead to save floor space and can be driven with cog or V-belts. Medium-sized units are usually V-belt driven and motors are arranged along the side of the pump, or with motor behind and in line with the pump. On larger pumps requiring over 746 kW (1000 hp), speed reduction is accomplished using gearboxes with constant-speed motors. Very large pumps, such as dredge units, are usually driven with diesel engines.

12.4 Installation, operation, and maintenance

12.4.1 Installation

When slurry pumps are operated, special precautions must be taken. The basic reference, *ANSI/HI Standards 1.4, Centrifugal Pumps* on installation, operation, and maintenance remains applicable.

12.4.1.1 Special requirements

Because slurry pumps may have special requirements for operation, close attention to application details is important. The user must discuss these items in detail with the manufacturer and account for them in the system design and pump selection. These include, but are not limited to, oversize solids, solids settling considerations, viscosity effects, and the effect of froth or entrained air. Pumps should be installed with provisions for flushing solids during extended stoppage or prior to repairs.

12.4.2 Nozzle loads

Testing of nozzle loads on the above-described pumps has shown that misalignment between the pump and motor shaft occurs from movement of the casing relative to the base plate. The loading required for the movement is less than that which would produce excessive stresses in the nozzles or internal distortion of parts. The amount of loading that results in movement depends on the size of hold-down bolts, the amount of torque applied to the bolts, and the bolt material.

The orientation of the nozzle loads, as identified in Figure 12.41, are for those applied to cast-iron casings mounted on the machined mounting surface(s) of a carbon steel base plate. The loads are for ASTM A307 Grade A bolts with no lubrication applied to the bolt threads.

12.4.2.1 Driver and pump

The allowable radial movement of the pump and of the shaft measured at the coupling due to nozzle loading shall not exceed 0.13 mm (0.005 in.) parallel to the initial alignment. Axial movement of the pump shaft at the coupling has not been considered.

12.4.2.2 Limiting factors

Tests have shown that the limiting factors are not due to:

a) Bending of the shaft at the seal chamber or stuffing box

b) Internal distortion of parts

Limiting factors are due to:

a) Tension stress of the hold-down bolting (pump casing to the base plate)

b) Movement of the casing relative to the base plate

c) Grade of bolt

d) Torque applied to the bolts

e) Bending stress in the nozzles

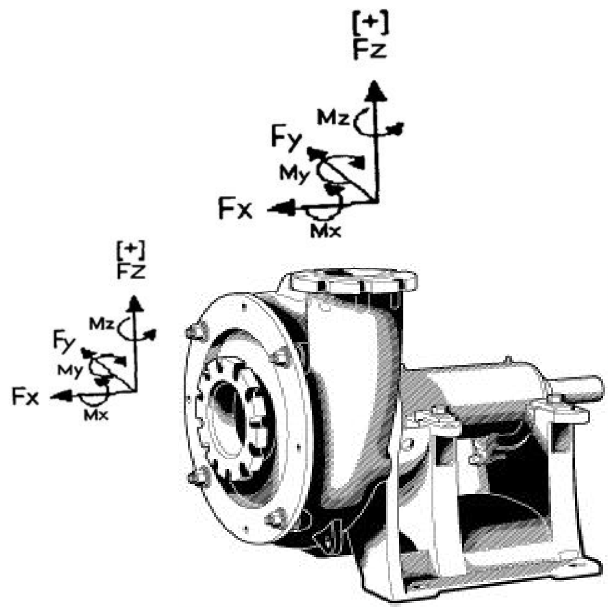


Figure 12.41 — Figure showing direction of forces and moments being applied to suction and discharge nozzles

12.4.2.3 Pump hold-down bolts

The maximum allowable tensile stress for the hold-down bolts is 90% of ASTM A307 Grade A yield strength. The maximum allowable shear stress for the hold-down bolts is 25% of ASTM A307 Grade A yield strength.

The pump shall be bolted to the base plate and sufficiently tightened to prevent slippage or movement relative to the base plate. Refer to *API 686, Appendix E* for the required torque values. It may be necessary to arrange for periodic tightening of the bolts to maintain the required torque.

It can be argued that some of the bolts may be bottomed-out against the wall of the bolt hole at initial installation. It can also be argued that with small bolts, the shank of the bolt may bend as the side force overcomes the friction force. To keep the presentation in Section 12.4.2.4 uncomplicated, friction force will be the criterion.

12.4.2.4 Assumed effect of nozzle loading

Forces in the X and Y directions, and the moments about the Z axis (see Figure 12.41), are assumed to be distributed equally to all the hold-down fasteners in all four feet. Movement of the pump occurs when the force overcomes the static horizontal friction force on all four feet induced by the torque of the hold-down bolts. The static weight of the pump is relatively small compared to the force induced by the torque of the bolts.

Forces in the Z direction, and moments about the X- or Y-axis, are assumed to be distributed to fasteners in two feet resulting in yielding of the fasteners. The yielding load is the difference between the torque stress applied bolts and the yield stress of the fastener.

It is also assumed that the suction and discharge pipes do not restrain pump movement.

Friction force:
Static coefficient of friction
Cast iron against carbon steel: $\mu = 0.4$

Maximum allowable loads based on hold-down bolts			
Type of load			
Moments N-m (lb-ft)		Forces N (lbf)	
M_x and M_y	M_z	F_x and F_y	F_z

For a combination loading use the square root of the sum of the squares as follows:

$$\sqrt{\left(\frac{M_y}{M_y\text{max}}\right)^2 + \left(\frac{M_z}{M_z\text{max}}\right)^2 + \left(\frac{M_x}{M_x\text{max}}\right)^2 + \left(\frac{F}{F\text{max}}\right)^2} \le 1$$

Where:

M_y = Vertical bending moment

M_z = Horizontal bending moment

M_x = Torsion moment

F = Tension or compression load (direction of load does not change sign in equation.)

12.4.2.5 Allowable nozzle load based on hold-down capability criterion

The following four criterion must be met:

a) The load must not cause the pump to move horizontally relative to the rigid baseplate.

b) The load must not cause the pump to move vertically relative to the rigid baseplate.

c) The maximum tensile stress in the hold down bolts must not exceed 90% of ASTM A307 Grade A fastener yield strength (275,800).

$$s_{ta} = 275.8 \times 0.9 = 248.2 \text{ MPa } (36{,}000 \text{ psi})$$

d) The maximum shear stress in the hold down bolts must not exceed 25% of ASTM A307 Grade A fastener yield strength (275,800).

$$s_{sa} = 275.8 \times 0.25 = 68.95 \text{ MPa } (10{,}000 \text{ psi})$$

12.4.2.6 Allowable sideways slide force per bolt

$$F_a = \frac{A \times E_d \times \mu \times s_{sa}}{FS}$$

Where:

F_a = Allowable sideways slide force per bolt, N (lbf)

A = Cross-sectional area per bolt, mm^2 (in^2)

E_d = Dry lubricant effectiveness, (50% = 0.5)

μ = Coefficient of static friction. Cast iron against steel, (0.4)

s_{sa} = Allowable shear stress, 68.95 MPa (10,000 psi)

FS = Factor of safety (2.0)

$$F_a = \frac{A \times 0.5 \times 0.4 \times 68.95}{2}$$

$$F_a = 6.895 \times A(\text{mm}^2) \quad \text{N per bolt}$$

$$F_a = 1000 \times A(\text{in}^2) \quad \text{lbf per bolt}$$

Conversion factors:

To convert kilopascals (Pa) to pounds per square inch (psi), multiply by 0.145.

To convert force in pounds (lbf) to Newtons (N), multipy by 4.448.

To convert area in square inches (in^2) to square millimeters (mm^2), multiply by 645.2.

To keep the analysis simple, moments and forces applied to the sides or that are off-center from center of rotation, are not adjusted to the center.

12.4.2.7 Sliding forces and moments

For Figures 12.42 through 12.44 the sliding forces and moments are:

Force		Moment	
Suction	Discharge	Suction	Discharge
F_{xs} F_{ys}	F_{xd} F_{yd}	M_{zs}	M_{zd}

12.4.2.8 Calculating allowable forces on hold-down bolts

Calculate the total allowable force (F_t) for the hold-down bolts on one side of the pump where they will be in tension. Then determine how much force or moment is permitted at the suction or discharge connection.

$$F_t = s_{ta} \times A \times n$$

Where:

F_t = Total allowable force, N (lbf)

s_{ta} = Allowable tensile stress, kPa (psi)

A = Cross-sectional area per bolt, mm^2 (in^2)

n = Number of effective bolts in tension

In the example shown in Section 12.4.2.6, the allowable tensile stress for the hold-down bolts is 248,220 kPa (36,000 psi). The total allowable force may be calculated as:

$$F_t = 248{,}220 \times A \times n, \quad \text{N}$$

or

$$F_t = 36{,}000 \times A \times n, \quad \text{lbf}$$

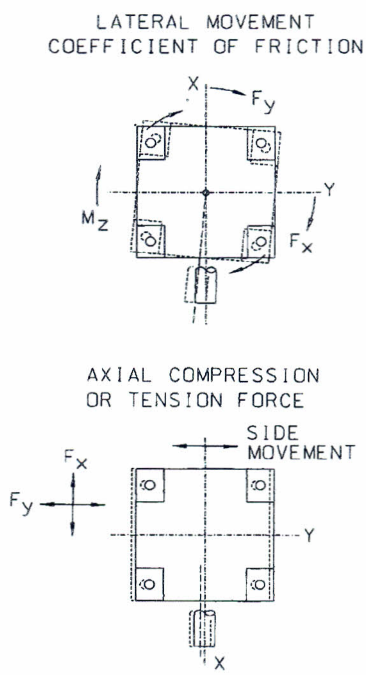


Figure 12.42 — Sliding movement when forces and moments are applied to pump feet

Nozzle Configuration		
Type Figure		
	Connection	
Figure 12.43	Suction	Discharge
A	Side	Top
B	Side	Side
C	Top	Top suction and discharge, both on centerline
D	Side	Side
E	Top	Top suction and discharge, offset from centerline
F	End	Top discharge offset from centerline

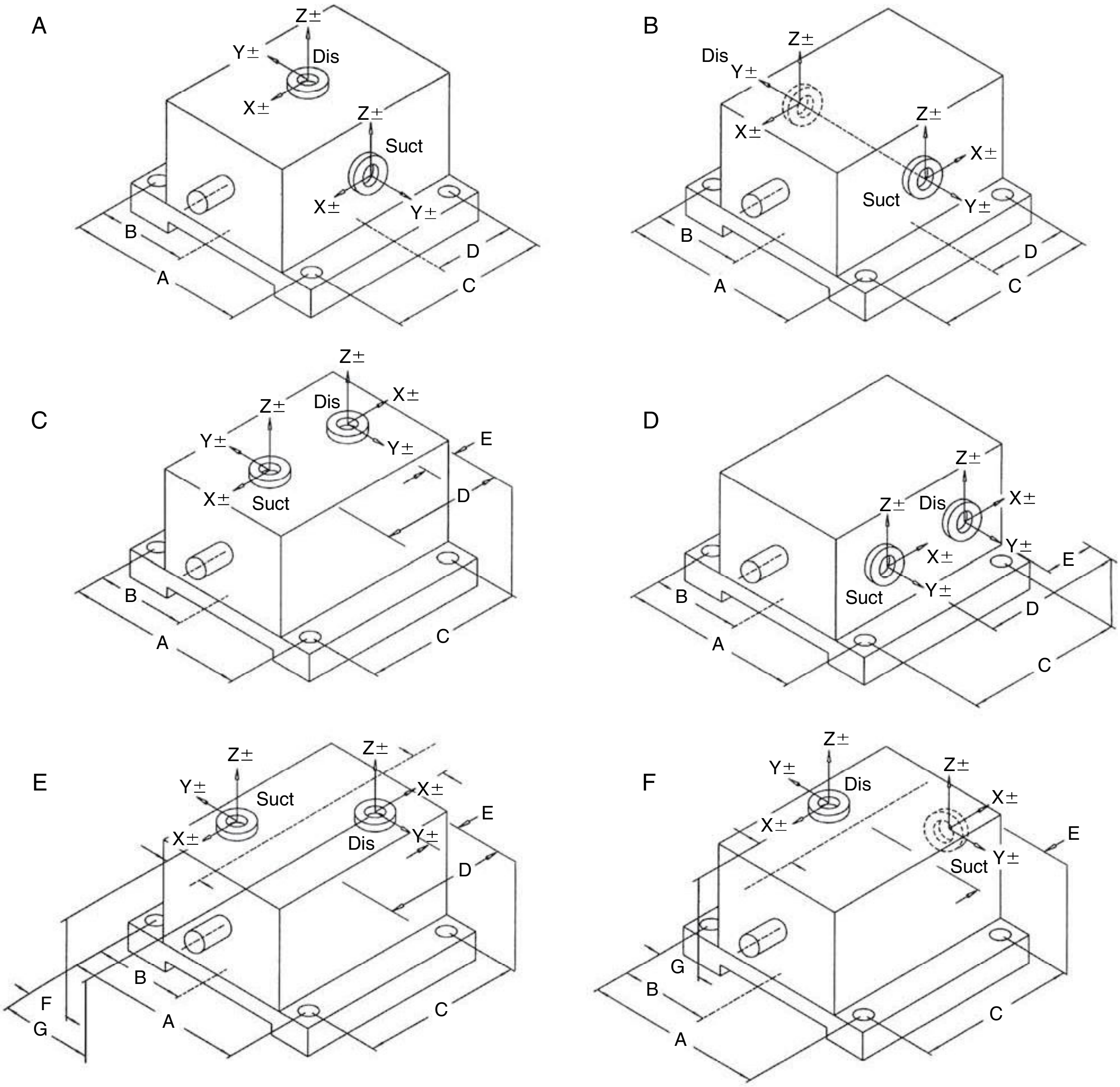


Figure 12.43 — Forces and moments applied to pumps with various suction and discharge nozzle locations

12.4.2.9 Bolts in tension

Table 12.12 — Calculation of allowable forces – N (lbf) and moments – N-m (lb-ft) on the suction(s) and discharge(d) connections

Total Allowable Force on the Bolts = F_t						
Figure 12.43	M_{xs}	M_{ys}	F_{zs}	M_{xd}	M_{yd}	F_{zd}
A	$F_t \times A$	$F_t \times \frac{C}{D} \times C$	F_t	$F_t \times \frac{A}{B} \times A$	$F_t \times \frac{C}{D} \times C$	$F_t \times \frac{A}{B}$
B	$F_t \times A$	$F_t \times \frac{C}{D} \times C$	F_t	0	$F_t \times \frac{C}{D} \times C$	F_t
C	$F_t \times \frac{A}{B} \times A$	$F_t \times \frac{C}{D} \times C$	$F_t \times \frac{A}{B}$	$F_t \times \frac{A}{B} \times A$	$F_t \times \frac{C}{E} \times C$	$F_t \times \frac{A}{B}$
D	$F_t \times A$	$F_t \times \frac{C}{D} \times C$	F_t	$F_t \times \frac{C}{E} \times C$	$F_t \times \frac{C}{E} \times C$	F_t
E	$F_t \times \frac{A}{F} \times A$	$F_t \times \frac{C}{D} \times C$	$F_t \times \frac{A}{F}$	$F_t \times \frac{A}{G} \times A$	$F_t \times \frac{C}{E} \times C$	$F_t \times \frac{A}{G}$
F	$F_t \times \frac{A}{B} \times A$	0	$F_t \times \frac{A}{B}$	$F_t \times \frac{A}{G} \times A$	$F_t \times \frac{C}{E} \times C$	$F_t \times \frac{A}{G}$

NOTE: Use caution with these units. Linear dimensions are in meters. When using US Customary units and linear dimensions are given in inches, divide by 12 to calculate lb-ft.

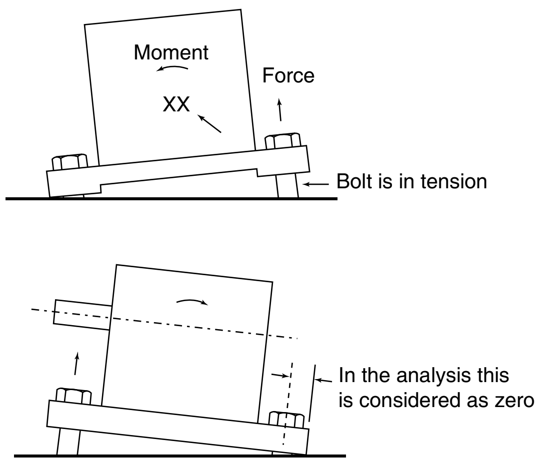


Figure 12.44 — Bolts in tension when a moment is applied to the pump

12.4.2.10 Extension procedure

The nozzle load procedure can be extended further to establish the actual stresses of the hold-down bolts using the method outlined in Appendix B.1, reference 14.

12.4.3 Connecting piping

The suction and discharge piping should be subject to *ANSI/HI 1.4.3.6 and 1.4.3.7 Discharge* and *Suction Piping Requirements* except where the special properties of slurries may require specific modification. Never use the pump itself as an anchorage point for the piping. The permissible pipeline forces must not be exceeded.

Suction lines should be self-venting. If the source of pumpage is below the pump, the line should slope upward toward the pump; if the source is above the pump, the line should slope downward toward the pump. The pipelines should be anchored in close proximity to the pump and should be connected without transmitting any stresses or strains. The diameter of the pipe should be at least equal to the diameter of the pump nozzle.

Thermal expansions of the pipelines must be compensated by appropriate measures so as not to impose any extra loads on the pump exceeding the permissible pipeline forces and moments.

Before starting, the procedures of *ANSI/HI 1.4* should be observed.

If the impeller is threaded onto the shaft, running in the wrong direction of rotation or applying dynamic braking, even momentarily, may cause the impeller to unscrew. This can result in extensive damage to both the hydraulic and mechanical components.

12.4.4 Commissioning

The user should read the manufacturer's instruction manual and follow the instructions. At a minimum that should include the following:

- The operating data; the oil level, if required; and the direction of rotation of the driver must be checked. The pump must be primed.
- Make sure the unit is properly connected to the electric power supply and is equipped with all protection devices.
- Make sure all auxiliary connections are connected and functioning.

12.4.5 Start-up

Most slurry pipelines start-up and shut-down on clean water, in which case the start-up procedure should follow *ANSI/HI 1.4*.

Where there is need to start-up and shut-down with the slurry still in the line, the manufacturer's instruction book should be read and the particular characteristic of the specific slurry should be considered.

In the case of settling slurries, this could require the avoidance of plugging due to settling by eliminating low spots in the pipeline and limiting the concentration at which a shut-down occurs.

12.4.6 Storage of elastomer linings

Pumps with elastomer linings and spare liners should be stored in a cool, dark location free from electrical equipment, motors, or any other ozone-generating devices. Exposure to petrochemical products, direct sunlight, or temperatures in excess of 50°C (120°F) must be avoided.

Properly stored elastomer parts should retain their properties for about two years for gum rubber, or five years for Neoprene® or urethane. The parts should be periodically inspected for the presence of a soft chalky layer, easily rubbed off, which would indicate deterioration. Darkening or discoloration of elastomer parts over time is a natural occurrence and does not by itself indicate any loss of properties.

12.4.7 Impeller removal

Refer to the manufacturers' operating and maintenance instructions. Never apply heat to any impeller.

12.4.8 Axial adjustment of the bearing housing

Designs with axial running clearances must be adjusted periodically to reestablish recommended internal running clearances to compensate for wear. Refer to the manufacturers' instruction manuals for recommended values.

12.4.9 Piping system design

The general requirements in *ANSI/HI 1.4* for piping and *ANSI/HI 9.8* for sump should be followed when designing piping systems.

Slurry piping must be sized as outlined in Section 12.3.2 to achieve the correct range of velocity for the expected pumping conditions, otherwise, high wear, excess energy consumption, or plugging may result.

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

- Avoid low spots.
- Use only vertical or horizontal pipelines for coarse settling slurries. Piping should be vertical or horizontal. Inclined pipelines may surge due to a backward drift or buildup of solids. Inclined pipelines also result in increased friction losses.
- Provide means to flush out the piping in sections where blockage may occur.
- Keep valves to a minimum. Where valves are necessary, use a style valve that has no pockets or areas inside the valve to accumulate deposits.
- Maintain accessibility to potential wear areas (joint, elbows, and geometry changes).
- Install gauge savers (Section 12.6.6.3) when gauges are used.

Good slurry sump design should be aimed at preventing uneven flow into the pump suction while providing for transportation of pumped solids to the pump suction. Wet wells or above ground feed hoppers for solids-bearing liquids require special considerations to allow for the effective transport of settling solids into the pump. These considerations include a geometry and provisions for cleaning of the structure to remove material that would otherwise be trapped and result in undesirable surging conditions.

The main geometrical principle is to minimize horizontal surfaces in the structure anywhere but directly under the pump suction, thereby directing all solids to a location where they may be transferred by the pumping equipment. Vertical or steeply sloped (at least 45 degrees above horizontal) sides shall be provided for the transition from upstream conduits or channels to the tank outlet to the pump. This will greatly reduce the chance of solids settling and sloughing into the pump causing severe surging and reduced pump wear life. Coatings or treatments that decrease the sump surfaces roughness and friction will also help to transport settling solids to the pump suction nozzle. Where reducers are necessary, utilize the eccentric type with the flat side down to promote drainage and minimize buildup of solids.

The sump volume should be selected so that a reasonable balance between pump off times and the maximum number of pump starts per hour is found. Excessive periods of off time will create increasingly difficult sediment removal. The minimum wet well level shall be set in accordance with recommendations of ANSI/HI 9.8 on pump intake design to prevent surface vortices. Taller hoppers or deeper wells are preferred over wider flat-bottom vessels.

Sump design should include baffles, submerged inlet, or other methods to prevent entraining air in the pumpage, which will result in frothing. If frothing cannot be avoided, it must be accounted for in the system design and operation (Section 12.3.3).

12.4.10 Possible operating problems

It should be noted that the pump always operates at the intersection of the pump curve and the pipeline "system" curve.

During the initial stages of operation, pump load on the driver should be checked. If there is an excess amount of power being drawn by the pump, it may be caused by the system head being lower than predicted, thus resulting in higher flow rates and power consumption. This sometimes happens when a safety factor is applied to the head requirements during the design of the system. Cavitation may also occur under these high flow conditions. To correct this problem, the pump speed, impeller diameter, or both should be reduced. Gray iron, ductile iron, and stainless steel impellers are often trimmed to change performance. Hard iron and elastomer impellers are difficult or impossible to turn down, so speed is usually changed with either belt drives or variable-speed drives to change performance.

If actual supply flow rates are lower than predicted, the sump may be emptied causing the system to surge and accelerate pump wear. Pump speed or impeller diameter should be decreased or makeup water supply increased to keep the sump at the highest stable level possible. If the flow variations are too great, a variable-speed motor may be required. This problem is especially common in applications with a high proportion of static head, such as mill discharge and cyclone feed. It can be further aggravated by operation well below the best efficiency flow rate of the pump where the pump head curve is relatively flat. Under these conditions, minor fluctuations in the system resistance caused by variations in solids concentration or size can result in surging flow rates.

Avoid prolonged operation at flows well below the optimum flow rate. This causes recirculation of slurry within the pump and accelerates localized wear.

In the event problems are encountered, the following information should be furnished to the pump supplier to assist in evaluation of the problem:

a) The approximate flow rate desired, including the actual minimum and maximum flow rate, if known.

b) Composition of the slurry, including specific gravity and material size.

c) The system static head (the difference in elevation between the water level on the suction side of the pump and the point of discharge).

d) The length and size of suction and discharge lines, including a description of the general arrangement including all fittings, bends, and valves.

e) If the discharge point is not atmospheric, consider discharge area requirements.

f) If suction is taken from a sump, provide the general arrangement, including internal dimensions, minimum and maximum sump levels referenced to the suction centerline of the pump, and suction pipe inlet configuration.

g) The available driver horsepower and rpm, speed of pump, and description of the ratio device between the pump and motor.

h) The impeller diameter, if different from that supplied with the pump.

The above information is especially important when a pump has been transferred from the duty for which it was selected to some other application.

In many instances, unusual wear in the pump or lower operating efficiencies are caused by a mismatch between the pump and the system. This can usually be corrected once the actual operating conditions are known.

12.4.11 Spare parts stock

Due to the erosive and/or corrosive action of slurry, many of the wetted components of the pump may require replacement in the course of normal maintenance. Inspection or overhaul of the mechanical components may also lead to the replacement of certain parts. Refer to the pump manufacturers' recommendations and previous maintenance records to establish suitable spares inventory.

12.4.12 Maintenance procedures for maximum part life

Slurry pumping is a difficult service and severe wear should be expected. The following procedures are often used to optimize useable service life.

12.4.12.1 Side liners

Uneven or localized wear is not uncommon. On many pump designs, the side liners can be rotated to several different positions. When local wear occurs, these liners can be rotated 180° to even out the wear and avoid premature failure.

12.4.12.2 Impeller

Pumps should be adjusted as needed to maintain proper running clearances. Excessive clearances between the impeller nose and suction liner will lead to accelerated wear and reduced efficiency.

An impeller does not need to be replaced until it fails to produce sufficient head for the application. Pump performance, not visible wear, should be the criteria for impeller replacement. Inspect for cracks or other defects in the shrouds or vanes that could lead to failure.

Uneven wear can affect the impeller balance, resulting in excessive vibration. This is rare, but can be corrected by rebalancing the impeller.

Never apply heat to an impeller when removing it. Refer to the manufacturer's instruction manual and use the appropriate procedure.

12.4.12.3 Hard irons

Abrasion-resistant irons are brittle and will crack if subjected to intense heat. Hard iron parts should never be welded or heated.

12.4.13 Operational considerations

The sump condition should be observed during operation to ensure that solids are not building up and sluffing off and that vortices are not forming. Air entrainment should be avoided.

The sump should not be pumped out (emptied) except for cleaning as it can result in surging that causes accelerated pump wear. Pump speed or impeller diameter can be reduced and makeup water can be added to avoid emptying the sump. If flow variations are too great, a variable-speed motor may be required.

In dredging applications where the suction pipe is lowered into the solids being pumped, it is useful to have pressure gauges on the suction and discharge to determine when cavitation occurs. This enables the operator to maintain maximum solid throughput.

12.5 Intentionally left blank

12.6 Testing

12.6.1 Scope

12.6.1.1 Hydrostatic tests

Testing shall be done in conformance with the *ANSI/HI Standard 1.6, Centrifugal Pumps Tests*.

12.6.1.2 Performance tests

Slurry pumps are not routinely tested in the factory. If facility testing is specified, it shall be done in accordance with *ANSI/HI 1.6* using clear water. The clear-water rating for the test shall be calculated from the slurry duty point using the derating rules in Section 12.3.3.

If vibration tests are required, then they shall be in accordance with ANSI/HI 9.6.4.

12.6.1.3 Optional slurry test

The standard test of Section 12.6.1.2 is normally sufficient to ensure satisfactory operation on-site. Rarely, a very extreme application may fall outside the normal range of experience where there is no guidance for determining the equivalent clear-water rating. Excess power (larger driver) is normally provided in such cases and speed or impeller diameter changes are made in the field to obtain the needed performance. If this is not possible, a facility test on the actual slurry to be handled is sometimes considered. Slurry tests may not be practical depending on available facilities and the actual slurry to be handled. Slurry tests are expensive, therefore, should only be considered for extremely critical services where there is no other alternative.

When a facility test on slurry is specified, the type of tests performed, and the auxiliary equipment used, should be agreed upon by the purchaser and manufacturer prior to the test and the requirements of Sections 12.6.1.4 through 12.6.6.4 shall apply.

12.6.1.4 Objective

This section provides uniform procedures for the hydraulic and mechanical testing (including documenting the test results) of the performance of a centrifugal slurry pump while operating and pumping a slurry. Test procedures will be defined which may be invoked by a contractual agreement between a purchaser and manufacturer. It is not intended to define any manufacturer's standard practice.

Variations in test procedures may exist without violating the intent of this standard. Exceptions may be taken if agreed upon by all parties involved, without sacrificing the validity of the applicable parts of this standard.

12.6.2 Test conditions

Unless otherwise specified, the capacity, head and efficiency are based on shop tests at a slurry ambient temperature of less than 38°C (100°F). If the facility cannot test at rated speed because of limitations in power or available speed changers, the pump may be tested at between 80% and 120% of rated speed. It is permissible on pumps greater than 225 kW (300 hp) to test at speeds between 60% and 140% of rated speed.

12.6.3 Manufacturer’s testing

When the tests are carried out at a manufacturer’s facility, the tests will be done in a closed loop out of an open sump or containment vessel.

In this case, the slurry will recirculate many more times through the system than it normally would, so the solid particles may degrade or change such that their effect on the pump will change.

With fine slurries in the 100-micrometer (0.004-in.) size range, or where there is a wide particle size distribution, the change will likely be less than what can be measured. With particles 3 mm (0.12 in.) and larger with no fines, there will be a measurable change during the test and special precautions and/or adjustments will need to be made.

The degradation effects will be increased when a valve is used to change the system resistance.

To minimize degradation, slurry loading and testing should be conducted as fast as possible.

Given that some degradation will occur, slurry samples should be collected at regular intervals from the start of loading until the end of the test. This way the degradation trend can be monitored and recorded in terms of particle shape and size distribution.

To minimize the effect of degradation on the results, the most important duty test point should be taken first. At the end of the test, the first data point should be repeated to measure any change in the effects of the slurry degradation.

12.6.4 Field tests

In actual pump applications, there is no degradation of the slurry to consider. Rather, there are variations of flow, concentration, solids size, and temperature. Typically it is only possible to get flow information from the user's indicated service readings rather than a full or constant speed characteristic. The fluctuations in flow and concentration will be significant and the results will have to be evaluated and averaged carefully.

Intermittent higher concentrations of slurry in the line are possible, so the flow and head measurement sensing locations should be kept as close as possible to each other to avoid inaccurate readings (Sections 12.6-12.6.3).

A clear-water baseline test is the best way to verify pump performance and check the field instrumentation. This may not be possible in the field, so accurate data collection and conversions are required. Note that readings may also vary from the design duty conditions if the pump has been in service for some time or the impeller clearances are not properly adjusted.

12.6.5 Wear tests

It is not practical to run a pump to the wear failure point in a closed-loop lab test due to the time and cost involved. The amount of new slurry required to compensate for degradation would distort the results.

The SAR Number Test procedure from *ASTM G75-95-Determination of Slurry Abrasivity (Miller Number)* and *Slurry Abrasion Response of Materials (SAR Number)* may be used to screen materials appropriate for a particular slurry service.

It is possible to identify areas of high wear and determine initial wear rates with short duration slurry runs using carefully positioned measuring locations and the latest coordinate measuring tools. Multiple coats of different colors of paint can also be used to quickly identify areas of high wear. Wear-life data gathered this way should be used as estimates only.

During field operation, it is possible to run parts to wear failure. This can establish the useful life and illustrate wear patterns that could show possible system anomalies. These actual life values are preferred but are valid only for the pump service conditions at the time of the test. When solids size, concentration, or rate of flow change significantly, the wear parts should be monitored and a satisfactory average life computed.

12.6.6 Instrumentation

12.6.6.1 Flow meters

Magnetic flow meters are preferred but a calibrated bend meter with suitable piezometer flush points also can be acceptable. Wherever possible, the magnetic flow meter should be checked on water against an orifice or other means of calibration.

12.6.6.2 Power measurement

The preferred method is a strain gauge-type torque bar or two-element-type wattmeter on the electric motor when there is a direct drive.

12.6.6.3 Head measurement

Perform according to *ANSI/HI 1.6* except that all tapping points should be equipped with sand traps, gauge savers and/or purge water flushing to protect the instruments.

12.6.6.4 Specific gravity measurement

This can be carried out by nuclear densitometer or inverted U-tube recording the difference in pressure over the sum of the upcoming and downgoing legs divided by two. The theory of the latter is outlined in Appendix B.1, reference 5.

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Appendix A:

Equipment data sheets

Equipment Data Sheet
Rotodynamic Slurry Pump
Metric Units

Insert Company Logo Here

User/ Equipment Descriptive Information

User Name:		
Title of Equipment:		
Specification No.:		
Quantity:	No. Working:	No. Standby:
Motor Req'd.: ☐ Yes ☐ No	Provided by:	
Mounted by:	Tag No.:	
EQUIPMENT No.(s):		
Material Requisition No.:		
System Description:		

Operating Conditions (User to complete) - EACH PUMP

Fluid Description:			
Chemical Characterisitcs of Slurry Mixture:			
Description: pH:	% Chlorides		
Solids Particulars:			
Max Particle Size (Microns):			
Concentration of Solids: % Cw		Solids d50: µm	
Liquids SG:	Solids SG:	Solids d85: µm	
Boiling Pt. Elevation: ˚C	Visc. @ Temp. mm²/s		
	Normal	Minimum	Design
Flow @ Temp (m³/hr):			
% Time			
Slurry SG:			
Operating Temp (˚C):			
Suction Pressure (kPa - gauge):			
Discharge Pressure (kPa - gauge):			
Diff Head (m):			
Vap Pressure @ Temp (kPa[abs]@˚C):			
NPSHA @ Design Flow & Temp (m):			

Supplier Information / Data

Manufacturer:
Size & Type:
Serial No.(s):

Construction (Supplier to complete)

Nozzles	Size	Rating	Facing	Location
Suction				
Discharge				

Pump Type:				
Jacketed:	Yes ☐	No ☐		
Pressure:	Max. Allowable	kPa - gauge		˚C
	Hydro Test	kPa - gauge		˚C
Connections:	☐ Casing Drain	☐ Casing Vent		
	☐ Gauge Connection	☐ Valved/ Plugged		
Impeller Type:	☐ Closed	☐ Semi-Open	☐ Open	
Impeller Dia. (mm):	(Normal)	(Min.)	(Design)	
Impeller Attachment:	☐ Key	☐ Threaded		
Casing Mount:	☐ Foot	☐ Centerline	☐ Vertical	☐ Inline
Casing Split:	☐ Axial	☐ Radial		
Bearings:	Thrust: (Mfr/Model)	(Location)		
	Radial: (Mfr/Model)	(Location)		
Lubrication:	☐ Oil Mist	☐ Grease	☐ Oil	
Coupling: Mfr/ Model:		Driver Half Mtd. By:		

Performance (Supplier to complete)

Proposed Curve No.:			
No. Stages:		Speed:	RPM
NPSHR:	m	Eff: %	
Seal Flush:		l/min @	kPa - gauge
Stuff Box Press:			kPa - gauge
Min. Continuous Flow:			l/min
Design Power:			kW
Max Power Design Impeller:			kW
Max Head Design Impeller:			m
Rotation Facing Coupling:		Cw ☐	☐ Ccw

☐ Seals (User ☐ or Supplier to complete, please indicate)			
Bearing Housing:			
Drive End:	Mfr:/Model		Dia. (mm)
Non-Drive End:	Mfr:/Model		Dia. (mm)
Shaft/ Wet-End:			
Type Stuffing Box:	☐ Packing Material Type Size & No. of Rings	☐ Mech Seal Mfr: Size Face Material Carrier Type	

Driver (Supplier to complete)		
Driver Type: ☐ Electric	☐ Engine:	☐ Other (Specify):
Duty Rated: kW	Speed: RPM	Mfr:
Volts/ph/Hz:	Frame:	No. of Poles:
Drive Type:	☐ VSD ☐ Fixed ☐ Gear ☐ V Belt	☐ Direct

Test	Test Code	Witness
☐ Performance		☐
☐ Hydro		☐
☐ NPSH		☐
☐ Shop Inspection		☐
☐ Dismantle and Inspect after Test		☐

Shipping (Supplier to Complete)			
Pump:	kg	Driver:	kg
Base Plate:	kg		
Gearbox/Coupling:		kg	
Total Shipping wt.:		kg	

Additional Notes (User or Supplier to complete)

Rev.No.:	0	1	2	3
Prepared by & Date:				
Reviewed by & Date:				
User Approval & Date:				
Supplier Approval & Date:				
Status:				

Materials (Supplier to complete)
Casing:
Impeller:
Volute Liner:
Casing Liner:
Impeller Wear Rings:
Casing Wear Rings:
Shaft:
Shaft Sleeve:
Throat Bushing:
Stuffing Box:
Lantern Ring:
Casing Bolts:
Packing:
Casing Gasket:
Coupling Halves:
Base Plate:
Bearing Housing:
Safety Guard:

Insert Outline Drawing of Pump and Pump Driver Here

Equipment Data Sheet
Rotodynamic Slurry Pump
US Units

Insert Company Logo Here

User/ Equipment Descriptive Information

User Name:		
Title of Equipment:		
Specification No.:		
Quantity:	No. Working:	No. Standby:
Motor Req'd.: ☐ Yes ☐ No	Provided by:	
Mounted by:	Tag No.:	
EQUIPMENT No.(s):		
Material Requisition No.:		
Systerm Description		

Operating Conditions (User to complete) - EACH PUMP

Fluid Description:			
Chemical Characterisitcs of Slurry Mixture:			
Description: pH:		% Chlorides	
Solids Particulars:			
Slurry Abrasivity (Miller Number):			
Max Particle Size (Microns):			
Concentration of Solids:	% Cw	Solids d50:	µm
Liquor SG:	Solids SG:	Solids d85:	µm
Boiling Pt. Elevation: ˚F	Visc. @ Temp.	cSt	
	Normal	Minimum	Design
Flow @ Temp (US gpm):			
% Time:			
Slurry SG:			
Operating Temp (˚F):			
Suction Pressure (psig):			
Discharge Pressure (psig):			
Diff Head (ft):			
Vap Pressure @ Temp (psia @ ˚F):			
NPSHA @ Design Flow & Temp (ft):			

Supplier Information / Data

Manufacturer:
Size & Type:
Serial No.(s):

Construction (Supplier to complete)

Nozzles	Size	Rating	Facing	Location
Suction				
Discharge				
Pump Type:				
Jacketed:	Yes ☐	No ☐		
Pressure:	Max. Allowable	psig		˚F
	Hydro Test	psig		˚F
Connections:	☐ Casing Drain		☐ Casing Vent	
	☐ Gauge Connection		☐ Valved/ Plugged	
Impeller Type:	☐ Closed	☐ Semi-Open	☐ Open	
Impeller Dia. (in.):	(Normal)	(Min.)	(Design)	
Impeller Attachment:	☐ Key	☐ Threaded		
Casing Mount:	☐ Foot	☐ Centerline	☐ Vertical	☐ Inline
Casing Split:	☐ Axial	☐ Radial		
Bearings:	Thrust: (Mfr/Model)		(Location)	
	Radial: (Mfr/Model)		(Location)	
Lubrication:	☐ Oil Mist	☐ Grease	☐ Oil	
Coupling: Mfr/ Model:		Driver Half Mtd. By:		

Performance (Supplier to complete)

Proposed Curve No.:		
No. Stages:	Speed:	RPM
NPSHR: ft.	Eff:	%
Seal Flush:	gpm @	psig
Stuff Box Press:		psi(a)
Min. Continuous Flow:		gpm
Design Power:		Horsepower
Max Power Design Impeller:		Horsepower
Max Head Design Impeller:		ft.
Rotation Facing Coupling:	☐ Cw	☐ Ccw

☐ Seals (User ☐ or Supplier to complete, please indicate)		
Bearing Housing:		
Drive End:	Mfr:/Model	Dia. (in)
Non-Drive End:	Mfr:/Model	Dia. (in)
Shaft/ Wet-End:		
Type Stuffing Box:	☐ Packing Material Type: Size & No. of Rings:	☐ Mech Seal Mfr: Face Materials Carrier Type

Driver (Supplier to complete)		
Driver Type: ☐ Electric	☐ Engine:	☐ Other (Specify):
Duty Rated: HP	Speed: RPM	Mfr:
Volts/ph/Hz:	Frame:	No. of Poles:
Drive Type:	☐ VSD ☐ Fixed ☐ Gear ☐ V Belt	☐ Direct

Test	Test Code	Witness
☐ Performance		☐
☐ Hydro		☐
☐ NPSH		☐
☐ Shop Inspection		☐
☐ Dismantle and Inspect after Test		☐

Shipping (Supplier to Complete)			
Pump:	lb.	Driver:	lb.
Base Plate:	lb.		
Gearbox/Coupling:		lb.	
Total Shipping wt.:		lb.	

Additional Notes (User or Supplier to complete)

Rev.No.:	0	1	2	3
Prepared by & Date:				
Reviewed by & Date:				
User Approval & Date:				
Supplier Approval & Date:				
Status:				

Materials (Supplier to complete)
Casing:
Impeller:
Volute Liner:
Casing Liner:
Impeller Wear Rings:
Casing Wear Rings:
Shaft:
Shaft Sleeve:
Throat Bushing:
Stuffing Box:
Lantern Ring:
Casing Bolts:
Packing:
Casing Gasket:
Coupling Halves:
Base Plate:
Bearing Housing:
Safety Guard:

Insert Outline Drawing of Pump and Pump Driver Here

Appendix B:

Source material and references

Published papers are available through the organizations noted, in accordance with copyright laws.

B.1 Source material

ANSI/HI Standards ANSI/HI 1.1 – 1.6; ANSI/HI 9.8; ANSI/HI 9.6

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B.3 Standards organizations

The following list includes website addresses available at the time this document was published. Additional organizations and standards may be found by searching the Internet.

Australia – www.standards.com.au
Canada – www.csa-international.org
China – www.csbts.cn.net
Europe – www.cenorm.be
France – www.afnor.fr
Germany – www.din.de
International – www.iso.org
Japan – www.jisc.org
Netherlands – www2.nen.nl
Norway – www.standards.no
Russia – www.gost.ru
Sweden – www.sis.se
UK – www.bsi-global.com
USA – AISI – www.steel.org
USA – ANSI – www.ansi.org
USA – ASTM – www.astm.org
USA – ASME – www.asme.org
USA – SAE – www.sae.org

Appendix C:

Conversions

C.1 Unit conversions

Symbol	Term	To convert from US Customary Unit (USCS)	Abbr	into Metric unit	Abbr	Multiply by Conversion factor[a, b]
A	Area	square inches	in^2	square millimeter	mm^2	645.2
D	Diameter	Inches	in	millimeter	mm	25.4
η (eta)	Efficiency	Percent	%	percent	%	1
G	Gravitational acceleration	feet/second squared	ft/s^2	meter/second squared	m/s^2	0.3048
H	Head	feet	ft	meter	m	0.3048
H_w	Total head	feet	ft	meter	m	0.3048
I	Static lift	feet	ft	meter	m	0.3048
n	Speed	revolutions/minute	rpm	revolutions/minute	rpm	1
NPSHA	Net positive suction head available	feet	ft	meter	m	0.3048
NPSHR	Net positive suction head required	feet	ft	meter	m	0.3048
N_s	Specific speed $N_s = nQ^{1/2}/H^{3/4}$	not used[a]	-	not used[a]	-	1.162[b]
ν (nu)	Kinematic viscosity	feet squared/second	ft^2/s	millimeter squared/sec	mm^2/s	92,903
π	pi = 3.1416	dimensionless	-	dimensionless	-	1
P	Pressure	pounds/square inch	psi	kilopascal	kPa	6.895
p	Power	horsepower	hp	kilowatt	kW	0.7457
Q	Capacity	cubic feet/second	ft^3/s	cubic meter/hour	m^3/h	101.94
Q	Capacity	US gallons/minute	US gpm	cubic meter/hour	m^3/h	0.2271
RM	Linear model ratio	dimensionless	-	dimensionless	-	1
RT	Radial thrust	pounds (force)	lbf	Newton	N	4.448
ρ (rho)	Density	pound mass/cubic foot	lbm/ft^3	kilogram/cubic meter	kg/m^3	16.02
S_m	Specific gravity (mixture)	dimensionless	-	dimensionless	-	1
T	Temperature	degrees Fahrenheit	°F	degrees Celsius	°C	(°F–32) × (5/9)
τ (tau)	Torque	pound-feet	lb-ft	Newton-meter	N•m	1.357
U	Residual unbalance	ounce-inches	oz-in	gram-centimeter	g-cm	72

Symbol	Term	To convert from US Customary Unit (USCS)	Abbr	into Metric unit	Abbr	Multiply by Conversion factor[a, b]
V	Velocity	feet/second	ft/s	meter/second	m/s	0.3048
ϕ (phi)	Velocity in vibration	inches/second	in/s	millimeters/second	mm/s	25.4
X	Exponent	none	none	none	none	1
Z	Elevation gauge distance above or below datum	feet	ft	meter	m	0.3048
C_v	Concentration by volume	percentage	%	percentage	%	1
C_w	Concentration by weight	percentage	%	percentage	%	1
ID	Inside diameter of pipe	inches	in	millimeter	mm	25.4

[a] Assumed dimensionless

[b] Where US units are ft, USGPM, or RPM, then the corresponding Metric units are m, m^3/h, or RPM

C.2 Metal specification equivalents

A list of international specifications for commonly used pump construction materials is outlined below. The most current revision should be used, so the date code is not included. Note that a particular material type may have multiple designations or grades within a specification. This information is included for reference only and does not constitute a recommendation for material use in any application. The complete list can be found in the *ASM International Worldwide Guide to Equivalent Irons and Steels*.

Material Name					Typical Application			
UTS (Ultimate Tensile) in MPa, YS (Yield) in MPa, El (Elongation) in %, Hardness as indicated								
ISO	DIN	ASTM	ASME	SAE	AS	NEN	JIS	GB
International	Germany	USA	USA	USA	Australia	Netherland	Japan	China
ANSI	UNS	NS	SIS					
USA	USA	Norway	Sweden					
Cast Iron, Ductile, 60-40-18					Plates, bases, clear-water pumps			
UTS: 414, YS: 276, El: 18, Hardness: 170 HB or 143 >187 Brinell								
ISO	DIN	ASTM	ASME	SAE	AS	NEN	JIS	GB
1083	1693	A536	SA395	J434	1831	2733	G5502	1384
Cast Iron, Gray, 30					Pedestals, housings, and mechanical parts			
UTS: 207 min, Hardness: 187 > 255 HB								
ISO	DIN	ASTM	ASME	SAE	AS	NEN	JIS	GB
1083	1693	A48	SA278	J434	1831	2733	G5502	1384
Cast Iron, Gray 45					Casings and pressure parts			
UTS: 310 min, Hardness: 207 > 269 HB								
ISO	DIN	ASTM	ASME	SAE	AS	NEN	JIS	GB
185	1691	A278	SA278	n/a	1830	6002A	G5501	9439

Material Name					Typical Application			
Steel Bar, 4140 Chromium Molybdenum					Shafts, bearing housing nuts and bolts			
UTS*: 900, YS*: 800, El: 16, Hardness*: 269 > 321 Brinell (*Varies with diameter)								
ISO	DIN	ASTM	ANSI	SAE	AS	NS	JIS	GB
R683-4	1652	A29	4140	J404	1444	CrMolV	G4107	3077
Stainless, Austenitic, Cn7M(S) (300 series)					Impellers, casings, stuffing boxes			
UTS: 425, YS: 170, El: 35, Hardness: 180 HB								
ISO	DIN	ASTM	ASTM	UNS	AS	NEN	JIS	GB
n/a	SEW410	A744	A743	J92810	n/a	n/a	G5121	n/a
Stainless, Martensitic, CA15 (400 series)					Impellers, casings, stuffing boxes			
UTS: 425, YS: 170, El: 35, Hardness: 180 HB								
ISO	DIN	ASTM	ASTM	UNS	AS	NEN	JIS	GB
n/a	10213	A487	A743M	J91150	2074	n/a	G5121	ZQ4299
Stainless, Duplex, CD4MCu					Sleeves, bushings, stuffing boxes			
UTS: 690, YS: 485, El: 16, Hardness: 200 HB								
ISO	DIN	ASTM	ASTM	UNS	AS	NEN	JIS	GB
n/a	10213	A744	A890	n/a	n/a	n/a	G5121	2100
Abrasion-Resistant Iron, Ni-Cr					Impellers, casings, stuffing boxes			
UTS: varies, Hardness: 450 > 650 HB								
ISO	DIN	ASTM	ASTM	SAE	AS	SIS	JIS	GB
n/a	WNr0.9630	A532	A518	n/a	n/a	140457	G5503	8491

C.3 Specific gravity of various materials

Mineral	Minimum	Maximum
Aluminum	2.70	
Anglesite	6.10	6.40
Apatite	3.15	3.27
Asbestos	2.20	3.30
Asurite	3.70	3.90
Barite	4.30	4.70
Beryl	2.68	2.76
Bornite	4.90	5.40
Braunite	4.70	4.90
Calamine	3.30	3.50
Calcite (Limestone)	2.71	Varies
Cassiterite	6.80	7.00
Cerussite	6.50	6.60

Mineral	Minimum	Maximum
Chalcocite	5.50	5.80
Chalcopyrite	4.10	4.30
Chromite	4.30	4.60
Chrysocolla	2.00	2.20
Cinnabar	8.00	8.20
Clay	2.70	
Coal, Anthracite	1.30	1.70
Coal, Bituminous	1.10	1.50
Coke	1.50	
Copper (Native)	8.50	9.00
Corundum	3.90	4.10
Covellite	4.60	
Cuprite	5.70	6.10
Cyanite	3.50	3.70
Diamond, Black	2.75	3.42

Mineral	Minimum	Maximum
Diamond, Gem Grade	3.50	3.56
Enargite	4.40	
Feldspar	2.70	
Ferberite	7.50	
Flue Dust, Blast Furnace	3.50	
Flourite	3.00	3.20
Franklinite	5.00	5.20
Galena	7.30	7.60
Garnet	3.50	4.30
Garnierite	2.30	2.80
Gold	15.60	19.30
Granite	2.72	
Gypsum	2.31	2.33
Hematite	4.90	5.30
Hubnerite	6.70	7.30
Ilmenite	4.30	5.50
Iron	7.30	7.80
Lime (Calcium Oxide)	3.40	
Limonite	3.60	4.00
Magnetite	4.96	5.18
Malachite	3.80	3.90
Mica	2.80	2.90
Millerite	5.30	5.90
Molybdenite	4.70	4.80
Nephilite	2.55	2.65
Nicolite	7.30	7.80
Pentlandite	4.60	5.10
Phosphate Rock	3.15	Varies
Platinum	14.00	21.45
Psilomelane	3.70	4.70
Pyrite	4.95	5.17
Pyrolusite	4.73	4.86
Pyrrhotite	4.50	4.60
Quartz	2.65	2.66
Rhodochrosite	3.30	3.76
Rutile	4.20	4.30
Salt Water	1.03	
Scheelite	5.90	6.20
Shale	2.38	2.75
Siderite	3.83	3.88

Mineral	Minimum	Maximum
Silica (SiO_2)	2.65	
Silver	10.00	12.00
Slag – Copper	3.72	
Slate	2.45	2.70
Smithsonite	4.30	4.45
Sphalerite	3.90	4.20
Spodumene	3.10	3.20
Stannite	4.30	4.50
Taconite Tailings	2.70	2.80
Talc	2.70	2.80
Tetrahedrite	4.40	5.10
Vanadinite	6.60	7.10
Wad	3.00	4.26
Willemite	3.90	4.30
Witherite	4.20	4.30
Wolframite	7.10	7.50
Zincite	5.40	5.70

Appendix D:

Index

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

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

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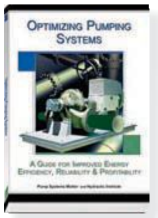
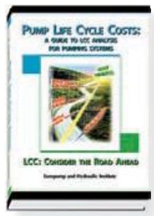
Creating Pump Standards Since 1917

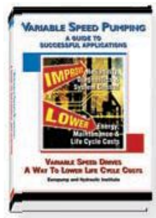
Available at eStore.Pumps.org



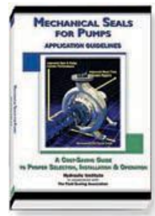
Optimizing Pumping Systems: A Guide to Improved Efficiency, Reliability and Profitability



Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems



Variable Speed Pumping: A Guide to Successful Applications



Mechanical Seals for Pumps: Application Guidelines

ANSI/HI
Pump Standards

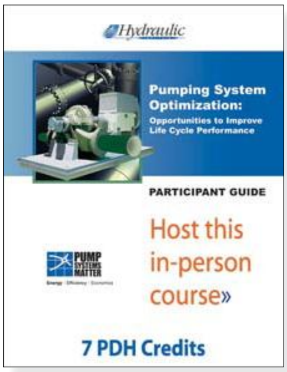
Individual Standards

- Hardcopy
- Downloadable

Complete Set — Hardcopy

ANSI/HI Pump Standards on CD — complete set

ANSI/HI Pump Standards by Subscription



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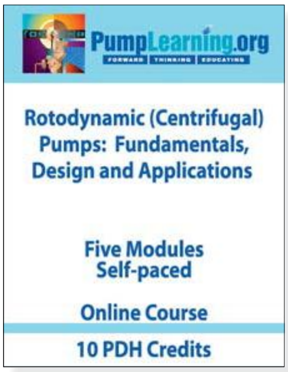

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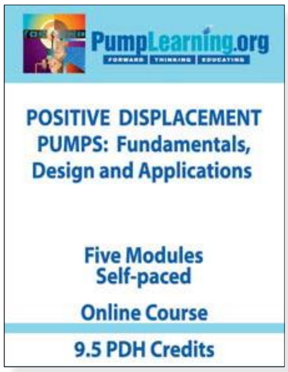

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Click on the text or image to go to the eStore page for the product of your choice.

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