ANSI/HI 12.1–12.6-2011

Approved American National Standard

ANSI

American National Standard for

Rotodynamic (Centrifugal) Slurry Pumps

for Nomenclature, Definitions, Applications, and Operation

ANSI/HI 12.1–12.6-2011

Hydraulic Institute

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

This page intentionally blank.

ANSI/HI 12.1–12.6-2011

American National Standard for

Rotodynamic (Centrifugal) Slurry Pumps

for Nomenclature, Definitions, Applications, and Operation

Sponsor

Hydraulic Institute

www.Pumps.org

Approved May 19, 2011

American National Standards Institute, Inc.

American National Standard

Approval of an American National Standard requires verification by ANSI that the requirements for due process, consensus and other criteria for approval have been met by the standards developer.

Consensus is established when, in the judgement of the ANSI Board of Standards Review, substantial agreement has been reached by directly and materially affected interests. Substantial agreement means much more than a simple majority, but not necessarily unanimity. Consensus requires that all views and objections be considered, and that a concerted effort be made toward their resolution.

The use of American National Standards is completely voluntary; their existence does not in any respect preclude anyone, whether he has approved the standards or not, from manufacturing, marketing, purchasing, or using products, processes, or procedures not conforming to the standards.

The American National Standards Institute does not develop standards and will in no circumstances give an interpretation of any American National Standard. Moreover, no person shall have the right or authority to issue an interpretation of an American National Standard in the name of the American National Standards Institute. Requests for interpretations should be addressed to the secretariat or sponsor whose name appears on the title page of this standard.

CAUTION NOTICE: This American National Standard may be revised or withdrawn at any time. The procedures of the American National Standards Institute require that action be taken periodically to reaffirm, revise, or withdraw this standard. Purchasers of American National Standards may receive current information on all standards by calling or writing the American National Standards Institute.

Published By

Hydraulic Institute
6 Campus Drive, First Floor North
Parsippany, NJ 07054-4406

www.Pumps.org

Printed in the United States of America

ISBN 978-0-9824330-6-5

Recycled paper

Contents

Figures

Tables

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

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

Revisions

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

Units of measurement

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

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

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.

4B Engineering and Consulting, LC
AR Wilfley & Sons, Inc.
Brown and Caldwell
DuPont
ekwestrel corp
Fluid Sealing Association
GIW Industries
Healy Engineering, Inc.
ITT – Industrial Process
ITT RCW
J.A.S. Solutions Ltd.
John Anspach Consulting
John Crane Inc.
Kemet Inc.
Las Vegas Valley Water District
Malcolm Pirnie
Moving Water Industries
Orange County Sanitation District
Peerless Pump
Powell Kugler, Inc.
Sulzer Pumps US Inc.
Weir Floway, Inc.
Weir Minerals North America

Committee list

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

Chair - Graeme R. Addie, GIW Industries, Inc.
Vice-Chair - Aleksander S. Roudnev, Weir Minerals North America

Committee Members	Company
Henri V. Azibert	A.W. Chesterton Company
Charles A. Cappellino	ITT - Industrial Process
Ralph P. Gabriel	John Crane Inc.
Greg S. Highfill	A.R. Wilfley & Sons, Inc.
Michael Mueller	Flowserve Corporation

Alternates

David C. Krebs	A.R. Wilfley & Sons, Inc.
Gary C. Leander	Weir Minerals North America
Robert J. Visintainer	GIW Industries, Inc.

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 used for abrasive slurries, 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 normative and sets out requirements, recommendations, and statements to define, select, apply, operate, and maintain slurry pumps. Requirements convey criteria to be fulfilled if compliance with the document is to be claimed and from which no deviation is permitted. Recommendations convey that, among several possibilities, one is particularly suitable, without excluding or prohibiting others.

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 often abrasive and, if not considered, 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, which is necessary to select, apply, operate, and maintain slurry pumps of different designs and materials of construction.

12.1.2 Pump types and nomenclature

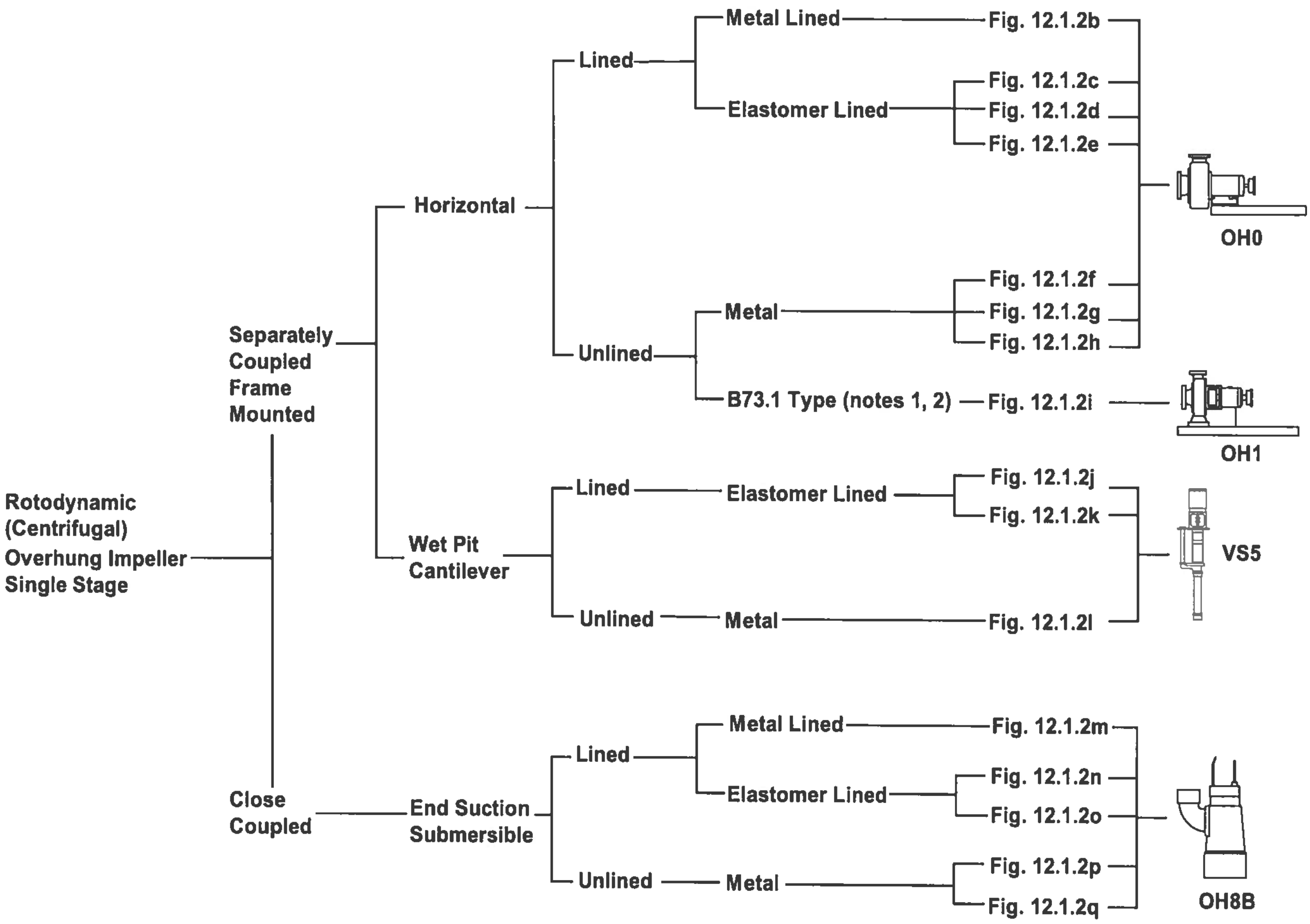
Figure 12.1.2a shows classifications of slurry pumps based on mechanical configuration. Figures 12.1.2b through 12.1.2q show typical constructions commonly used for each pump type. Lowercase letter part designations are for different manufacturer variants of the same type. Other variations are also acceptable.

While there are no rigid rules about where different mechanical configurations are to be applied, initial cost, wear parts (maintenance) cost, and arrangement 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 as class 3 and class 4 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 (described as class 3 in Section 12.3.4.2) but are more widely used in the lighter-class wear services (described as class 1 and class 2 in Section 12.3.4.2) 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 used in cleanup services, but there may be areas where they are used as process pumps. These are also limited to smaller sizes.



NOTES 1: B73.1 type pumps used in slurry services normally require greater than normal thicknesses on parts exposed to slurry.
2: Foot mounted.

Figure 12.1.2a — Rotodynamic (centrifugal) slurry pump types

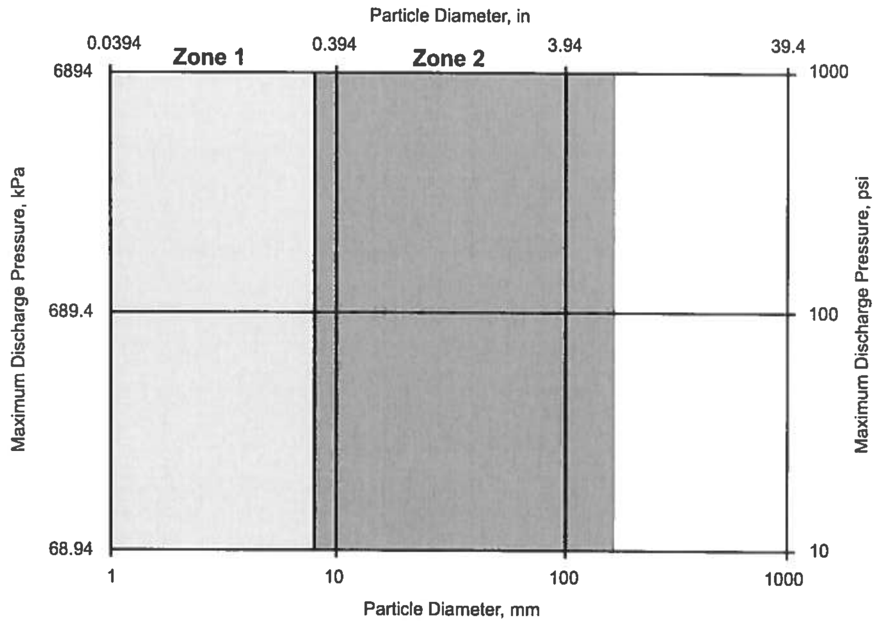
12.1.3 Definition of slurry

Slurry is a mixture of solids (specific gravity typically 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 used 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, specialized chemical particulates, 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 suspended particles. Slurry pumps vary in construction depending on the properties of the slurry to be pumped; they are usually more robust than the pumps used in clean-liquid services, and they often have replaceable wear parts. Figure 12.1.4 shows how various materials are typically used in slurry pumps. Maximum solid size is limited by pump size and passage. This is all covered in more detail in Section 12.3.4.1.

The maximum discharge pressures noted are for one or more single-stage pumps, operating in series, at one location. This shows no limitation for elastomer-lined pumps. The recommended heads per stage are shown in Table 12.3.5a located in Section 12.3.5 of this document.



Zone 1: Elastomer or metal

Zone 2: Metal (most commonly), see Section 12.3.7

Figure 12.1.4 — Typical material types and discharge pressure for particle size

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 use 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

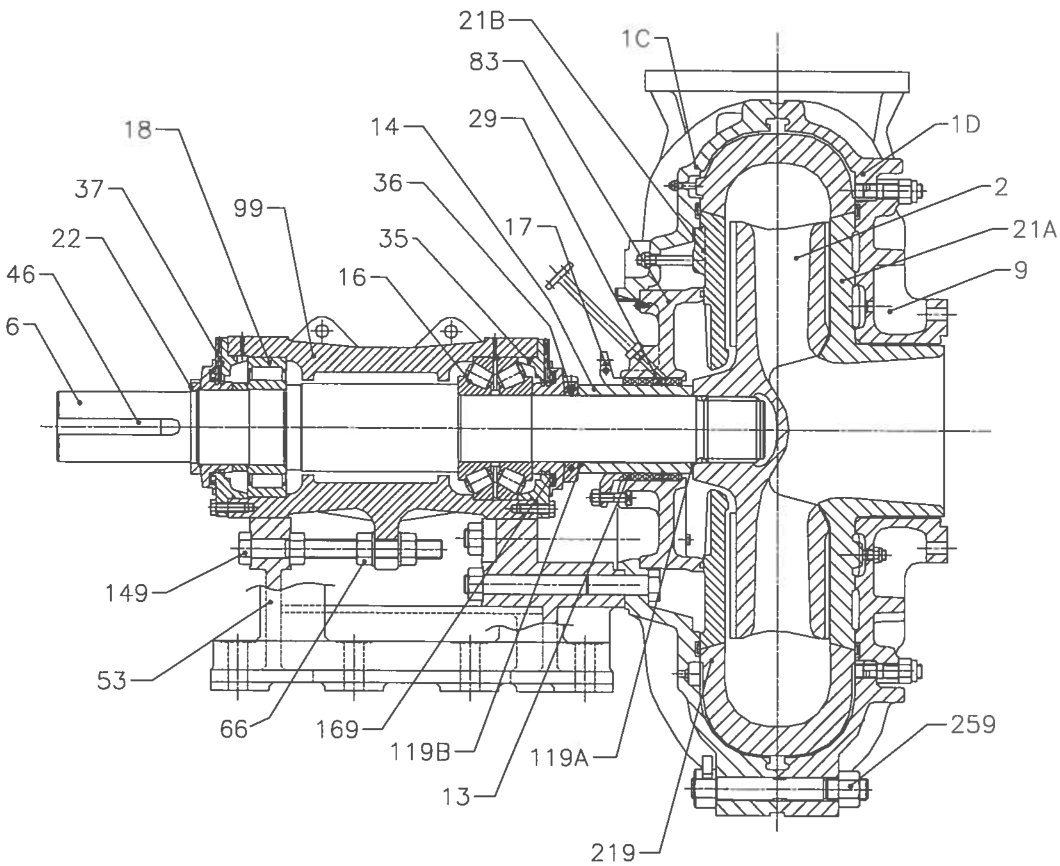
The construction drawings on the following pages are intended to provide a means for identifying the various pump types covered by the ANSI/HI Standards and to serve as the basis for a common language between the purchaser, manufacturer, and specification writer.

With a few exceptions, the individual part names on these drawings are numbered such that rotating parts have been assigned even numbers, while nonrotating 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.).

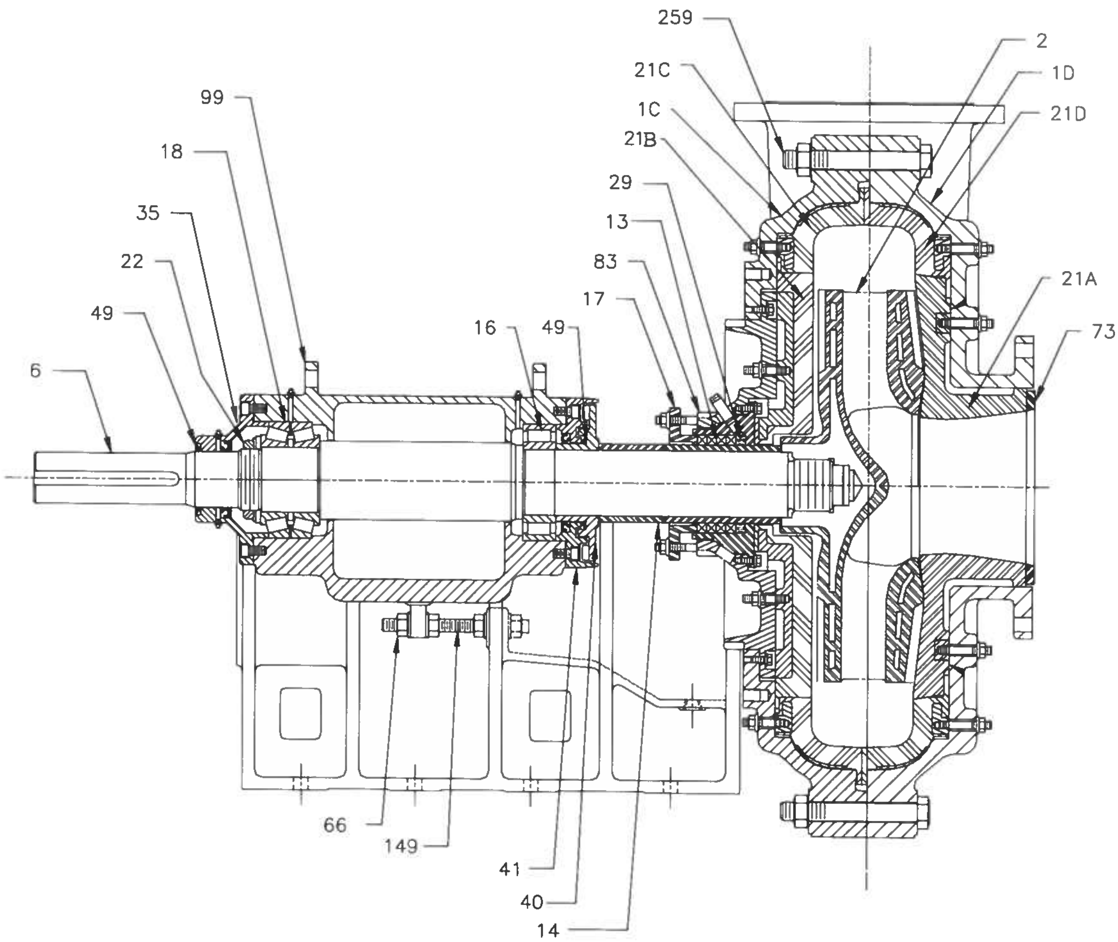
12.1.12 Part names

Tables 12.1.12a and 12.1.12b 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 12.1.2b through 12.1.2q. The part numbers used are for illustration purposes only and may vary with different manufacturers.



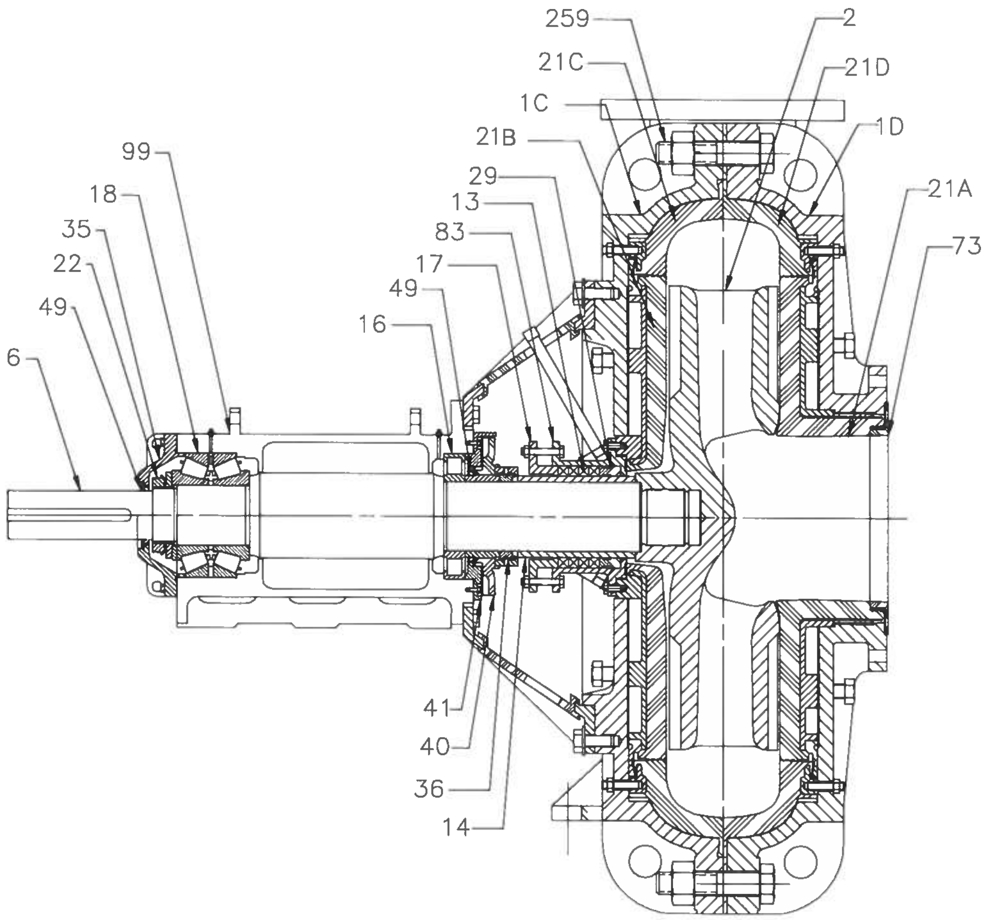
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
9	Cover, suction	13	Packing
14	Sleeve, shaft	16	Bearing, inboard
17	Gland	18	Bearing, outboard
21A	Liner, suction cover	21B	Liner, stuffing box cover
22	Locknut, bearing	29	Ring, lantern
35	Cover, bearing, inboard	36	Collar, release
37	Cover, bearing, outboard	46	Key, coupling
53	Base	66	Nut, shaft adjusting
83	Stuffing box	99	Housing, bearing
119A	O-ring	119B	O-ring
149	Screw, impeller adjusting	169	Seal, bearing housing
219	Liner, casing	259	Bolt, casing

Figure 12.1.2b — Overhung impeller, separately coupled, single-stage, frame-mounted, metal-lined pump (OH0)



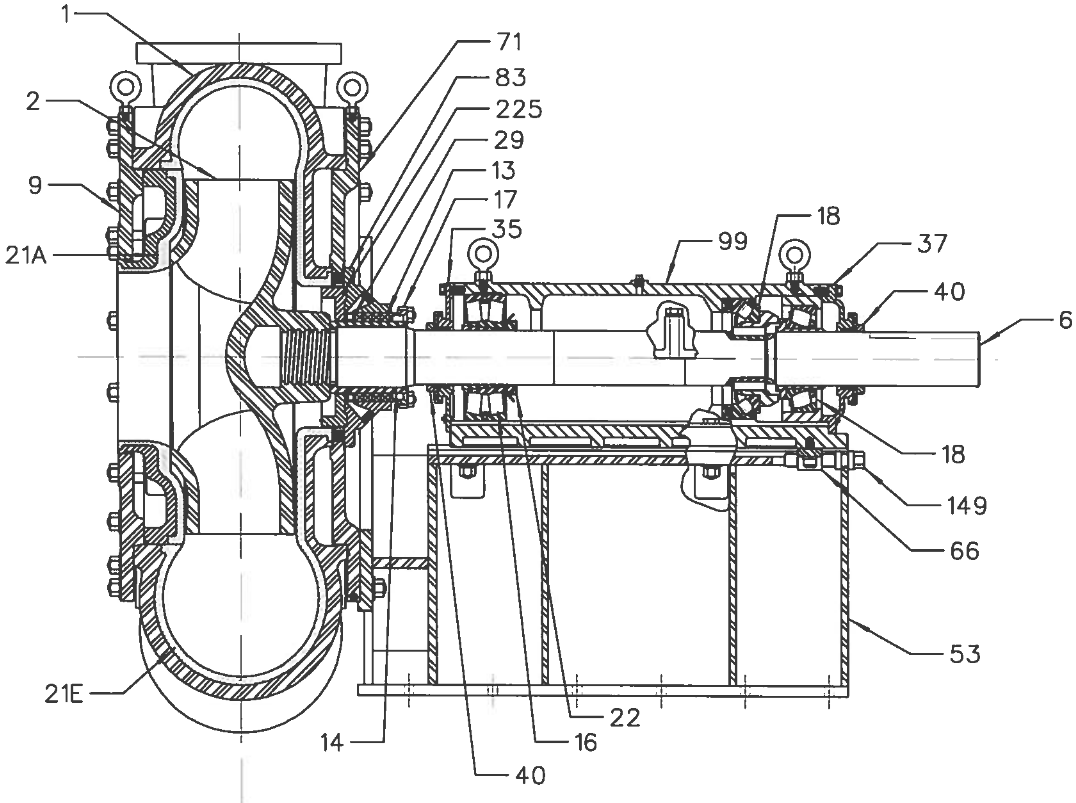
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21B	Liner, stuffing box cover	21C	Liner, gland half
21D	Liner, suction half	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard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

Figure 12.1.2c — Overhung impeller, separately coupled, single-stage, frame-mounted, elastomer-lined pump (OH0)



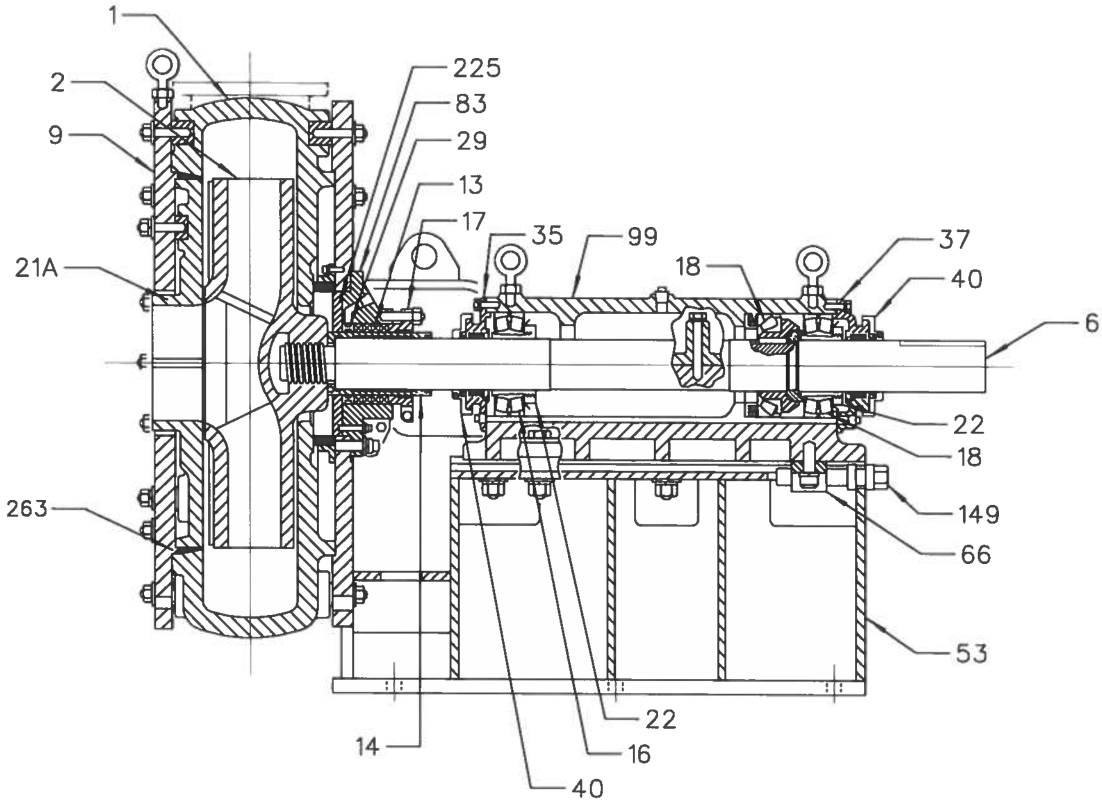
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21B	Liner, stuffing box cover	21C	Liner, gland half
21D	Liner, suction half	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
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 12.1.2d — Overhung impeller, separately coupled, single-stage, frame-mounted, elastomer-lined pump, adjustable sideliners (OH0)



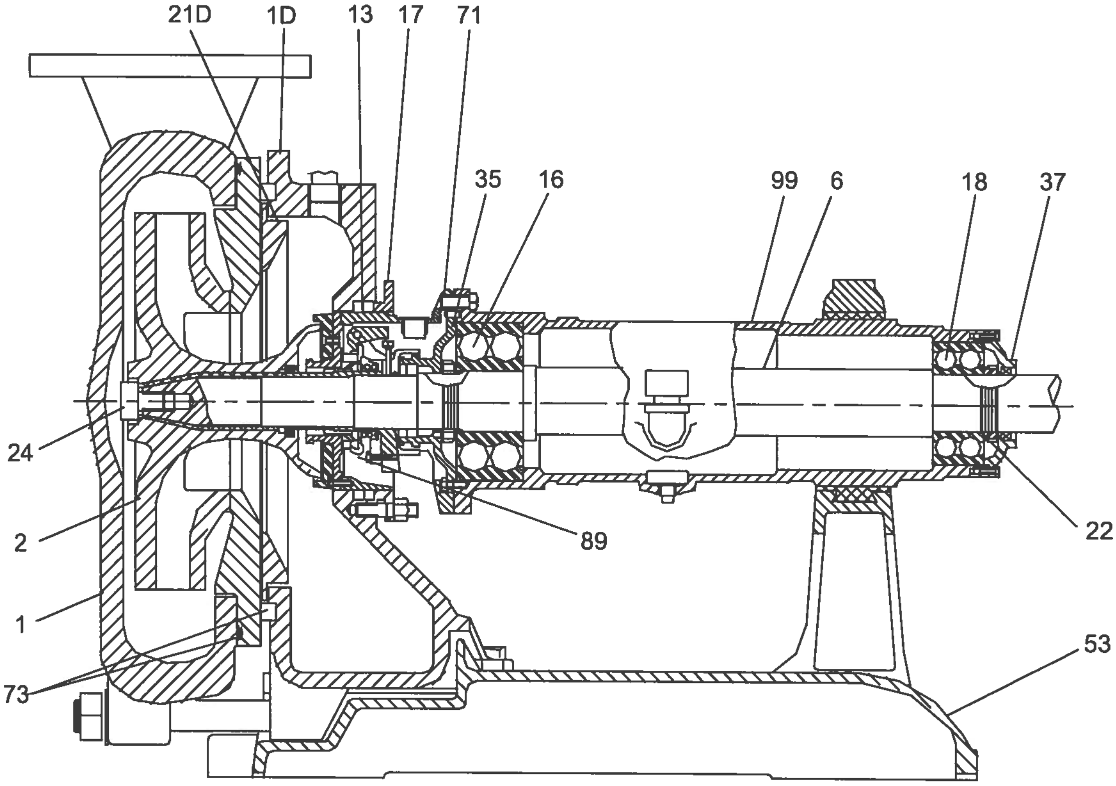
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21E	Liner, vulcanized	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
37	Cover, bearing, outboard	40	Deflector
53	Base	66	Nut, shaft adjusting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

Figure 12.1.2e — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, vulcanized-elastomer-lined pump (OH0)



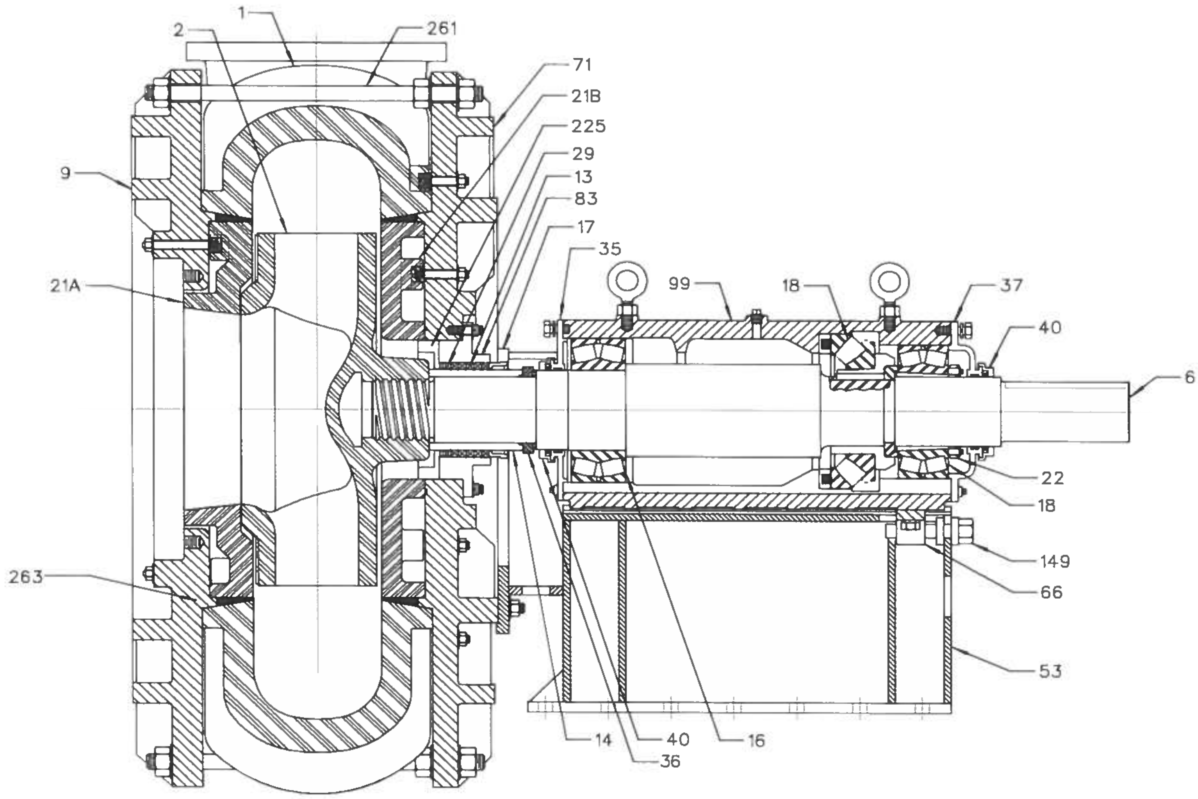
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
22	Locknut, bearing	29	Ring, lantern
35	Cover, bearing, inboard	37	Cover, bearing, outboard
40	Deflector	53	Base
66	Nut, shaft adjusting	83	Stuffing box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

Figure 12.1.2f — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, metal, unlined casing pump (OH0)



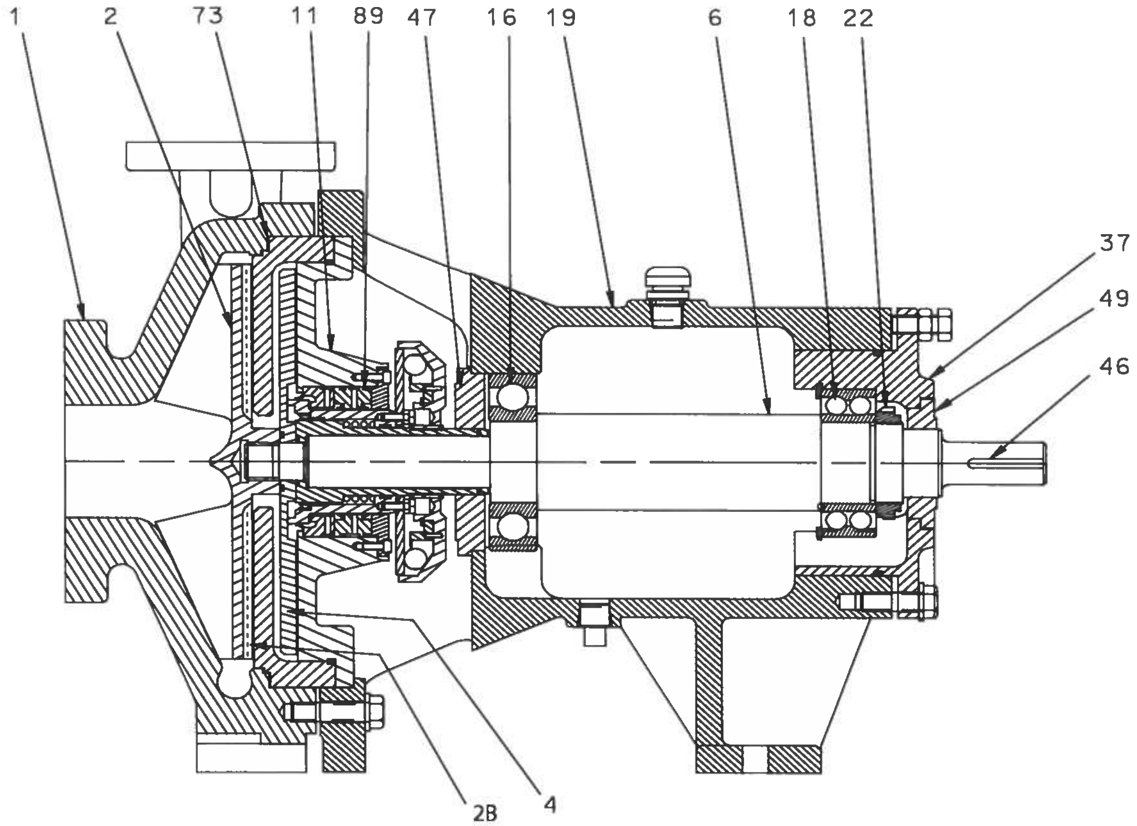
1	Casing	1D	Casing, suction half
2	Impeller	6	Shaft
13	Packing	16	Bearing, inboard
17	Gland	18	Bearing, outboard
21D	Liner, suction half	22	Locknut, bearing
24	Nut, impeller	35	Cover, bearing, inboard
37	Cover, bearing, outboard	53	Base
71	Adapter	73	Gasket
89	Seal	99	Housing, bearing

Figure 12.1.2g — Overhung impeller, separately coupled, single-stage, frame-mounted, side inlet, metal, unlined casing pump (OH0)



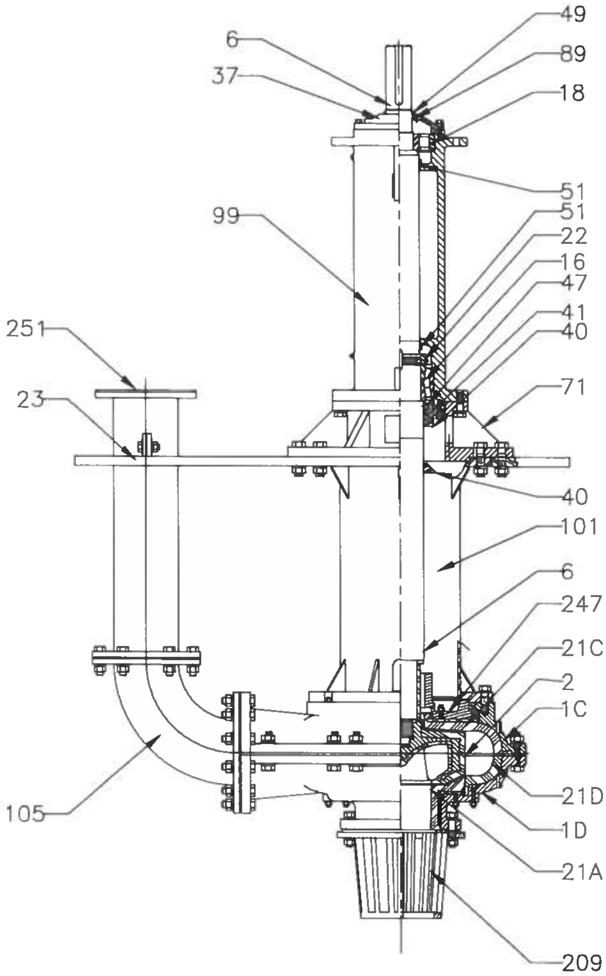
1	Casing	2	Impeller
6	Shaft	9	Cover, suction
13	Packing	14	Sleeve, shaft
16	Bearing, inboard	17	Gland
18	Bearing, outboard	21A	Liner, suction cover
21B	Liner, stuffing box cover	22	Locknut, bearing
29	Ring, lantern	35	Cover, bearing, inboard
36	Collar, release	37	Cover, bearing, outboard
40	Deflector	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 12.1.2h — Overhung impeller, separately coupled, single-stage, frame-mounted, end suction, metal, tie bolt plate construction pump (OH0)



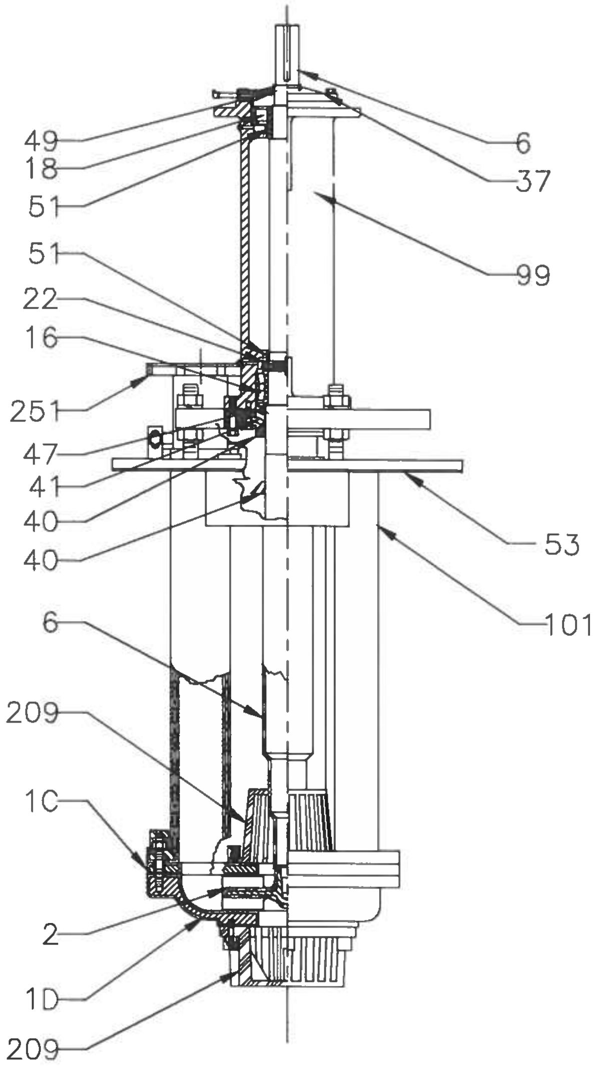
1	Casing	2	Impeller
2B	Expelling vanes	4	Expeller
6	Shaft	11	Cover, stuffing box or seal chamber
16	Bearing, inboard	18	Bearing, outboard
19	Frame	22	Locknut, bearing
37	Cover, bearing, outboard	46	Key, coupling
47	Seal, bearing cover, inboard	49	Seal, bearing cover, outboard
73	Gasket	89	Seal

Figure 12.1.2i — Overhung, open impeller, separately coupled, single-stage, frame-mounted, metal, ASME B73.1 type pump (OH1)



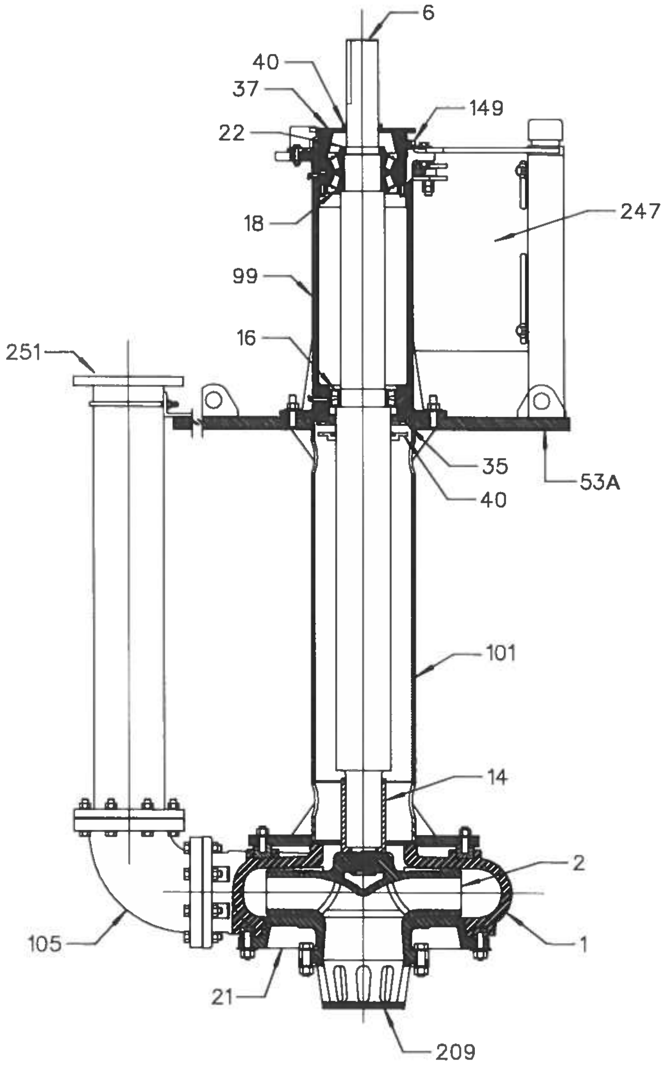
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
16	Bearing, inboard	18	Bearing, outboard
21A	Liner, suction cover	21C	Liner, gland half
21D	Liner, suction half	22	Locknut, bearing
23	Baseplate	37	Cover, bearing, outboard
40	Deflector	41	Cap, bearing, inboard
47	Seal, bearing cover, inboard	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 12.1.2j — Overhung impeller, separately coupled, single-stage, wet pit cantilever, elastomer-lined, single suction pump (VS5)



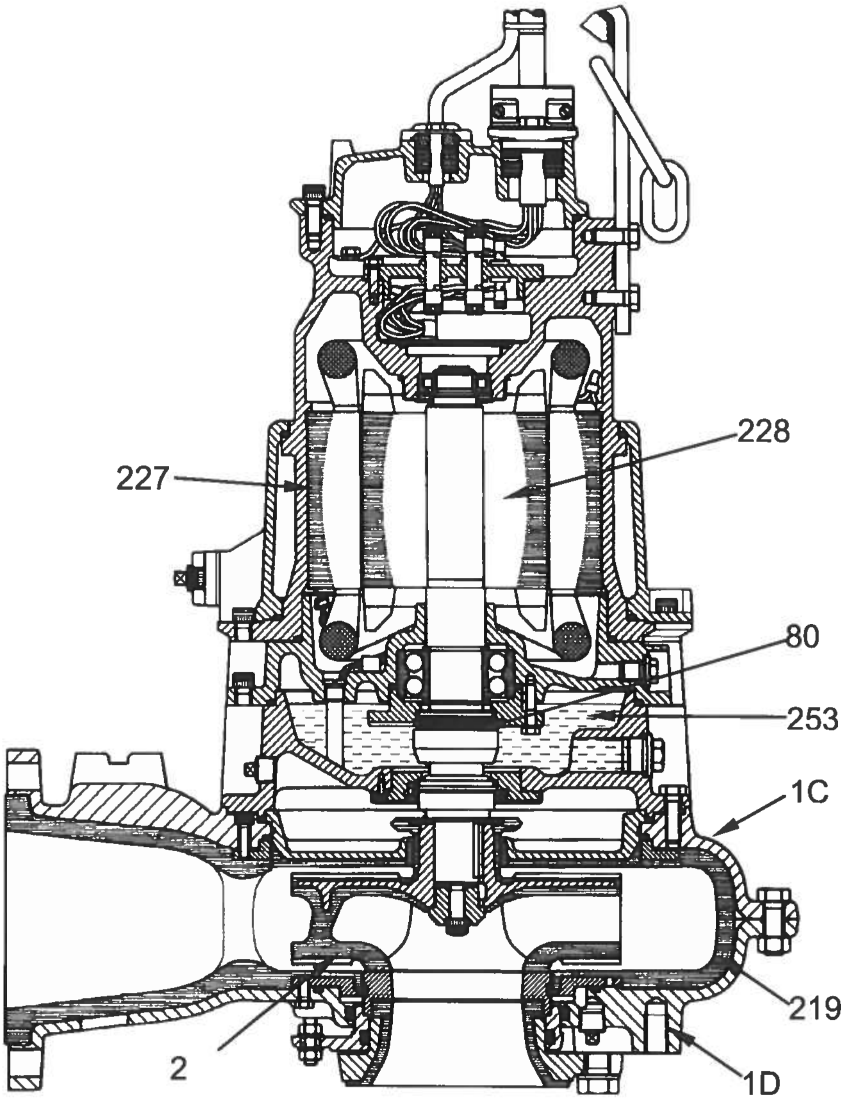
1C	Casing, upper	1D	Casing, lower
2	Impeller	6	Shaft
16	Bearing, inboard	18	Bearing, outboard
22	Locknut, bearing	37	Cover, bearing, outboard
40	Deflector	41	Cap, bearing, inboard
47	Seal, bearing cover, inboard	49	Seal, bearing cover, outboard
51	Retainer, grease	53	Base
99	Housing, bearing	101	Pipe, column
209	Strainer	251	Flange, discharge

Figure 12.1.2k — Overhung impeller, separately coupled, single-stage, wet pit cantilever, elastomer, vulcanized-lined, double suction pump (VS5)



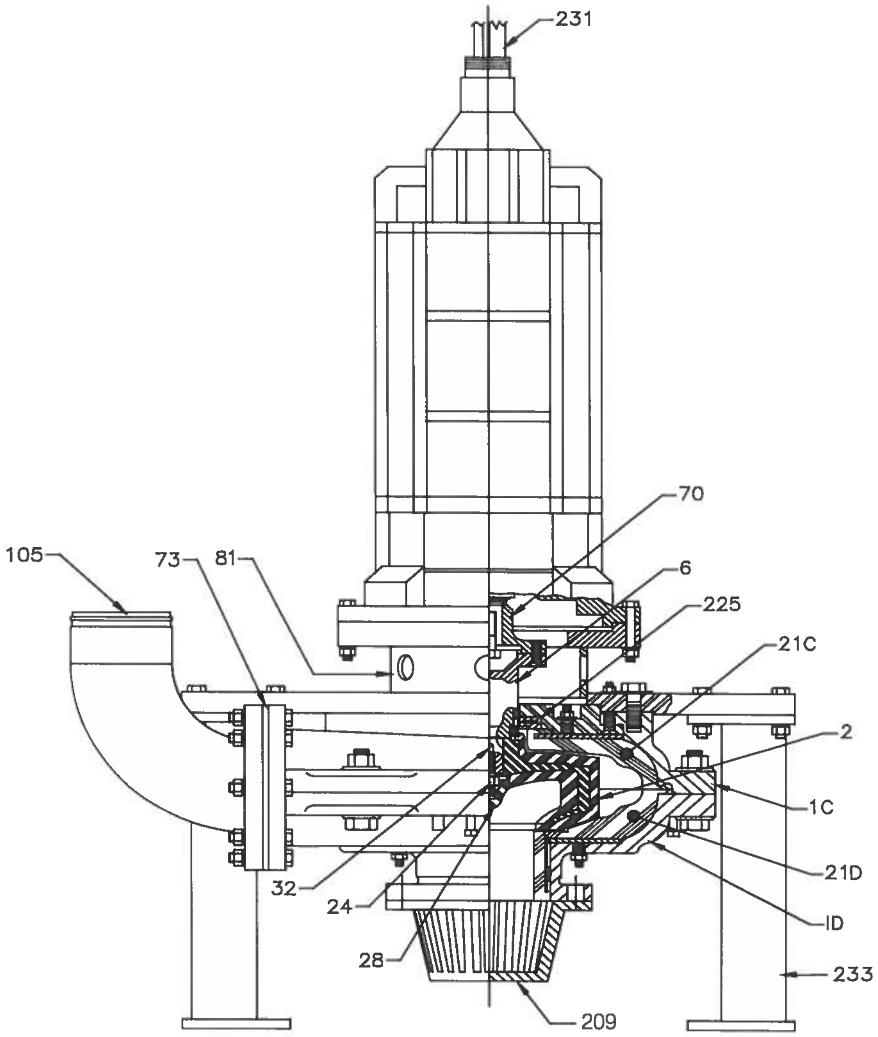
1	Casing	2	Impeller
6	Shaft	14	Sleeve, shaft
16	Bearing, inboard	18	Bearing, outboard
21	Liner, frame	22	Locknut, bearing
35	Cover, bearing, inboard	37	Cover, bearing, outboard
40	Deflector	53A	Plate, floor mounting
99	Housing, bearing	101	Pipe, column
105	Elbow, discharge	149	Screw, impeller adjusting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

Figure 12.1.2l — Overhung impeller, separately coupled, single-stage, wet pit cantilever, unlined, metal, single suction pump (VS5)



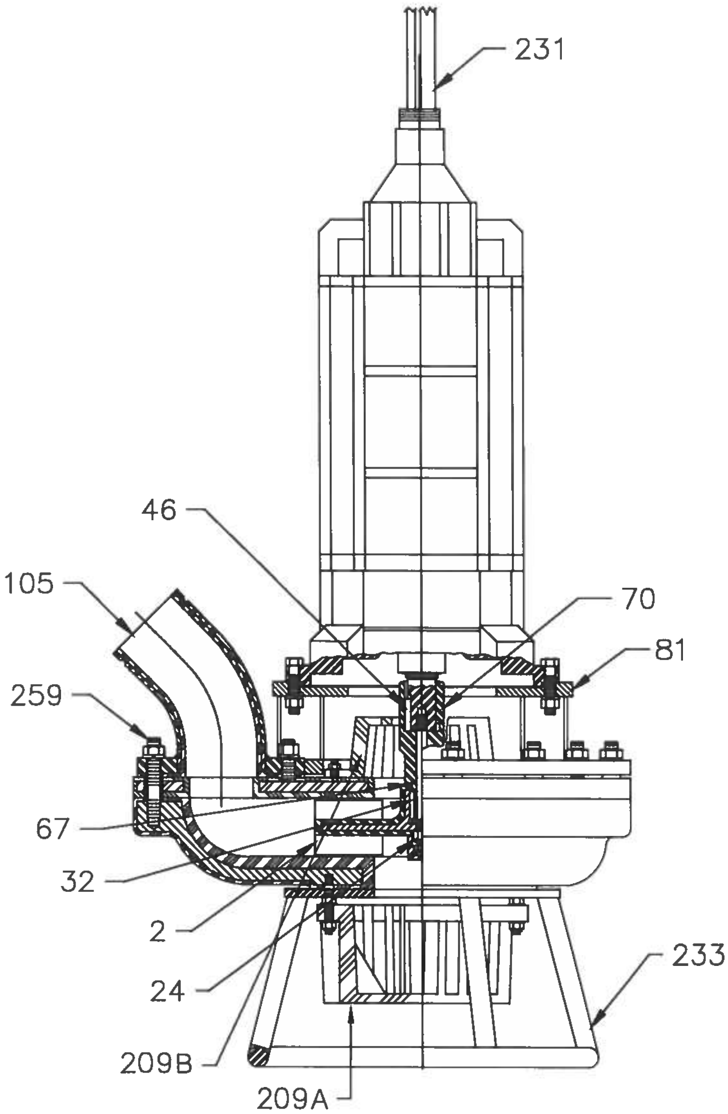
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	80	Seal, mechanical, rotating element
219	Liner, casing	227	Motor, stator
228	Motor, rotor	253	Chamber, barrier liquid, submersible

Figure 12.1.2m — Overhung impeller, close-coupled, single-stage, submersible, elastomer-coated, single suction pump (OH8B)



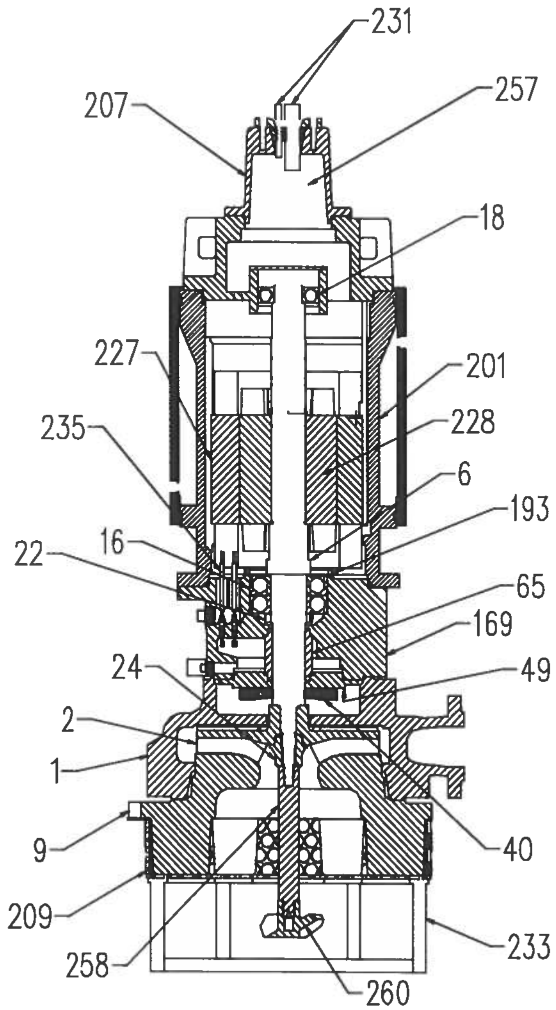
1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
21C	Liner, gland half	21D	Liner, suction half
24	Nut, impeller	28	Gasket, impeller screw
32	Key, impeller	70	Coupling, shaft
73	Gasket	81	Pedestal, driver
105	Elbow, discharge	209	Strainer
225	Plate, wear	231	Cable, electric power supply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

Figure 12.1.2n — Overhung impeller, close-coupled, single-stage, submersible, elastomer-lined, single suction pump (OH8B)



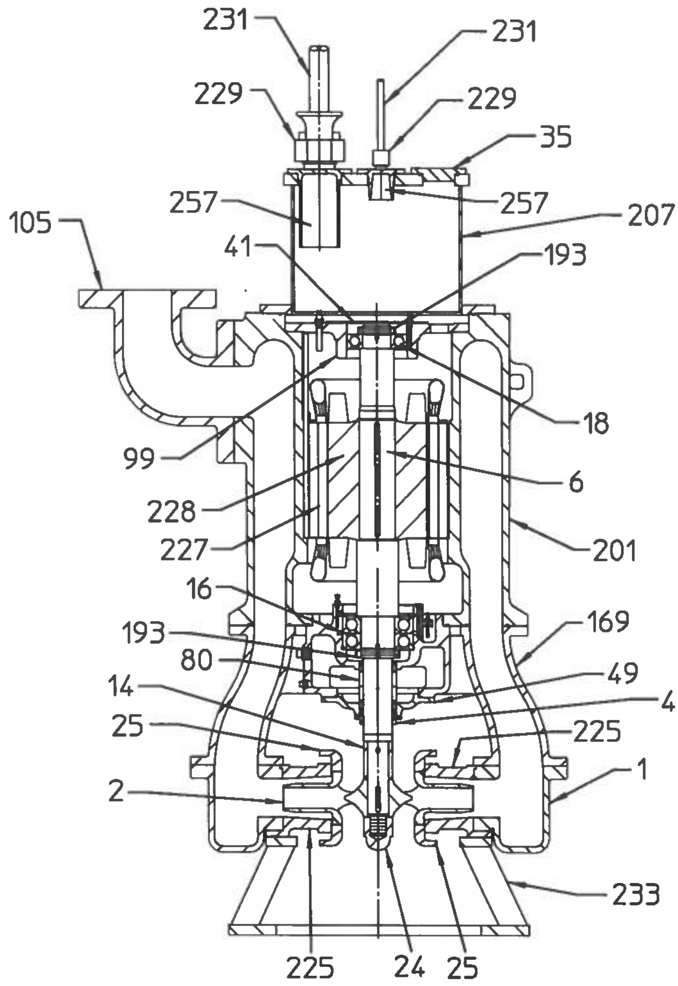
2	Impeller	24	Nut, impeller
32	Key, impeller	46	Key, coupling
67	Shim	70	Coupling, shaft
81	Pedestal, driver	105	Elbow, discharge
209A	Strainer, lower	209B	Strainer, upper
231	Cable, electric power supply or control	233	Stand, pump
259	Bolt, casing		

Figure 12.1.2o — Overhung impeller, close-coupled, single-stage, submersible, elastomer-lined, double suction pump (OH8B)



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 12.1.2p — Overhung impeller, close-coupled, single-stage, end suction, metal, submersible pump with agitator (OH8B)



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 12.1.2q — Overhung impeller, close-coupled, single-stage, submersible, metal, double suction pump (OH8B)

Table 12.1.12a — 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.
Baseplate	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	1C	Csg gld half	The gland half of a radially split casing.
Casing, suction half	1D	Csg suc half	The suction half of a radially split casing.
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.

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
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	4	Explr	A secondary impeller fitted with vanes used to reduce or balance pressure at the stuffing box of a slurry pump.
Expelling vanes	2B	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.

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
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 12.1.12a — 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 a slurry pump.
Liner, frame	21	Lnr fr	A part within the frame carrying one or more of the bearings.
Liner, gland half	21C	Lnr gld half	A part within the casing, gland half.
Liner, stuffing-box cover	21B	Lnr stfg box cov	A part within the stuffing-box cover.
Liner, suction cover	21A	Lnr suct cov	A part within the suction cover.
Liner, suction half	21D	Lnr suct half	A part within the casing, suction half.
Liner, vulcanized	21E	Lnr vul	An elastomer liner within the casing.
Locknut, bearing	22	Lknut brg	Fastening that positions an antifriction bearing on the 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	103	Disch pipe	Pipe used to provide a convenient discharge connection at or above the floor mounting plate of vertical sump pumps.

Table 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

Part name	Item No.	Abbreviation	Definition
Plate, floor mounting	53A	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 12.1.12a — Slurry pump nomenclature – alphabetical listing (*continued*)

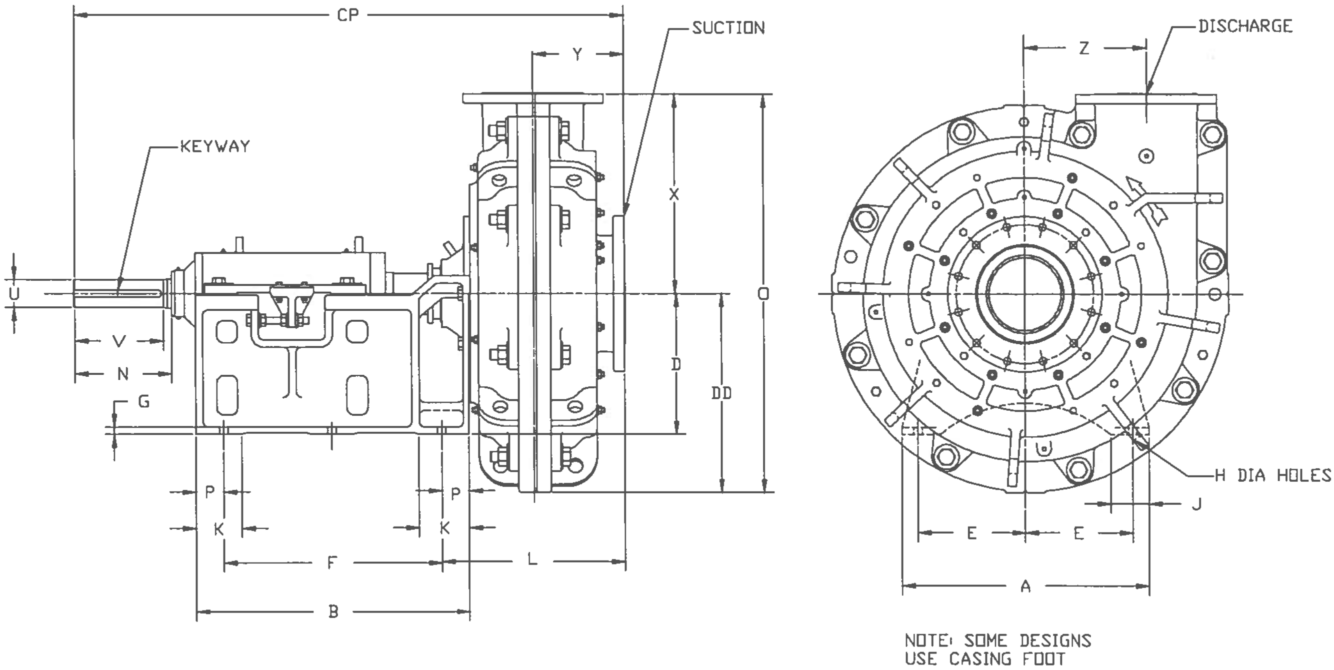
Part name	Item No.	Abbreviation	Definition
Seal, mechanical, stationary element	65	Seal mech sta elem	A subassembly consisting of one or more parts mounted in or on a stuffing box and having a smooth, flat sealing face.
Shaft	6	Sft	The cylindrical member on which the impeller is mounted and through which 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 relative position or fill in a gap.
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.
Stand, pump	233	Stnd pmp	A part that will support the pump weight.
Strainer	209	Str	Device used to prevent oversized objects from entering pump.
Strainer, lower	209A	Str, lwr	A device used to prevent large objects from entering the pump, located above the impeller centerline.
Strainer, upper	209B	Str, upr	A device used to prevent large objects from entering the pump, located below the impeller centerline.
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 disk rotating with the pump shaft to carry the lubricant from the reservoir to the bearing.

Table 12.1.12b — Slurry pump nomenclature - numerical listing

1	Casing	67	Shim
1C	Casing, gland half	68	Collar, shaft
1D	Casing, suction half	69	Lockwasher
2	Impeller	70	Coupling, shaft
2B	Expelling vanes	71	Adapter
4	Expeller	73	Gasket
6	Shaft	74	Journal, thrust bearing
9	Cover, suction	75	Stuffing box, auxiliary
11	Cover, stuffing box or seal chamber	76	Key, bearing journal
13	Packing	78	Spacer, bearing
14	Sleeve, shaft	80	Seal, mechanical, rotating element
16	Bearing, inboard	83	Stuffing box
17	Gland	88	Spacer, coupling
18	Bearing, outboard	89	Seal
19	Frame	99	Housing, bearing
20	Nut, shaft sleeve	101	Pipe, column
21	Liner, frame	103	Pipe, discharge
21A	Liner, suction cover	105	Elbow, discharge
21B	Liner, stuffing-box cover	107	Shield, oil retaining
21C	Liner, gland half	117	Bushing, pressure reducing
21D	Liner, suction half	119	O-ring
21E	Liner, vulcanized	120	Coupling, oil pump
22	Locknut, bearing	121	Pump, oil
23	Baseplate	123	Cover, bearing end
24	Nut, impeller	131	Guard, coupling
25	Ring, suction cover	133	Gland, stuffing box, auxiliary
26	Screw, impeller	143	Gauge, sight, oil
28	Gasket, impeller screw	149	Screw, impeller, adjusting
29	Ring, lantern	169	Seal, bearing housing
31	Housing, bearing, inboard	171	Bushing, throttle, auxiliary
32	Key, impeller	193	Retainer, bearing
33	Housing, bearing, outboard	201	Housing, stator
34	Sleeve, impeller hub	207	Cover, motor end
35	Cover, bearing, inboard	209	Strainer
36	Collar, release	209A	Strainer, lower
37	Cover, bearing, outboard	209B	Strainer, upper
38	Gasket, shaft sleeve	219	Liner, casing
39	Bushing, bearing	225	Plate, wear
40	Deflector	227	Motor, stator
41	Cap, bearing, inboard	228	Motor, rotor
42	Coupling half, driver	229	Clamp, cable
43	Cap, bearing outboard	231	Cable, electric control
44	Coupling half, pump	231	Cable, electric power supply
45	Cover, oil bearing cap	233	Stand, pump
46	Key, coupling	235	Probe, moisture detection
47	Seal, bearing cover, inboard	237	Housing, seal
49	Seal, bearing cover, outboard	241	Jacket, submersible motor
50	Locknut, coupling	247	Adapter, casing
51	Retainer, grease	249	Support, discharge pipe
53	Base	251	Flange, discharge
53A	Plate, floor mounting	253	Chamber, barrier liquid, submersible
57	Elbow, suction	255	Seal, cable jacket
60	Ring, oil	257	Seal, cable, epoxy
61	Plate, side	258	Extender, shaft
62	Thrower (oil or grease)	259	Bolt, casing
63	Bushing, stuffing box	260	Agitator, mechanical
65	Seal, mechanical, stationary element	261	Bolt, tie
66	Nut, shaft adjusting	263	Gasket, snap ring

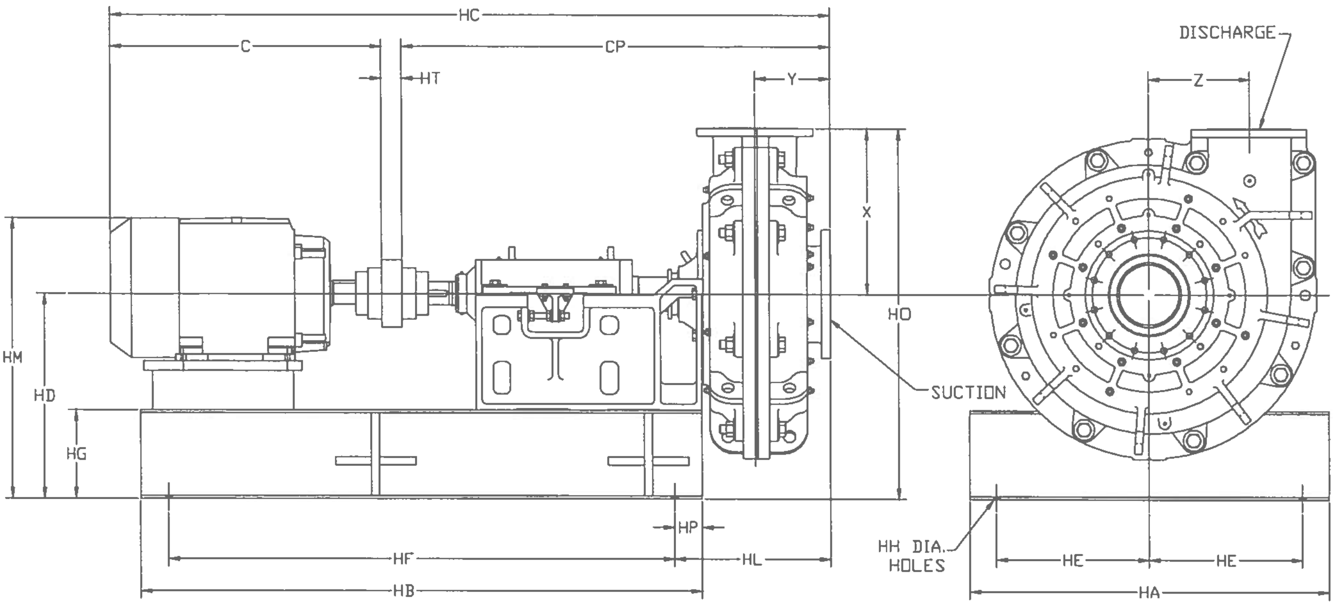
12.1.13 Letter dimensional designations

The letter designations used in Figures 12.1.13a through 12.1.13c provide a common means of identifying various pump dimensions and serve as a common language that will be mutually understandable to the purchaser, manufacturer, and anyone writing specifications for pumps and pumping equipment.



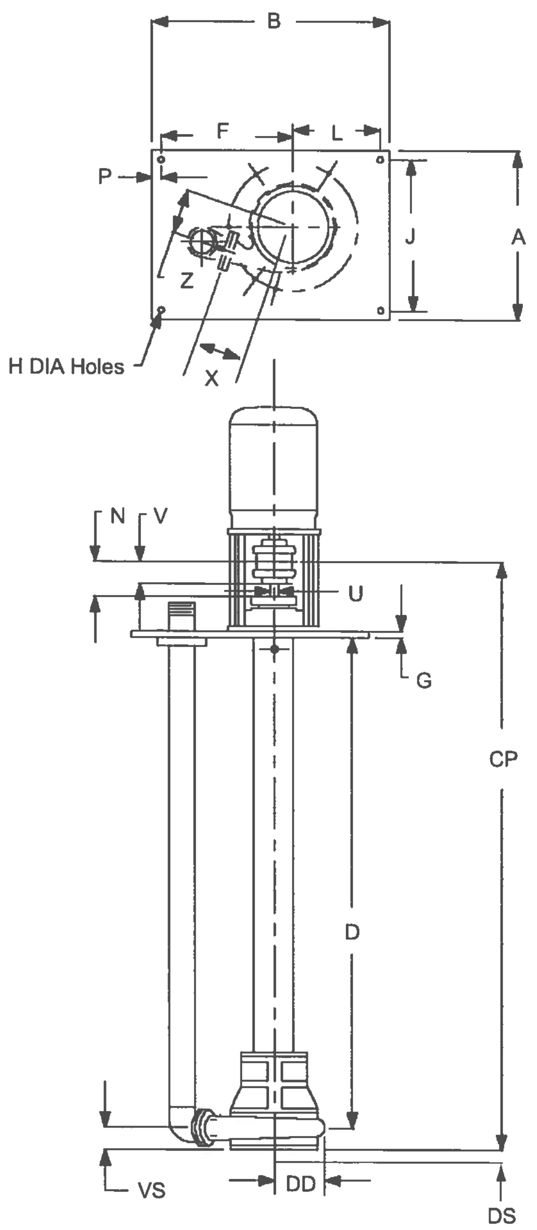
A	Width of base support	K	Length of support pads for hold-down bolts
B	Length of base support	L	Horizontal distance from suction nozzle face to centerline nearest hold-down bolt holes
CP	Length of pump	N	Distance from end of bearing housing to end of shaft
D	Vertical height from bottom of base support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

Figure 12.1.13a — Horizontal pump dimensions



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

Figure 12.1.13b — Direct drive pump and motor assembly dimensions



A Width of base support

B Length of base support

CP Length of pump

D Vertical height from 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 baseplate

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 from end of bearing housing to end of shaft

P Length from edge of support or baseplate to centerline of bolt holes

U Diameter of straight shaft-coupling end

X Distance from discharge face to centerline of pump

VS Vertical distance from centerline discharge nozzle to suction nozzle face

Z Centerline discharge nozzle to centerline of pump

Figure 12.1.13c — Vertical pump dimension

12.2 Definitions

This section defines terms used in slurry pump applications. Principal symbols, terms, and units are described in Table 12.2a and subscripts in Table 12.2b.

Table 12.2a — Principal symbols

Symbol	Term	To convert from US customary units (USCS)	Abbr	Into metric units	Abbr	Multiply by conversion factor
A	Area	square inch	in^2	square millimeter	mm^2	645.2
C_v	Concentration by volume	percentage	%	percentage	%	1
C_w	Concentration by weight	percentage	%	percentage	%	1
Δ (delta)	Difference	dimensionless[a]	—	dimensionless	—	—
D	Diameter	inch	in	millimeter	mm	25.4
η (eta)	Efficiency	percent	%	percent	%	1
γ (gamma)	Specific weight	Pound force per cubic foot	lbf/ft^3	Newton per cubic meter	N/m^3	157.1
g	Gravitational acceleration	foot/second squared	ft/s^2	meter/second squared	m/s^2	0.3048
h	Head (general term)	foot	ft	meter	m	0.3048
H	Pump total head	foot	ft	meter	m	0.3048
l	Static lift	foot	ft	meter	m	0.3048
ID	Inside diameter of pipe	inch	in	millimeter	mm	25.4
L	Length	inch	in	millimeter	mm	25.4
μ (mu)	Coefficient of friction	dimensionless	—	dimensionless	—	—
n	Speed	revolution/minute	rpm	revolution/minute	rpm	1
NPSHA	Net positive suction head available	foot	ft	meter	m	0.3048
NPSHR	Net positive suction head required	foot	ft	meter	m	0.3048
n_s (N_s)	Specific speed $n_s (N_s) = nQ^{0.5}/H^{0.75}$	Index number	—	Index number	—	0.0194[b]
ν (nu)	Kinematic viscosity	foot squared/second	ft^2/s	meter squared/second	m^2/s	0.092903
ϕ (phi)	Velocity in vibration	inch/second	in/s	millimeter/second	mm/s	25.4
π	pi = 3.1416	dimensionless	—	dimensionless	—	1
p	Pressure	pound/square inch	psi	kilopascal	kPa	6.895
P	Power	horsepower	hp	kilowatt	kW	0.7457
Q	Rate of flow (capacity)	US gallon/minute	US gpm	cubic meter/second	m^3/s	0.0000631
Q	Rate of flow (capacity)	US gallon/minute	US gpm	cubic meter/hour	m^3/h	0.2271

Table 12.2a — Principal symbols (*continued*)

Symbol	Term	To convert from US customary units (USCS)	Abbr	Into metric units	Abbr	Multiply by conversion factor
RT	Radial thrust	pound (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	Specific gravity (general term)					
S_m	Specific gravity (relative density) of slurry (mixture)	dimensionless	—	dimensionless	—	1
S_s	Specific gravity (relative density) of solid particles					
S_w	Specific gravity (relative density) of water, or carrier liquid					
T	Temperature	degree Fahrenheit	°F	degree Celsius	°C	(°F–32) × (5/9)
τ (tau)	Torque	pound-foot	lb•ft	newton-meter	N•m	1.357
U	Residual unbalance	ounce-inch	oz-in	gram-centimeter	g-cm	72
v	Velocity	foot/second	ft/s	meter/second	m/s	0.3048
w	Width	inch	in	millimeter	mm	25.4
X	Exponent	none	none	none	none	1
Z	Elevation gauge distance above or below datum	foot	ft	meter	m	0.3048

[a] Δ is a dimensionless symbol used to indicate a difference. This term takes on the units of the measured or calculated quantity associated with the difference.

[b] Where US cutomary units are ft, US gpm, and rpm, then the corresponding metric units are m, m^3/s, and rpm.

Table 12.2b — Subscripts

Subscript	Term
a	Absolute
atm	Atmospheric
b	Barometric
d	Discharge

Subscript	Term
min	Minimum
mot	Motor
OA	Overall unit
ot	Operating temperature

Subscript	Term
sol	Solid
stp	Condition at which the particles stop moving along the bottom of the pipe
t	Theoretical
v	Velocity (and volume in C_v)

Table 12.2b — Subscripts (*continued*)

Subscript	Term	Subscript	Term	Subscript	Term
g	Gauge	s	Suction (and solid particles in Ss)	vp	Vapor pressure
m	Slurry (mixture)	sg	Specific gravity	w	Water (and weight in C_w)
max	Maximum	smax	Slurry concentration corresponding to highest deposit velocity		

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 terms *flow rate* and *capacity* are 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 expressed in meters (feet) of the 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. Negative pressure or vacuum readings can also be expressed in millimeters of mercury (mm Hg) and inches of mercury (in. Hg).

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:

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

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 (ft/s); and g = gravitational acceleration, m/s^2 (ft/s^2).

(Metric)

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

(US customary units)

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

12.2.3.3 Elevation head (*Z*)

The potential energy of the liquid due to its elevation relative the pump's datum expressed in meters (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 is the plane through the upper eye, as shown in Figure 12.2.3.4.

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.

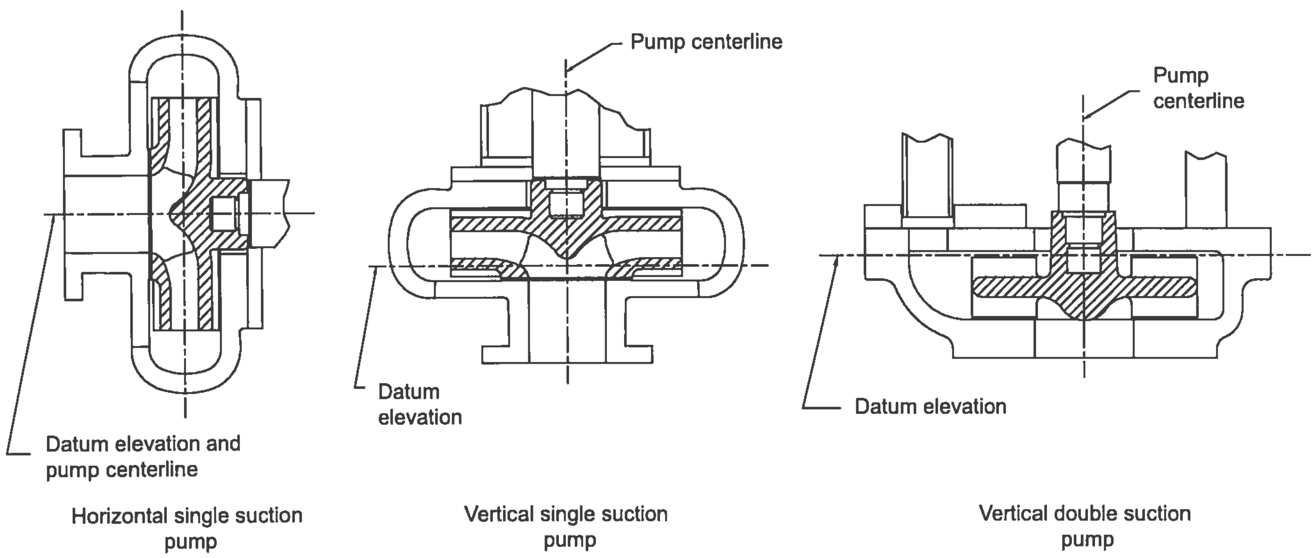


Figure 12.2.3.4 — Datum elevations for various slurry pump designs

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.

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

12.2.3.8 Pump total head (H)

Pump total head (H) 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.

$$H = h_d - h_s$$

12.2.3.9 Atmospheric head (h_{atm})

Local atmospheric pressure expressed in meters or feet of liquid being pumped 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 Shutoff

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 slurry pump at the specified speeds, as limited by cavitation, vibration, noise, heating, shaft deflection, and fatigue, and taking into account wear considerations (see Section 12.3.4). 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 of mixture:

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

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)

NPSH required (NPSHR) by a pump is the value of NPSH available (NPSHA) needed to operate the pump at a specific rate of flow. This value is normally recommended by a pump vendor.

NPSH3 is the value of NPSHR that results in a 3% loss of head (first-stage head in a multistage pump) determined by the vendor by testing with water.

12.2.5.4 Maximum suction pressure

This is the highest suction 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 (shaft) 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 (useful) power (P_u)

The power imparted to the liquid by the pump.

(Metric)

$$P_u = \frac{Q \times H \times s}{367.1} \text{ kW}$$

(US customary units)

$$P_u = \frac{Q \times H \times s}{3960} \text{ bhp}$$

12.2.6.4 Pump efficiency (η_p)

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

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

12.2.6.5 Overall efficiency (η_{OA})

This is the ratio of the energy imparted to the liquid (P_u) 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.

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

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 14.6 *Rotodynamic Pumps - Hydraulic Performance Acceptance Tests.*

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 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.

12.2.8.9 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.

12.2.8.10 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 (solids) 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 (μm) but may also be in other units, such as the Tyler Mesh as shown in Figure 12.2.9.4. This d50 designation is used by some engineers to calculate system requirements and pump performance. Figure 12.2.9.4 shows how it may be used to classify slurries and provides different d50 size equivalents.

12.2.9.5 Solids d85 size

The particle size where 85% by weight passes through a designated size screen. The d85 size is normally expressed in micrometers (μm), but may also be in other units, such as the as the Tyler Mesh, as shown in Figure 12.2.9.4.

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 that 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)

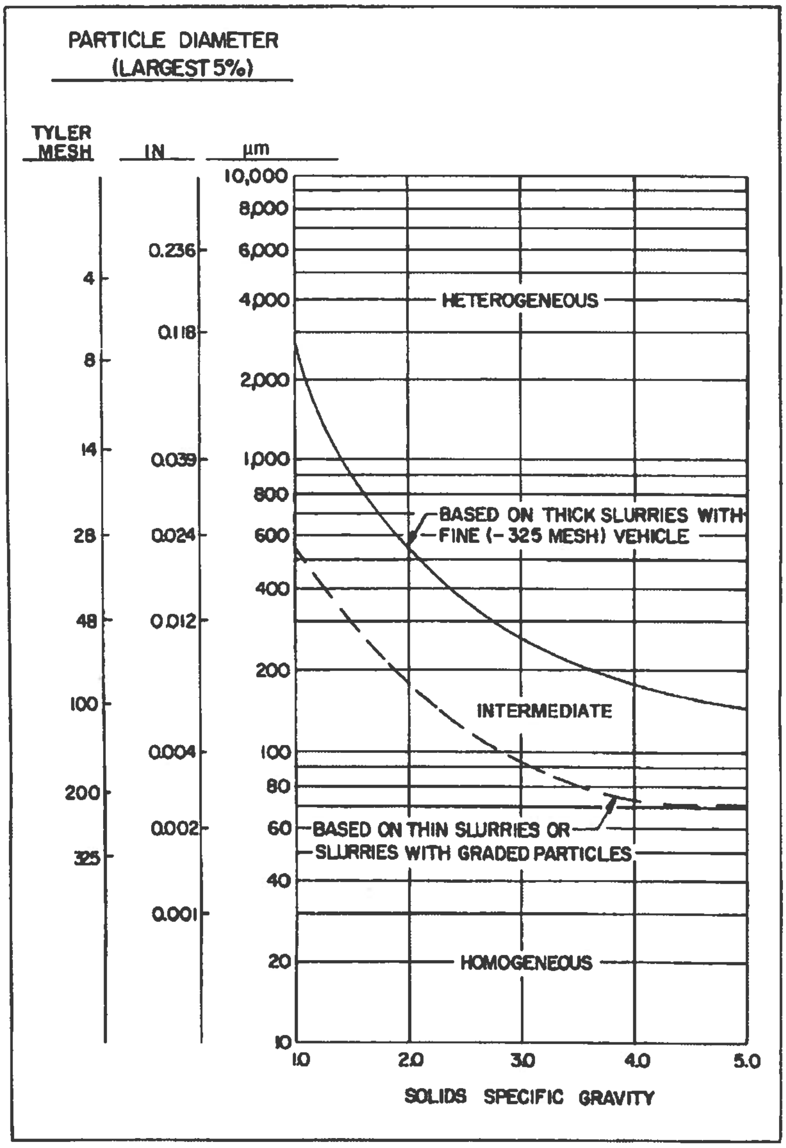
A type of slurry flow in which the solids are thoroughly mixed in the flowing stream and a negligible amount of the solids are sliding along the conduit wall.

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 Nonsettling 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.



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

Figure 12.2.9.4 — Schematic classification of slurries in industrial pipeline applications

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_{stp})

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_{smax})

The deposit velocity, at the slurry concentration, that results in the highest value of V_{stp}.

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 D.2.

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 Water head (H_w)

Head developed when pumping water, expressed in meters or feet of water column.

12.2.9.23 Slurry head (H_m)

Head developed when pumping slurry, expressed in meters or feet of slurry column.

12.2.9.24 Head ratio (H_r)

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

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

12.2.9.25 Efficiency ratio (η_r)

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

12.2.9.26 Head reduction factor (R_h)

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

12.2.9.27 Efficiency reduction factor (R_η)

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

12.2.9.28 Water efficiency (η_w)

Efficiency achieved when pumping clear water, expressed in percent.

12.2.9.29 Water power (P_w)

Power required to pump water, expressed in kilowatts or horsepower.

12.2.9.30 Slurry efficiency (η_m)

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

12.2.9.31 Slurry power (P_m)

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

12.2.9.32 Specific gravity correction factor (C_{sg})

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

$$C_{sg} = \left(\frac{S_s - 1}{1.65}\right)^{0.65}$$

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

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

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

12.2.9.34 Concentration correction factor (C_{cv})

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

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

12.2.9.35 Stationary bed

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

12.2.9.36 Laminar region

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

12.2.9.37 Turbulent region

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

12.2.9.38 Transition region

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

12.2.9.39 Slurry service classes

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

12.2.9.40 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.

12.2.9.41 Slurry abrasivity

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

12.2.9.42 Abrasive wear

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

12.2.9.43 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.

12.2.9.44 Corrosion

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

12.2.9.45 Erosion

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

12.2.9.46 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.

12.2.9.47 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*.

12.2.9.48 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*.

12.2.9.49 Specific energy (E_{sp})

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

12.2.9.50 Wear coefficient (W_c)

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

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 1.1-1.2 *Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions* 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 12.3.2.1 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 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 12.3.2.2.

12.3.2.3 Slurry types

Depending mostly on the size of the particles, slurries tend to be classified as settling or nonsettling. Nonsettling slurries act in a homogeneous manner and, in most cases, exhibit non-Newtonian characteristics at high concentration.

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 12.2.9.4 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_{stp}) 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_{stp} or the pipe will plug. Therefore, a system must be designed for the lowest acceptable value of settling velocity.

V_{stp} 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_{stp} varies with concentration. It is lowest at high concentrations and increases as concentration decreases. The point where V_{stp} reaches a maximum is defined as the maximum

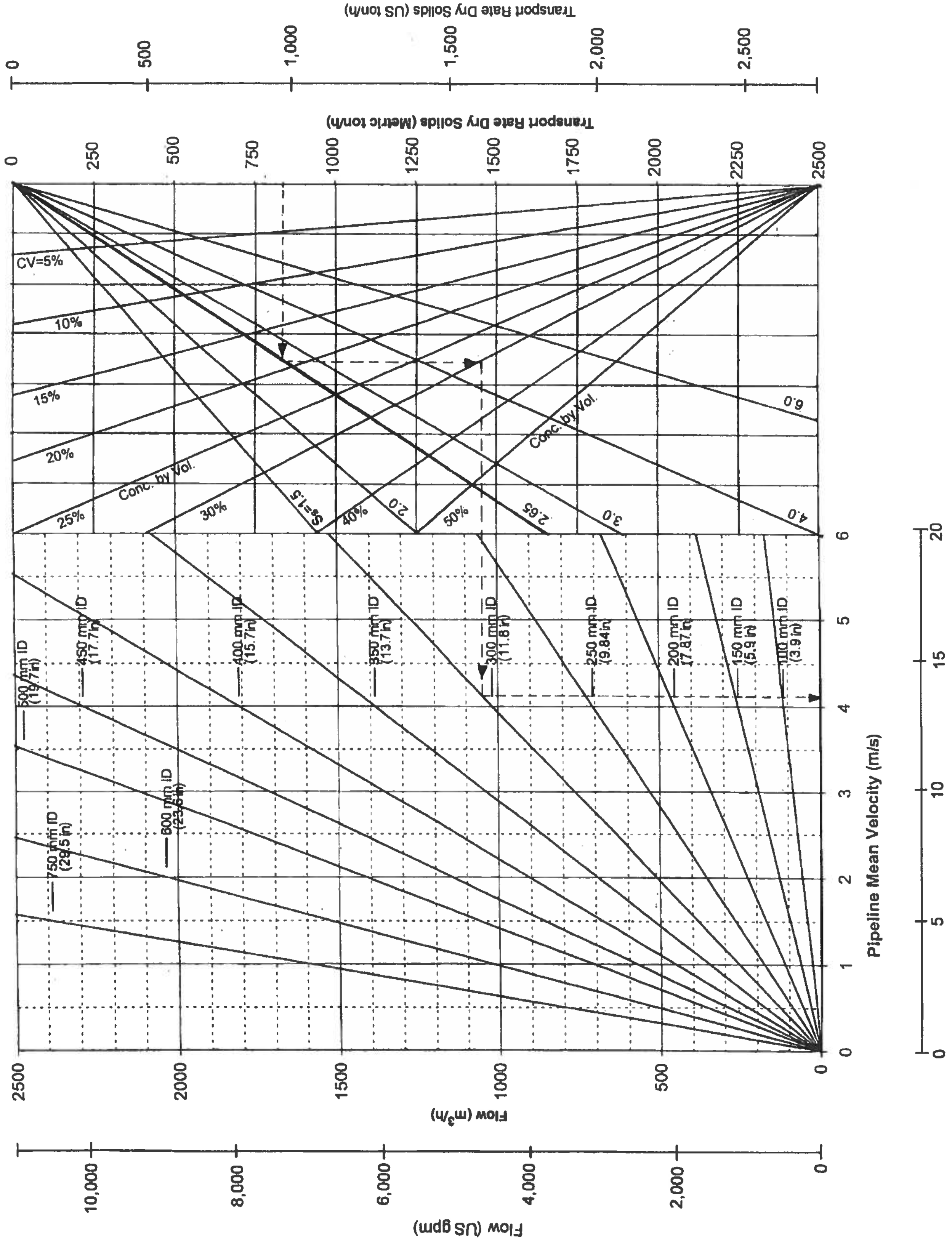


Figure 12.3.2.1 — Solids transport rate

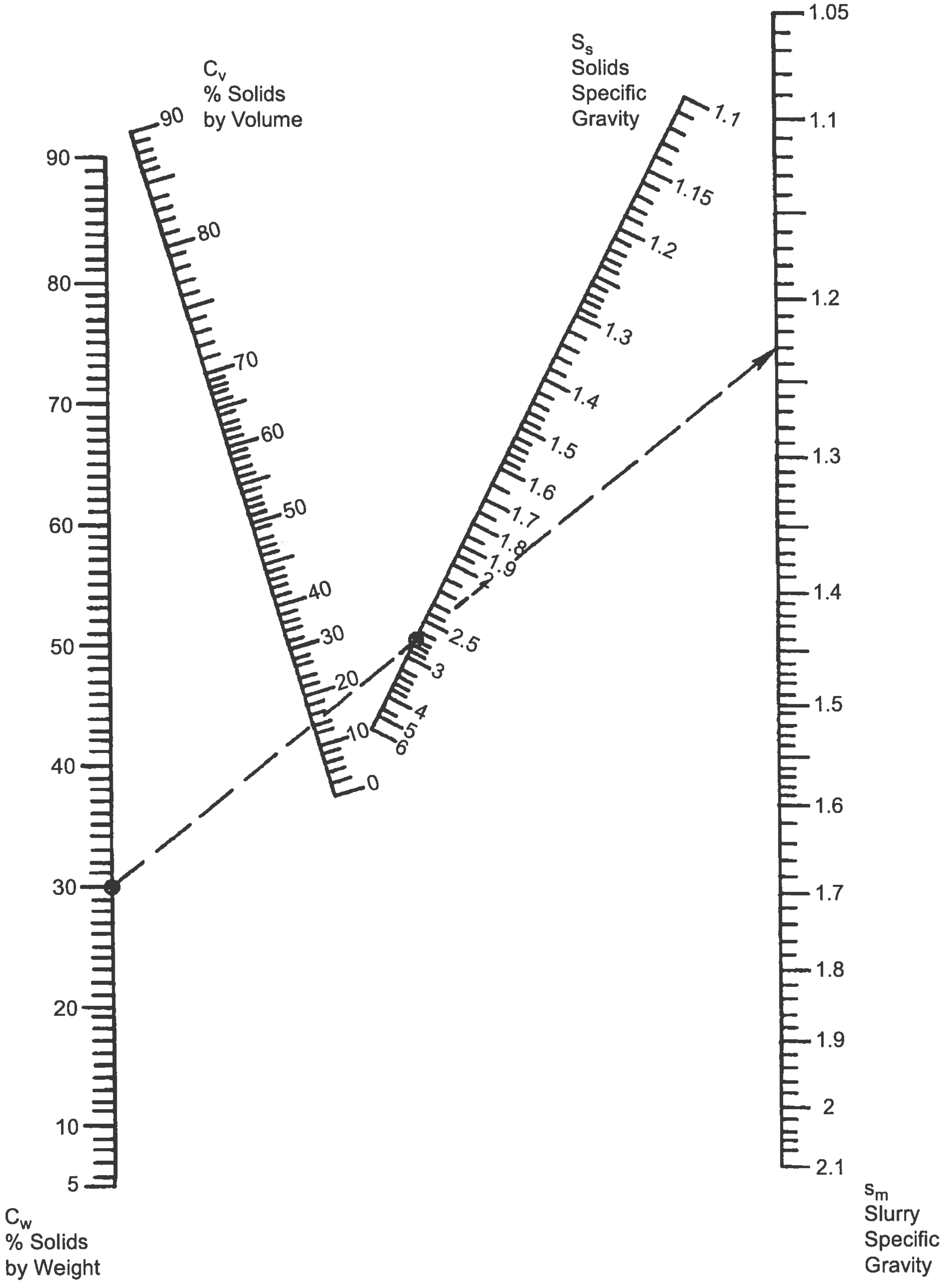


Figure 12.3.2.2 — Nomograph for the relationship of concentration to specific gravity in aqueous slurries

value of deposit velocity or maximum velocity at limit of stationary deposition (V_{smax}). The nomograph in Figure 12.3.2.4 can be used to determine V_{smax}. Because concentration cannot usually be controlled, the pumping system should preferably be designed so that the velocity in the piping always exceeds V_{smax} by at least a 10% margin.

To obtain a value for V_{smax}, draw a straight line on the left-hand half of Figure 12.3.2.4 from the appropriate pipe diameter value over the solids particle diameter line. Where that line intersects, the middle axis gives the value of V_{smax}. In the cases of solids specific gravities other than 2.65, a second line from the center axis V_{smax} 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 12.3.2.4 can also be used to evaluate the sensitivity of the system to changes in particle size. For example, V_{smax} 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_{smax} by more than 40%.

If the particle size is not closely controlled, the worst-case particle size (that giving the highest V_{smax}), should be used to determine V_{smax} for design. This will typically 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_{smax}, 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 12.3.2.5 along with the head and efficiency derating terms used.

Effects of solids on a slurry pump cavitation performance are dependent on the slurry type and the pump design and can be highly variable. The value of net positive suction head required in order not to exceed 3% head drop, NPSH3, will increase, in most circumstances.

For settling slurries of low to medium concentration, a modest increase in NPSH3 can be expected. For a particular application, this increase can be conservatively estimated by dividing the value of NPSH3 on water by the head derating factor discussed below.

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

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

When the solids–fluid mixture as shown in Figure 12.2.9.4 is considered homogeneous, the pump performance viscosity correction methods discussed in Section 12.3.2.6 can be applied. For heterogeneous settling slurries, a method using the solids size and pump impeller diameter is 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 are available.

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

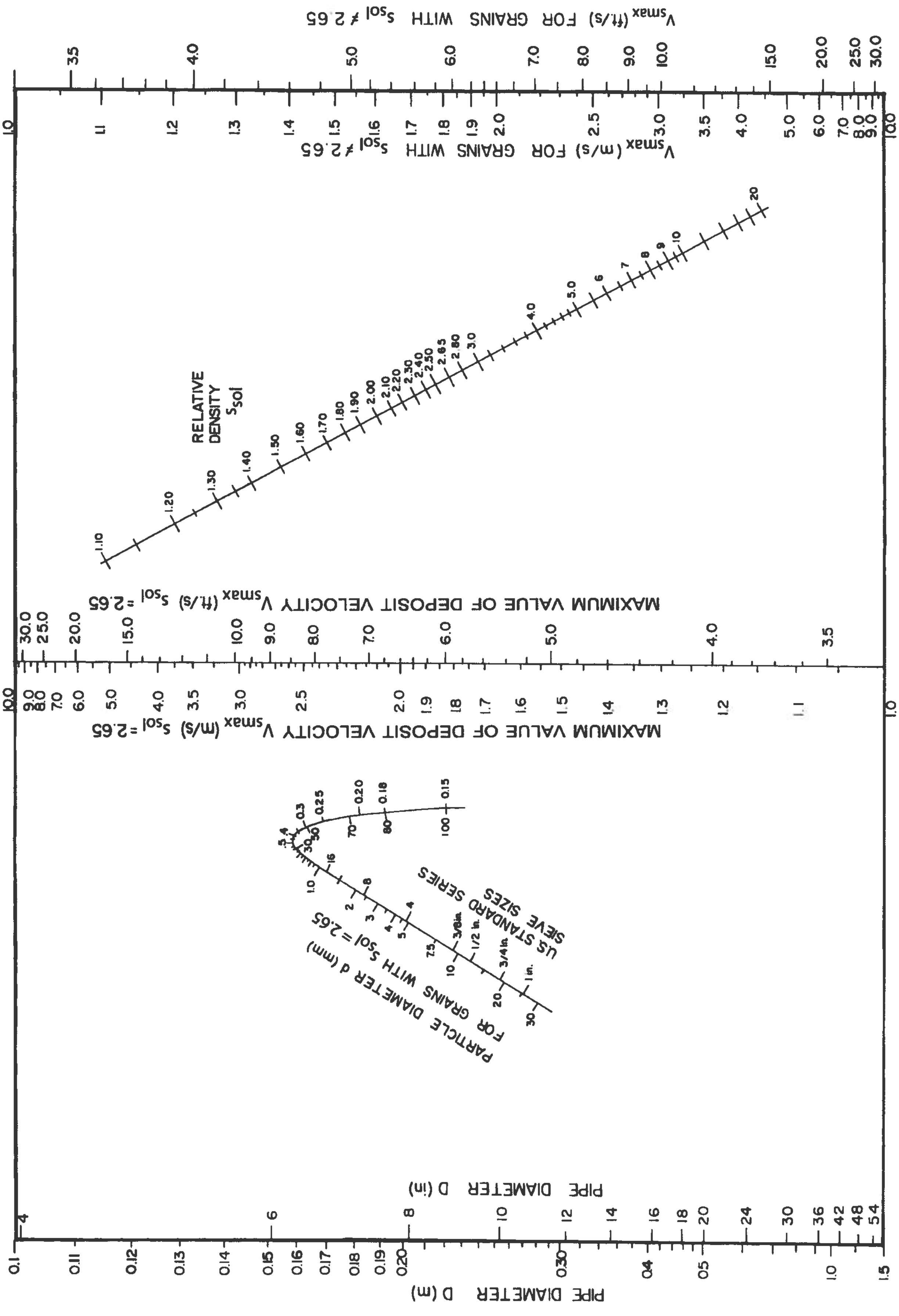


Figure 12.3.2.4 — Nomograph for maximum velocity at limit of stationary deposition of solids

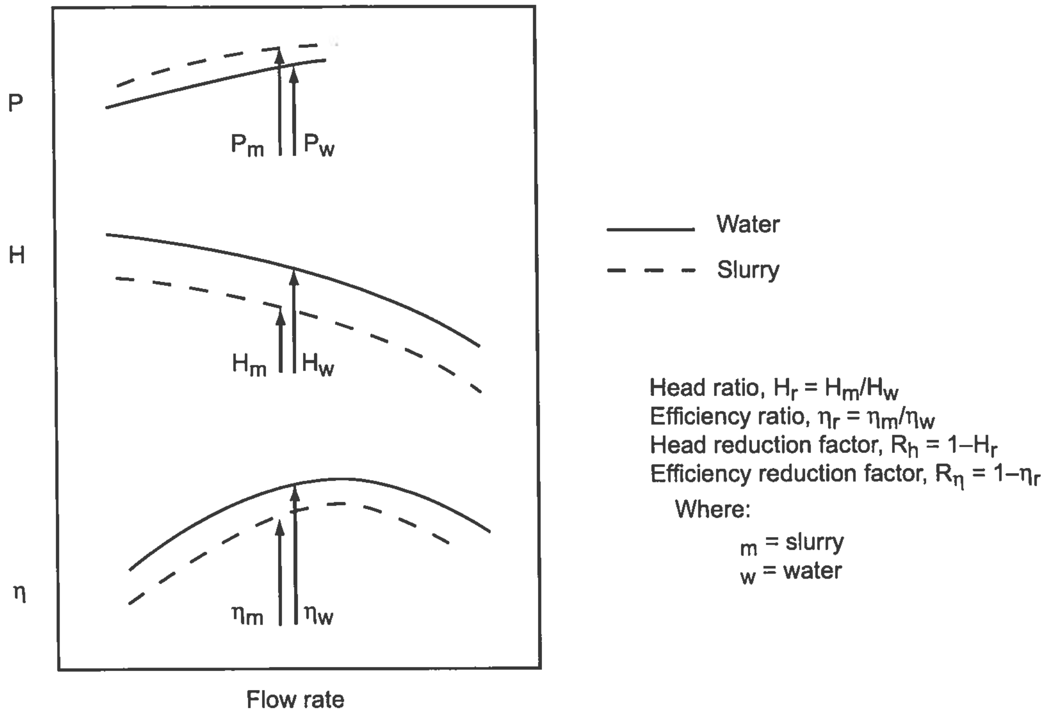


Figure 12.3.2.5 — Effect of settling slurry on pump characteristics (schematic)

12.3.2.6 Performance derating based on viscosity

Newtonian behavior is characterized by a fluid that is free flowing and has constant viscosity regardless of shear rate. The rheological plot of a Newtonian fluid, correspondingly, shows zero yield stress and a linear (straight-line) plot of shear stress versus shear rate. Low concentration, fine-particle slurries often exhibit this behavior.

When the solids–fluid mixture as shown on Figure 12.2.9.4 is considered homogeneous and exhibits Newtonian behavior, the ANSI/HI method for pump performance viscosity correction can be applied as outlined in ANSI/HI 9.6.7 *Effects of Liquid Viscosity on Rotodynamic Pump Performance.* With typical industrial slurries, viscosity is usually 50 to 100 times that of water (at 20 °C [68 °F]), resulting in reductions of 5 to 10% in head and efficiency, depending on the flow rate and head.

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.

At higher concentrations, homogeneous, nonsettling slurries often are not free flowing and correspondingly exhibit a yield stress. The presence of this yield stress results in non-Newtonian behavior with nonconstant apparent viscosity, which varies with shear rate and/or flow. This varying viscosity and the uncertainty associated with determining an appropriate shear rate make the use of ANSI/HI 9.6.7 cumbersome, less accurate, and likely to underestimate the performance derating at low-volume flow. Neglecting a possible severe performance derating at low relative rate of flow could result in a pump performance curve that intersects the system curve at multiple operating points and results in unstable flow. The pump manufacturer should be consulted for guidance regarding non-Newtonian effects on pump performance.

12.3.2.7 Performance derating based on solids size and content

Where the slurry is heterogeneous, Figure 12.3.2.7 can be used to determine the head and efficiency reductions from the original water performance for different sizes of slurry pumps of low to medium specific speeds for a slurry mixture concentration by volume of 15% and with negligible portions of less than 75-µm fines.

For solids of S_{sol} 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_{sg}, C_{fp}, and C_{cv} noted below, applied concurrently.

Specific gravity correction factor (C_{sg}), where $C_{sg} = [(S_s - 1)/1.65]^{0.65}$ or using Table 12.3.2.7.

Table 12.3.2.7 — 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

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

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 12.3.2.7, have an R_h of 8% and an H_r of 0.92. The head

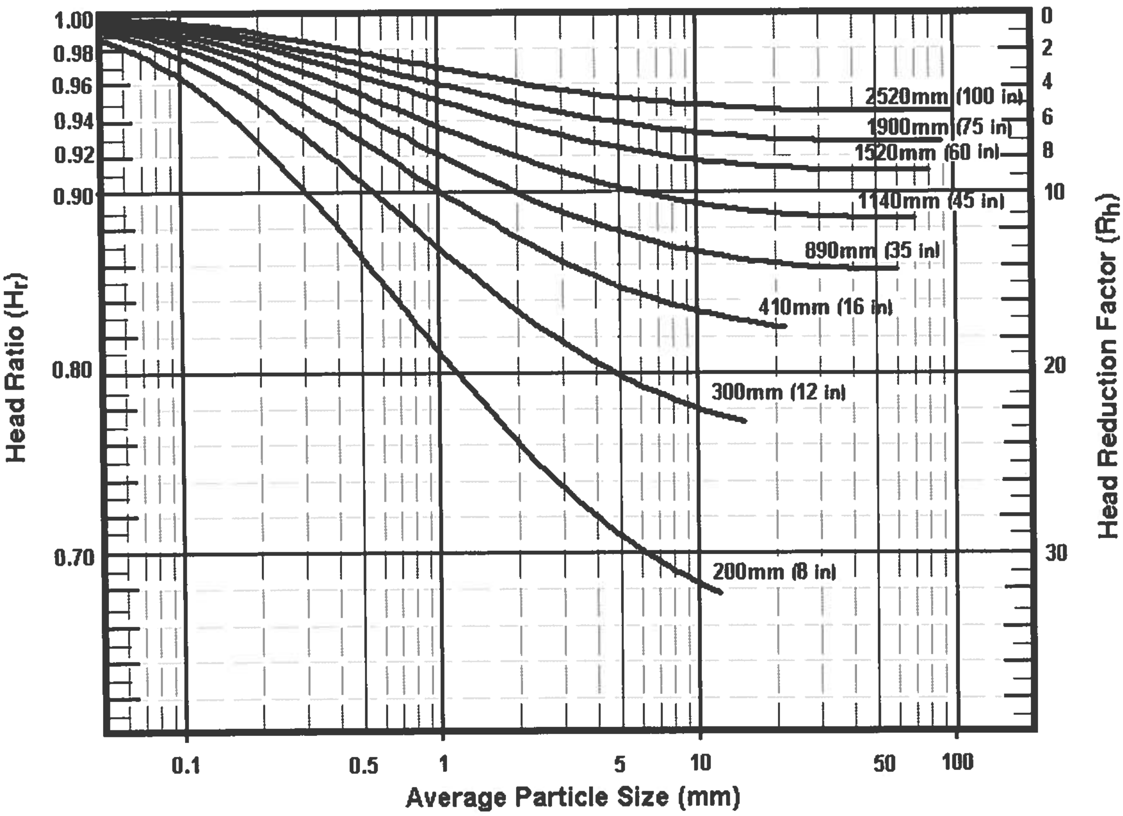


Figure 12.3.2.7 — Effect of average particle size and impeller diameter on H_r and R_h
(For solids concentration by volume, C_v = 15% with solids S_s = 2.65 and a negligible amount of fine particles. Impeller diameters are given in millimeters and inches.)

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 µm, then the R_h value above would become $R_h = 16 \times (1 - 0.20)^2 = 10.2$ and the head produced would be 61 × 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 × 1.13 = 11.5 and the head produced would be 61 × 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, which 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, used.

12.3.3 Froth pumping

Froth is an aerated liquid medium (slurry) that occurs naturally or is created on purpose. Natural occurrence may be due to the nature of the ore processed in the mineral industries, creating a general nuisance in many cases. 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 with the addition of polymers 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 centrifugal 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 that are 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 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 usual 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.

Oversizing the pump increases its ability to handle multiphase mixture containing air or gas due to the following reasons:

- The larger inlet diameter provides for more physical space to cope with an accumulation of air without restricting the fluid flow into the pump.
- The increased impeller eye diameter requires suction pipe of larger diameter. This leads to reduction in the suction line velocity, thus reducing line losses, and therefore increases the available NPSH of the system.

- The impeller of larger diameter will run at a lower speed of rotation, which will result in reduction of the NPSH required by the pump.

It is important to understand that it is the volume of air at the pump inlet that affects the pump performance. Air volumetric concentration directly depends on suction pressure. For this reason, the froth pumps should always operate with positive suction pressure, and the higher the pressure, the better the pump performance that can be expected.

Although applying a larger pump may be considered a cost implication, this approach has proven to work well in many processes, especially with vertical pumps, applications of which normally fall into the low system head range, with heads seldom over 20 m (65.6 ft). 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.

For practical reasons, such as space or head generation, some installations are better suited to horizontal pumps. When standard horizontal pumps are used, large froth factors are applied, resulting in pumps typically one or two sizes larger than would normally be used for nonaerated duties. Open and semiopen impellers are preferred to closed impellers because of the larger vane passageways and their theoretical ability to handle a larger air bubble before air binding.

Special-purpose horizontal froth pumps are available with increased inlets, flow inducers, and specially profiled vanes to increase air-handling capability.

Where higher heads are required, multiple pumps can be used in series. With increased suction pressure, subsequent stages are expected to experience reduced air volume and improved performance. Testing of conventional horizontal pumps handling air-entrained liquids, however, has still shown significant performance derating at elevated suction pressure, even with relatively low air volume.

Pumps for mineral froth service can be either elastomer-lined or constructed of hard metal to provide abrasion and corrosion resistance. Pumps for bitumen froth duties are typically hard metal, although liners of synthetic elastomers, such as neoprene (see Section 12.3.7.4), are a viable choice, depending on maximum particle size.

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, and the pump manufacturer should be consulted for sizing recommendations. Some typical vertical pump froth factors for common processes are given in Table 12.3.3. These are only approximate values; the most reliable factors will come from the users.

Table 12.3.3 — 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

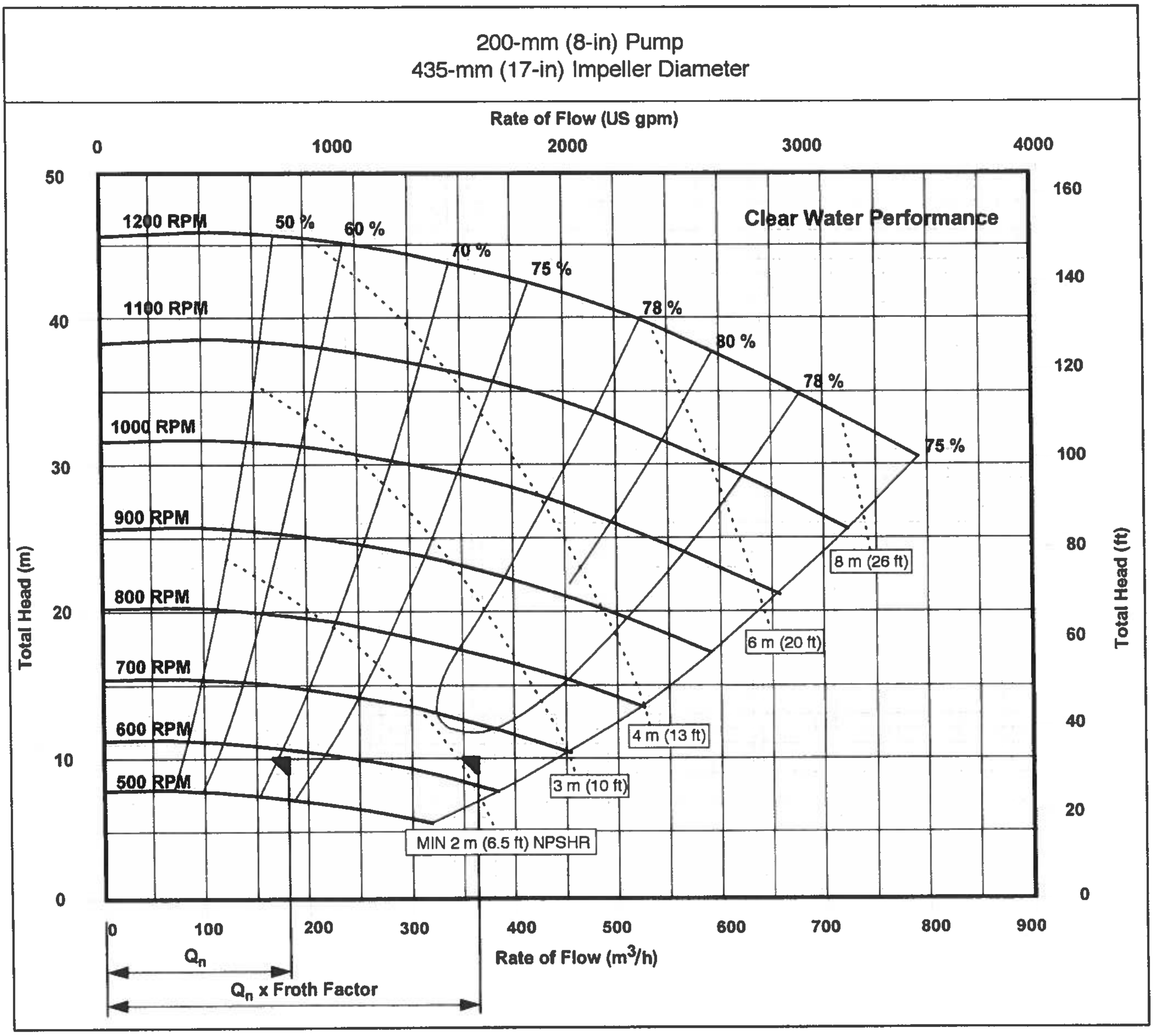


Figure 12.3.3 — Application of froth factor to pump selection

Figure 12.3.3 shows one traditional method in which a froth factor is used to increase the process design capacity and select a conventional pump for froth service. Example: molybdenum rougher concentrates with a design capacity of 182 m^3/h (800 US gpm) at a required head of 10 m (33 ft). A 150-mm (6-in) discharge size pump would normally be selected for this flow, but this example 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. In this method 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 when the system is operated without froth. These conditions will exist at system start-up, after power outages, or when system shut-downs 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.

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.

Other manufacturer/pump design-specific methods exist that use both air content and system curve review to determine pump selection, performance derating, speed, and expected power draw.

The sump design will affect 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 methods used to enhance the pump performance when handling difficult froths include:

- Misting, via nozzles, to provide a gentle rain-like spray to pierce the froth bubbles
- A partly submerged feed box within the sump to control the entrance of air and slurry and allow for venting in the remainder of the sump
- Inclined plates to separate or baffle the air-entrained slurry feed into the sump from the pump suction and provide additional surface area, a torturous path, additional travel time within the sump, and a dead zone under the baffle, all to enhance liberation of air before reaching the pump suction
- A vent on the suction line to allow air/froth to escape
- A tall sump to maximize suction static height and corresponding suction pressure and reduce air volume at the pump inlet through the relationship $p_1v_1 = p_2v_2$
- The use of eccentric reducers at the sump and pump suction, with the reducer at the sump having the belly side up to vent back into the sump, and the reducer at the pump suction having the belly side down, so as to not trap air in the inlet pipe

12.3.4 Wear in centrifugal slurry 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 to 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 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 12.3.4.1a 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 12.3.4.1a 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 12.3.4.1a 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.

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.

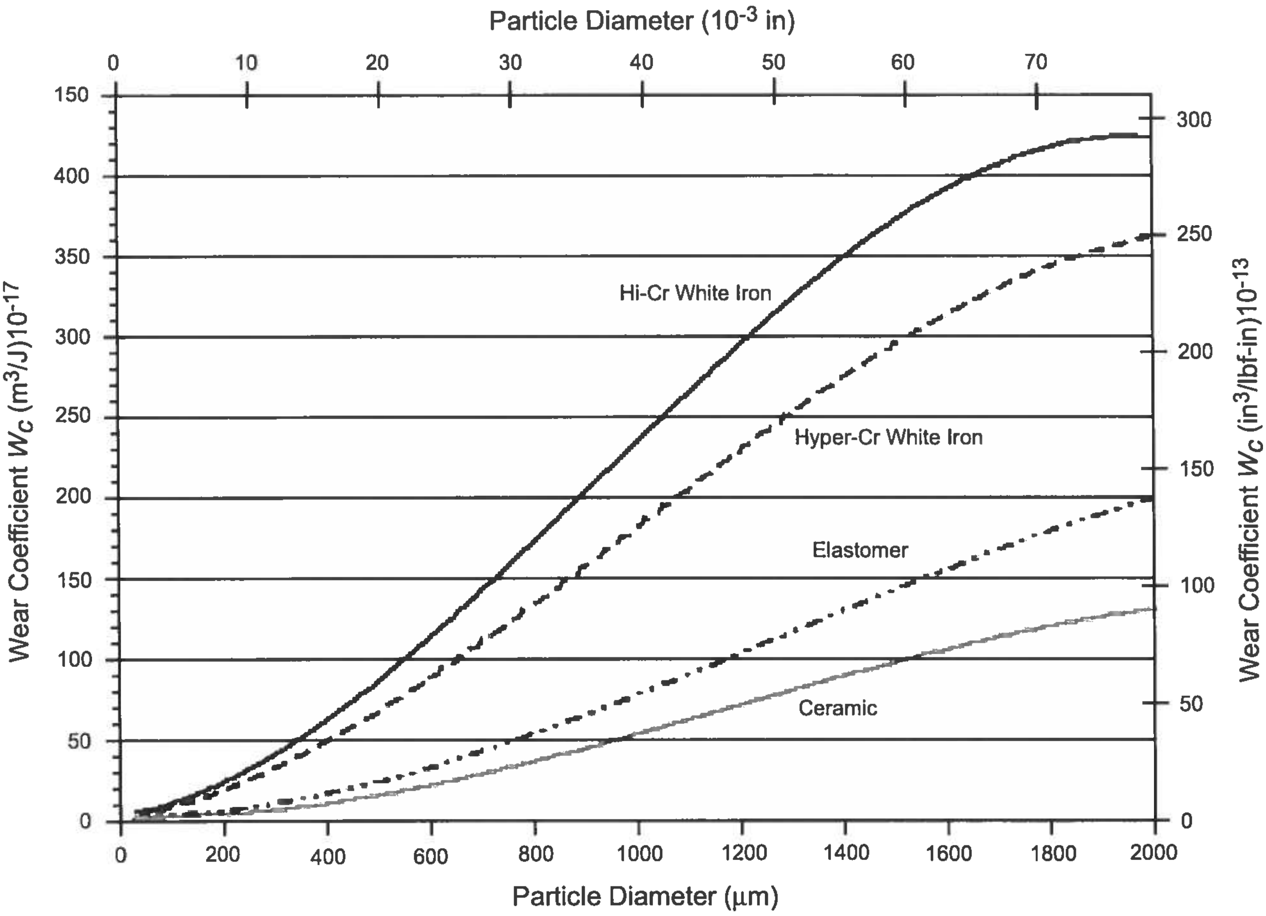


Figure 12.3.4.1a — Wear coefficient W_c for different resisting materials in a neutral pH media for different average-sized abrading particles

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 12.3.4.1b 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 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.

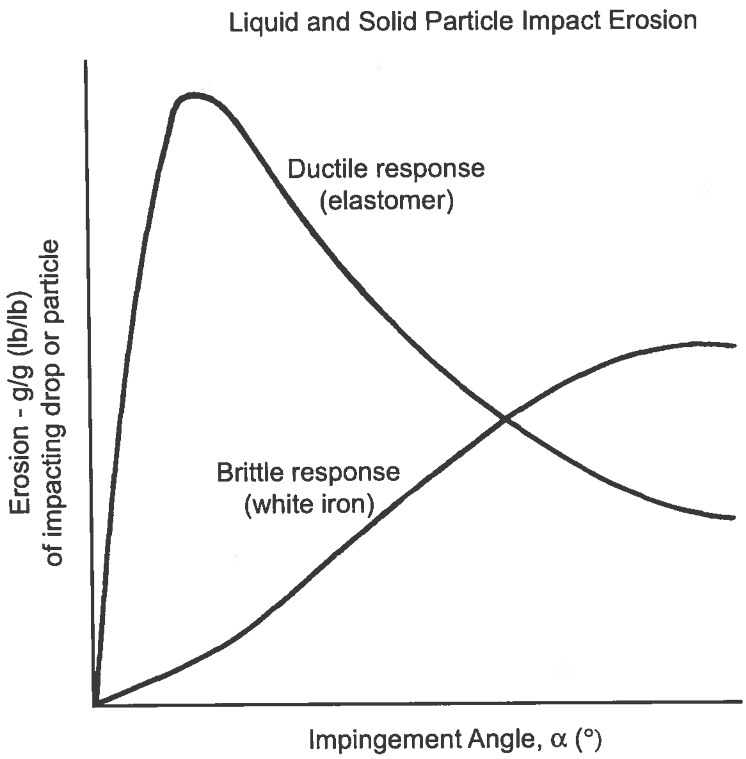


Figure 12.3.4.1b — Erosion response for different impingement angles and materials

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 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 12.3.4.2a, provides a practical tool for pump selection and, in conjunction with Table 12.3.5a, a means of recommending limiting pump operating heads.

The boundary lines between the service class areas in the chart approximate limits of constant wear modified for practical considerations and experience.

Capital and operating cost considerations are such that different (higher specific speed) designs may be employed for the lighter service classes.

Slurry service ranking shown in Figure 12.3.4.2a 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. For slurry solids other than silica sand, the ASTM 075-95 Miller number may be utilized to account for the different solids abrasivity in the use of Figure 12.3.4.2a. Here Figure 12.3.4.2b of Miller number versus modified abrasivity (A_{mod}) may be employed to obtain corrected values of S_m and d50 using the relations:

$$S_m \text{ corrected} = S_m \text{ actual} \times 0.4\, A_{mod} + 0.6$$

$$\text{d50 corrected} = \text{d50 actual} \times (A_{mod})^{0.4}$$

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.

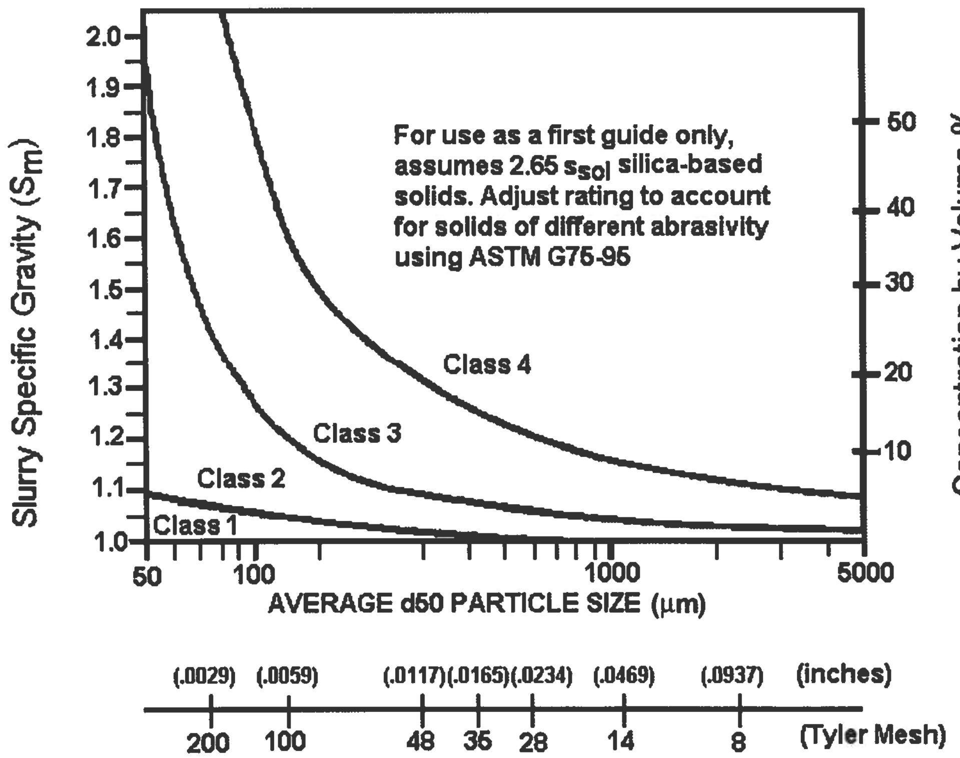


Figure 12.3.4.2a — Service class chart for slurry pump erosive wear

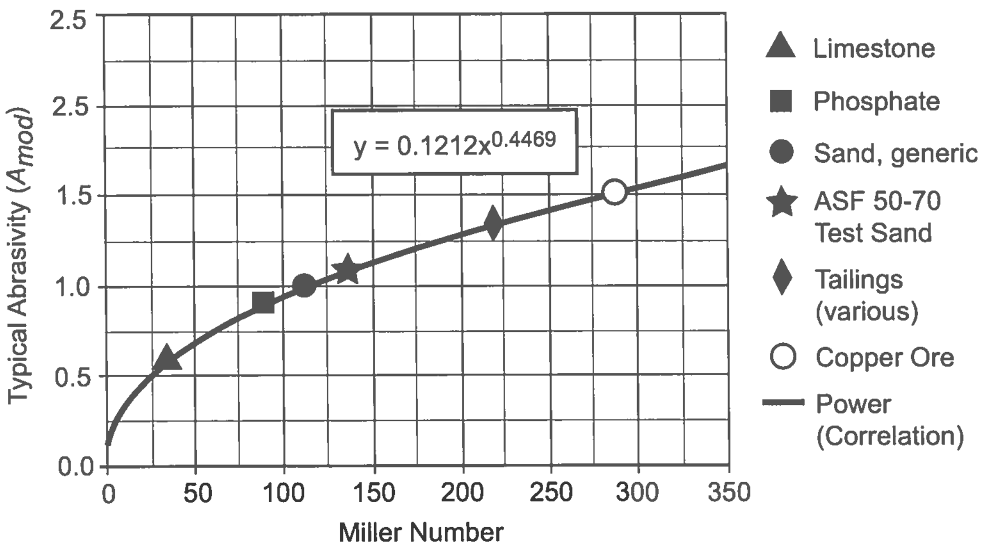


Figure 12.3.4.2b — Miller number versus material abrasivity

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 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 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 μm (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 12.3.5a provides recommended service limitations for different service classes that, when coupled with proper design and material selection, has resulted in acceptable wear life.

The heads referred to in Table 12.3.2.5a are actual heads after being corrected with the methods outlined in Section 12.3.2.5.

Table 12.3.5a — Recommended service limitations for acceptable wear

	Service class			
	1	2	3	4
Head per stage: m (ft)	105 345	73 240	55 180	40 130
Impeller peripheral speed: All-metal pump m/s (ft/min)	46 9000	38 7500	33 6500	28 5500
Rubber-lined pump	Head generated by impellers made of natural rubber is generally limited to 40 m (130 ft) per stage, which corresponds to peripheral speed of 28 m/s (5500 ft/s). Synthetic elastomers may allow higher limits.			

While the best service life (shut-down for some wear parts change out) of a pump in class 4 service can be limited to four months or less, the expectation for the lower number classes (at the same head) can be approximately quadrupled for each change to a lower class number. At the higher allowable heads in Table 12.3.5a, the wear lives, however, are at least doubled.

Capital and operating cost considerations are such that different (higher specific speed) designs may be employed for the lighter service classes.

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. 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.

If the service conditions are intermittent, or if experience warrants, then lower class and/or higher impeller peripheral speeds and heads can be used.

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 a concentric 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.

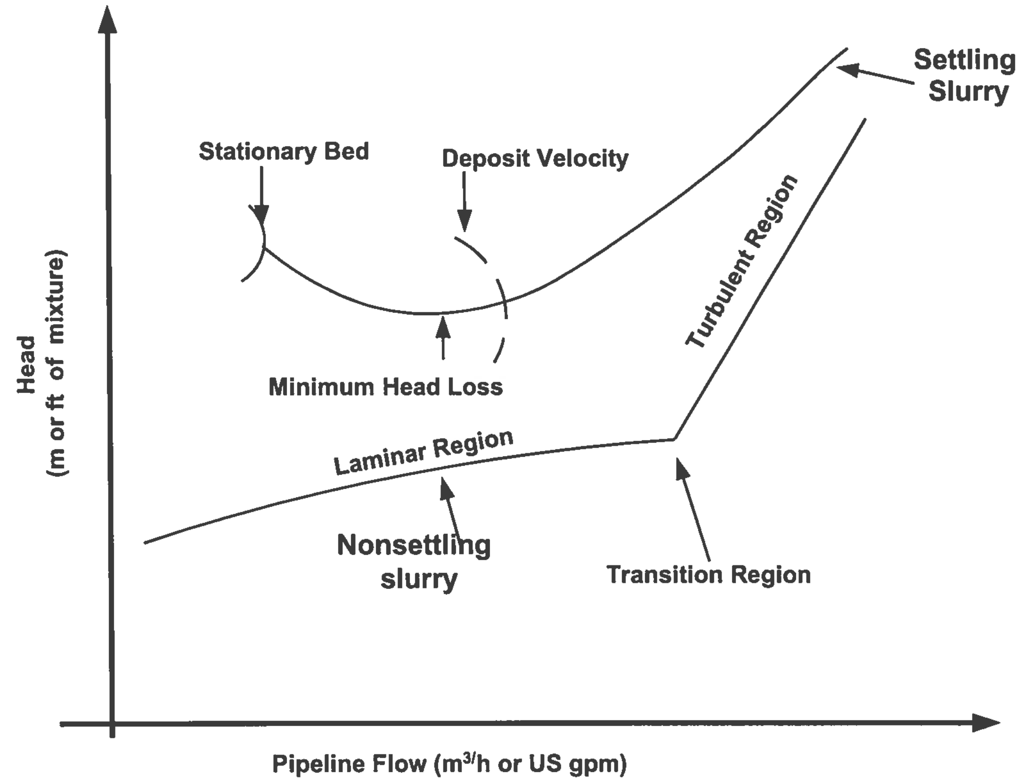


Figure 12.3.6 — Typical constant concentration slurry pipeline friction loss characteristics

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 12.3.5b shows the generally acceptable range of flow rates for the different designs.

Table 12.3.5b — Acceptable range of relative flow rates for slurry pump casing optimum wear

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%

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 12.3.6 gives a generalized depiction of slurry effects in piping systems.

System head and velocity requirements for nonsettling slurries can be treated very similarly to clear viscous liquids. Systems should be designed to operate near the transition point (refer to Figure 12.3.6) to obtain energy-efficient and stable operation.

System head and velocity requirements for settling slurries can be determined by similar means, but the deposit velocity (Figure 12.3.2.4) 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 12.3.6) or 110% of the V_{smax}.

Wherever possible, energy usage and operating cost must be minimized within the constraints imposed by the slurries themselves and the associated wear. Here it will be noted that the total cost measurement and any efficiency comparison will be to the transport solids, not water, and the cost should include energy, capital, rotating assembly maintenance, wear parts replacement cost, and any associated downtime cost related to stoppages to replace wear parts.

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 length 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 because of 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 12.3.4.2, introduced earlier, may be used to rank purely erosive wear. Table 12.3.7.1 is a selection guide for various materials commonly used in these services along with their appropriate erosive wear service class. Appendix D.1 lists the various international standards for materials typically used in slurry pump construction.

Table 12.3.7.1 — 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	Corrosive
Duplex stainless steel	Moderate	2	Corrosive

Table 12.3.7.1 — General suitability of wetted materials (*continued*)

Wetted Material	Abrasive characteristics of pumpage	Applicable wear service class	Corrosive characteristics of pumpage
Super-duplex stainless steel	Moderate	2	Highly corrosive
Elastomers	Severe, fine particles	3[a]	Mildly corrosive

[a] Primarily for impellers. Elastomeric liners are used in class 4 service dependent on application specifics.

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 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 2:1, 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 some 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.

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 usually 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 corrosion-resistant, but are limited to very light slurry applications.

Duplex stainless steels (ASTM A890 CD4MCu) are two-phase alloys, which contain both ferrite and austenite in the microstructure. This provides better corrosion resistance than high-alloyed martensitic steels, and better strength and erosion resistance than austenitic stainless steels. These steels are used for light slurries with aggressive carrier liquids.

Super-duplex stainless steels (ASTM A890 CD3MWCuN), in addition to main properties of duplex steels, provide further improved resistance to acids, acid-chlorides, caustic solutions, and other tough environments in the chemical/petrochemical, pulp, and paper industries, often replacing 300 series stainless steel, austenitic steels, nickel-based alloys, and more common duplex stainless-steel grades.

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 erosion 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 erosion resistance may decrease.

The following synthetic 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 erosion-resistant as natural rubber, but is better than most others.

Polyurethane is often used for fine-particle slurries (less than 210 μm or 0.008 in), 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 erosion 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 erosion 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. Erosion resistance is only fair. Carboxylated nitrile (XNBR) has better erosion resistance. Hydrogenated nitrile (HNBR) has better erosion 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 semiopen 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 balance quality grade G40 on the high (large amount of residual unbalance) side and grade G6.3 on the low (small amount of residual unbalance) side as defined in ISO1940/1 *Mechanical vibration - Balance quality requirements for rotors in constant (rigid) state.*

Slurry pump impellers are typically disk shaped and can be balanced in a single plane. Low specific-speed impellers are usually statically balanced on balance rails or a roller shaft arrangement. For greater sensitivity and accuracy, single-plane balancing rotating on a commercial balancing machine is preferred in accordance with ISO1940/1 section 4.5.2 and/or as demonstrated by the manufacturer's documented experience.

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 12.3.8.2a), 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 a static level or pressure differential, such as a horizontal bearing housing with a lubricant level above the shaft seal surface.

Labyrinth seals, Figure 12.3.8.2b, 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 12.3.8.2c, 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 12.3.8.2. 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.

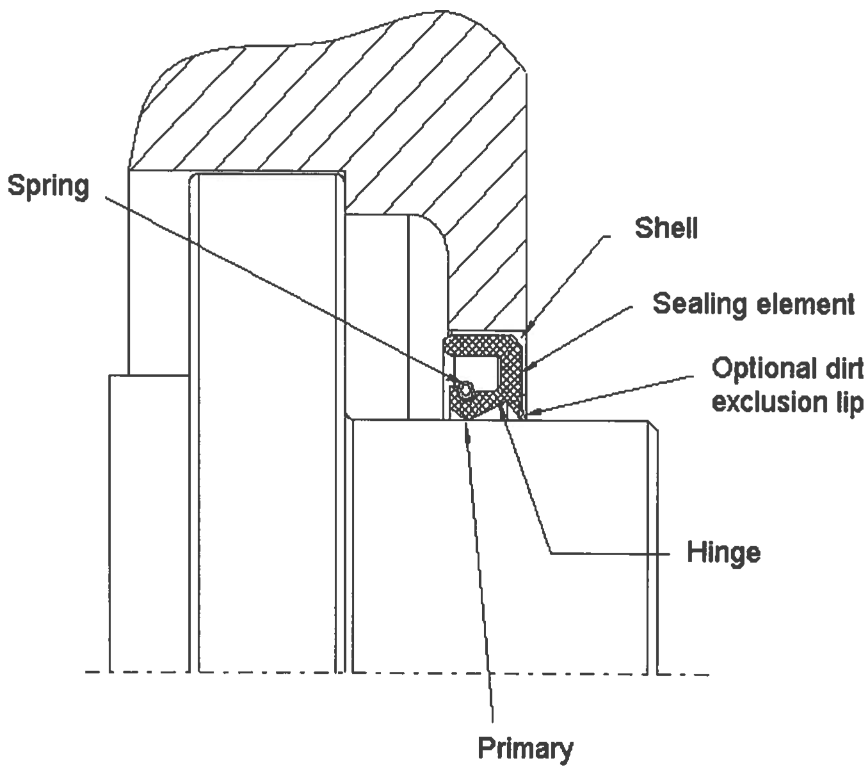


Figure 12.3.8.2a — Typical lip seal and its components

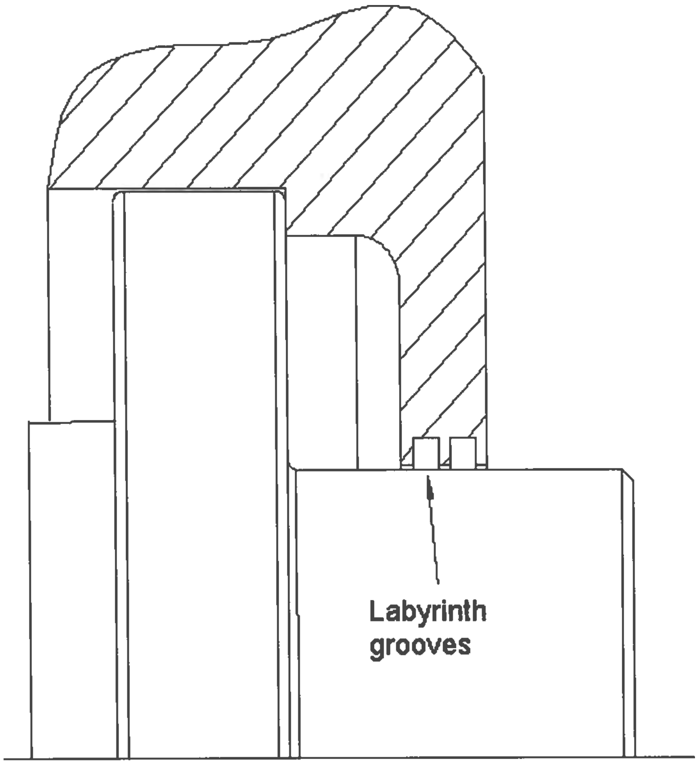


Figure 12.3.8.2b — Typical labyrinth seal

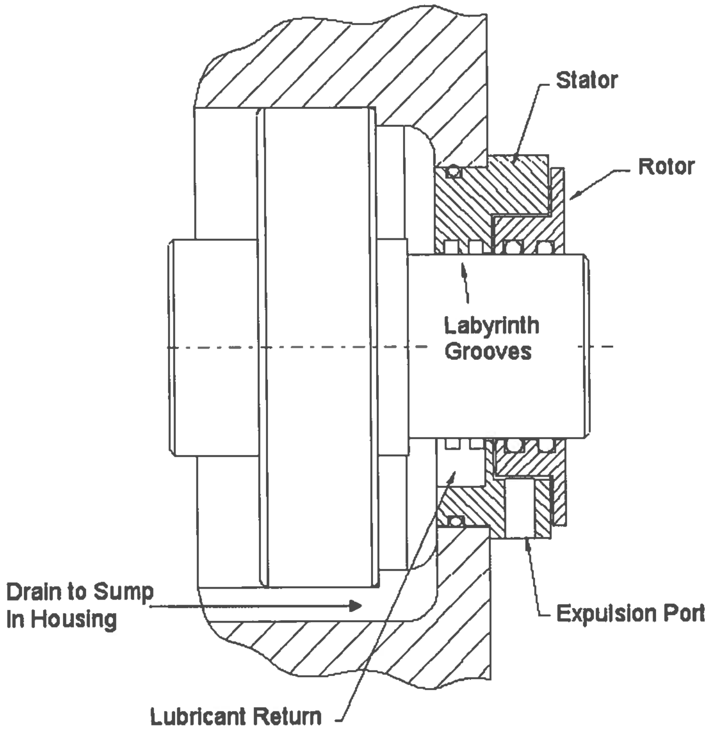


Figure 12.3.8.2c — Generic bearing isolator and its major components

Table 12.3.8.2 — Calculated fatigue life of bearings by slurry service class

Slurry service class (Ref. Figure 12.3.4.2a)	Minimum calculated bearing fatigue life (L_{10} life in hours)
1	17,500
2	35,000
3	50,000
4	50,000[a]

[a] For large pumps in class 4 service, calculated life should be increased by 2000 hours for every 25 mm (1 in) of suction diameter over 450 mm (18 in).

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 will shorten packing life.

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 12.3.8.3.1a, positions the lantern ring in front of the packing rings and a clean liquid is injected to prevent the slurry from reaching the packing. A flush injection at a pressure greater than the pump stuffing-box pressure is required. If the pressure is not known, the pump OEM should be contacted to verify the pressure. This arrangement is recommended for severely abrasive applications, wear service 3 and 4 (see Figure 12.3.4.2), that can tolerate considerable process fluid dilution. It is advantageous to use a restriction or throat bushing to reduce the flow of flush fluid into the product. The velocity of the flush fluid between the shaft sleeve and the bushing should be in the order of 3 to 5 m/s (10 to 15 ft/s) to ensure that no mixing of the slurry occurs with the flush fluid.

The "weep-type" arrangement (Figures 12.3.8.3.1b and c) restricts flow into the pump by positioning the lantern ring between the packing rings with a clean-liquid injection at a higher pressure, preferably 100 kPa (15 psi) above stuffing-box pressure. Product dilution is minimal with this arrangement, but the slurry can penetrate the packing and wear the packing sleeve or shaft if the packing has not been properly adjusted or if flush pressure is not reliable. This type of arrangement is recommended for wear service classes 2 and 3. Note that the type of packing on the product side should be abrasion-resistant, whereas the packing choice on the atmospheric side can be optimized for sealing.

Two different kinds of packing can be used and tailored for their primary function. One variation on this arrangement is the use of flow in and out of the stuffing box through the lantern ring (see Figure 12.3.8.3.1c). This increases the flow of flush fluid in the lantern ring. It can be used when the dead-ended injection in the lantern ring is insufficient to cool the packing in high-speed, high-service temperature, or high-pressure applications. This method can also be useful in regulating flush pressure while reducing flushing flow rate, and thus, product dilution.

The "dry-type" packing, Figure 12.3.8.6, 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 provides primary sealing 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 requires a packing that contains graphite or another solid lubricant. This type of seal is usually limited to wear service classes 1 and 2.

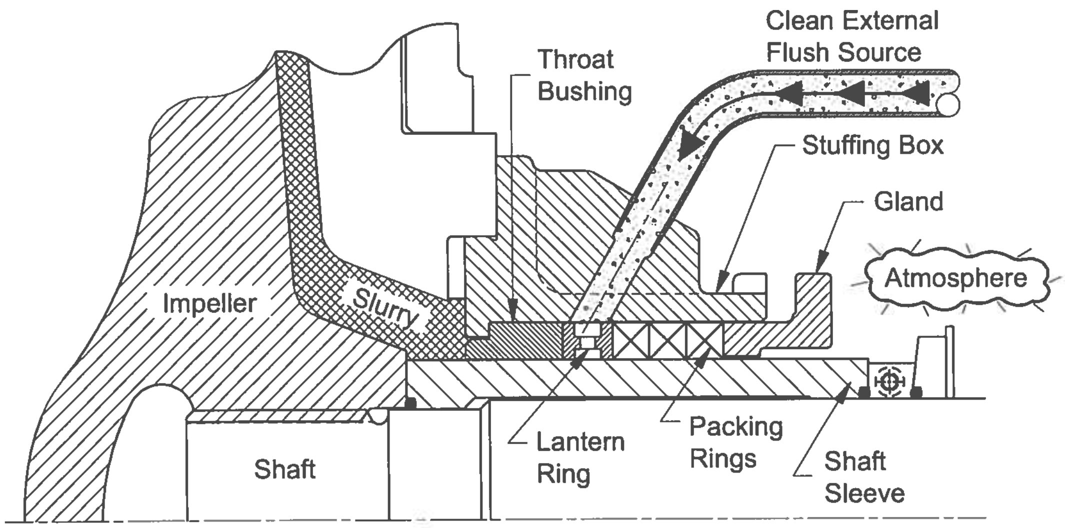


Figure 12.3.8.3.1a — "Flush-type" stuffing box with lantern ring in standard position

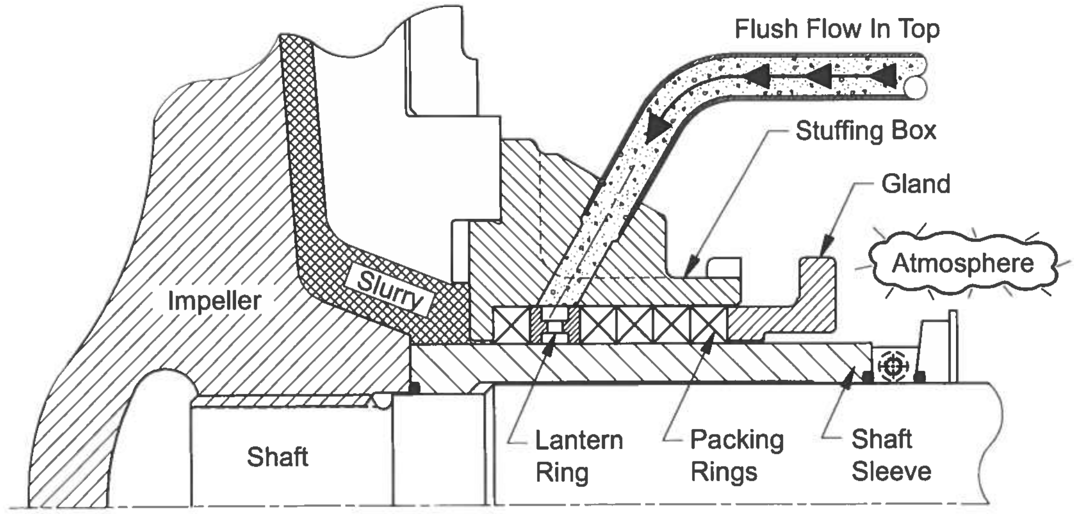


Figure 12.3.8.3.1b — "Weep-type" stuffing box with lantern ring

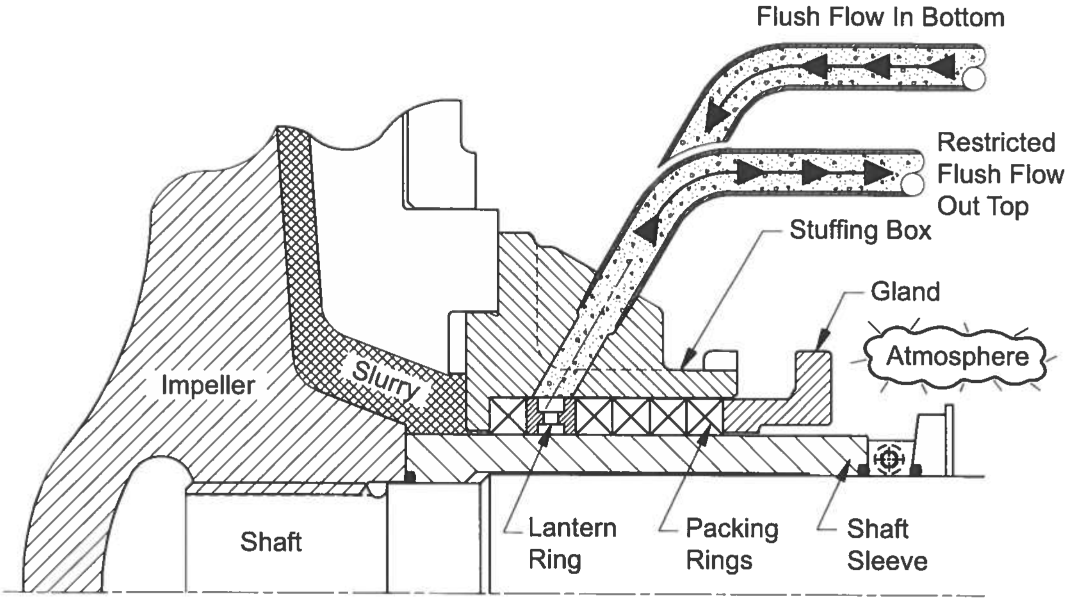


Figure 12.3.8.3.1c — Flow through flush

12.3.8.3.2 Packing installation and leakage

The stuffing box may or may not be packed before 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 tamped 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. Use a tamping tool to ensure uniform seating of each ring.

After the packing is fully installed, the gland should be left fairly loose. The flushing fluid should be applied, and then the pump started. 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. Initial leakage rates may be high until the packing is seated in and the packing gland has been properly adjusted to achieve a stabilized leakage rate.

Leakage rate will vary depending on the application, the type of packing used, pressure, shaft surface speed, and liquid sealed. For these reasons it is difficult to give or recommend a specific leakage rate. However the following formula can be used as a guide:

$$Q_{Leak} = \frac{2.5 \times \rho \times \Delta p \times \pi \times D \times w^3 \times SSF}{12 \times \nu \times L}$$

Where:

Q_{Leak} = Leakage rate, in mm^3/s (in^3/s)

ρ = Density, in kg/m^3 (lbm/ft^3) of sealed liquid

Δp = Pressure difference, in N/mm^2 (lbf/in^2), e.g., stuffing-box pressure - atmospheric, flush injection pressure - stuffing-box pressure

D = Shaft diameter, in mm (in)

w = Gap width, in mm (in)

ν = Kinematic viscosity, in m^2/s (ft^2/s) of sealed liquid

L = Length of packing set, in mm (in)

SSF = Surface speed factor, with SSF = rpm/1200

A value of w = 5 to 10 μm (0.20 to 0.40 10^{-3} in) is assumed for the gap. (This is dependent on the surface roughness of the shaft sleeve.) With a sleeve in good condition, the lower value should be used.

This formula tends to give values of what is to be expected with high-performance packing material. A practical guide for aqueous liquids would be to use 10 drops per minute per inch of shaft diameter. (There are between 8 to 18 drops in 1 mL, depending on the liquid properties). Note that leakage on the outside of the packing can be as much or more than the leakage from the shaft as a result of the condition of the stuffing-box bore and depending on how well the rings were cut and meet on their outside diameter.

When leakage cannot be controlled by adjusting the gland, all rings of packing should be replaced. The sleeve condition should be checked and the sleeve should be replaced if it shows significant scoring. Under no circumstance should the addition of a single ring of packing be used to control leakage.

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.

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 typically a combination of treated aramid yarn with an interlaced polytetrafluoroethylene (PTFE) multifilament yarn. Yarns made from carbon fibers are also used due to their high tensile strength and good thermal properties. Some braided construction use corners made from a high-strength, extrusion- and abrasion-resistant material with sides consisting of material with enhanced lubricity and superior thermal conductivity. 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 with anti-extrusion end rings and a pressurized gun to renew the material as it wears.

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 alloy steel with a minimum of 50 Rockwell C recommended for class 2.

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 with a hardness value greater than 60 Rc.

12.3.8.3.6 Centrifugal (dynamic) seals

A centrifugal seal is a dynamic seal that only operates when the pump is rotating and has no seal effect when the pump is not running. It consists of an expeller or set of expellers located in a separate chamber behind the impeller, which is typically fitted with expelling vanes on the back shroud.

When the pump is running, the centrifugal seal generates pressure p_e to equalize the pressure p_b, as shown in Figure 12.3.8.3.6, so that the pump operates without leakage.

A centrifugal (dynamic) seal needs 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 are that it must seal statically when the pump is shut down and it must run dry during pump operation. This can be accomplished by dry type packing, multiple lip seals, other proprietary devices, or mechanical seals with either dry run capabilities or fitted with a separate flush.

There is a maximum allowable suction pressure p_s above which, dependant on speed of rotation, a dynamic seal will not operate properly. For this reason, centrifugal seals are not effective on the second or higher stages of multiple pump installations, where the pumps are arranged to have the full discharge of the preceding stage applied to the suction of the following stage.

If the pumps are installed at specified intervals and elevations spread out along a slurry transport line, then 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 to 20% of the discharge pressure. An analysis should be made of the centrifugal seal performance, based on actual head, flow, and suction pressure, so that proper operation is ensured.

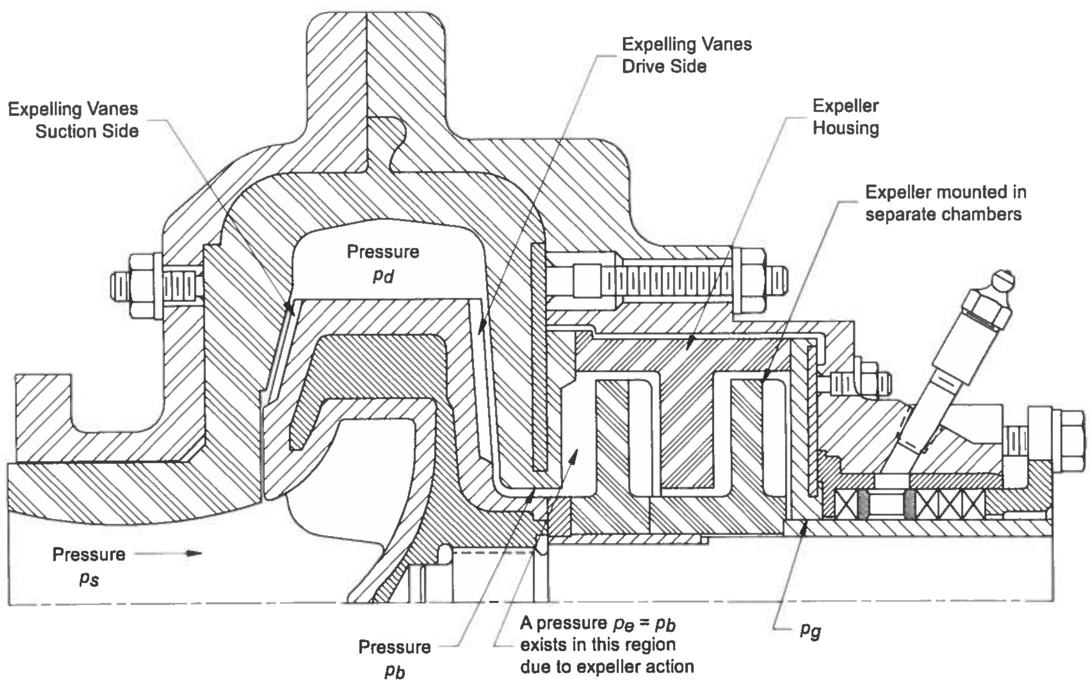


Figure 12.3.8.3.6 — Centrifugal (dynamic) seal with "dry-type" packing

12.3.8.3.7 Multiple lip seals

Multiple lip seals use sealing elements (lips) in contact with a rotating shaft sleeve to seal circumferentially.

In the typical arrangement, the sealing elements are facing into the fluid being pumped and therefore the hydraulic pressure in the seal chamber is acting as a closing force on the sealing elements.

Usually the sealing elements will be a PTFE or elastomer material running against a chrome oxide coating or against a shaft sleeve for service classes 3 and 4, as described in Section 12.3.8.3.5. The PTFE material is often blended with other materials, such as polyimide or carbon graphite, to enhance its properties for specific applications. The elastomers are usually an abrasion-resistant material, and they can be combined with PTFE or graphite to improve wear life during dry running, or manufactured in synthetic chemical-resistant materials. The seals can be built out of multiple elements (see Figure 12.3.8.3.7) and some are also available in unitized construction.

Normally these arrangements are used on lighter slurries at lower speed and pressure values, but performance can be enhanced by using the seals behind an expeller to reduce the pressure and influence of solids or with close-clearance backing rings to increase the lip support. The multiple lip seal can be nonlubricated (normally PTFE) or grease lubricated. For medium concentrations of solids, an external flush or quench (API Plan 32 or 62) is used to keep the environment around the seal clean. It can be injected in between lips to minimize the flow of flush liquid.

The advantages of the multiple lip seal are:

- There are no springs or other mechanical components to be clogged or fouled by the pumped product
- Compared to packing there is no, or less, leakage and no requirement for periodic adjustments

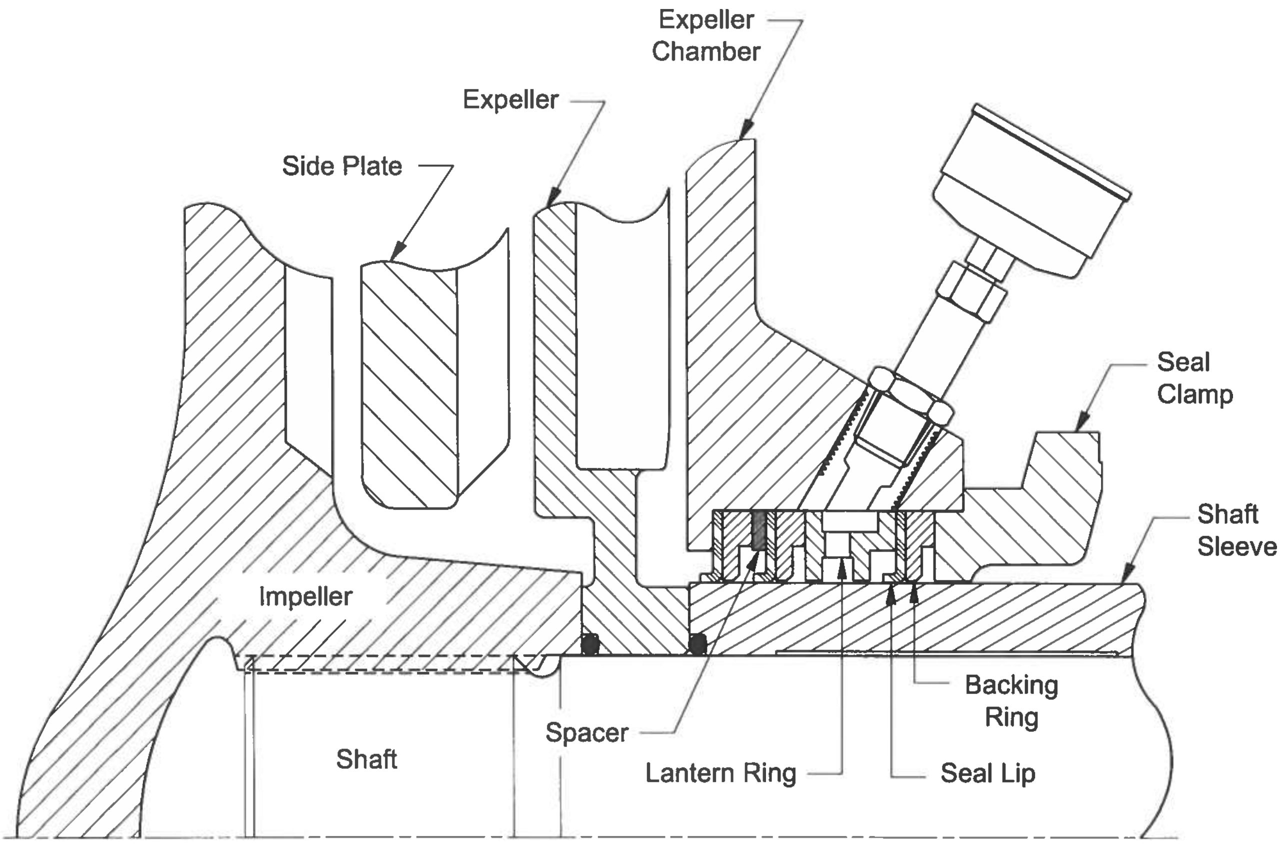


Figure 12.3.8.3.7 — Multiple elements lip seal with internal flush

- They are not sensitive to relative axial shaft motion

The disadvantages of the multiple lip seal are:

- Limited to relatively low speeds and pressure applications unless performance enhancing features are used
- Restricted to the temperature limitations of the sealing element material
- They are sensitive to relative radial shaft motion and centering, especially with close-clearance backing rings
- They require pump disassembly for replacement

12.3.8.3.8 Mechanical seal and seal chamber design

Mechanical shaft seals are used primarily in wear service classes 1 and 2 for nonsettling slurries where the solids size is small and concentrations are low. The overhung impeller, close-coupled, single-stage, end suction, metal, submersible type pump (refer to Figure 12.1.2p), commonly employs a mechanical seal. The heavier duty, horizontal type pumps (refer to Figure 12.1.2f) 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 12.3.8.3.8 may be used as a guide.

Table 12.3.8.3.8 — Application limits of single mechanical seals

Seal Type	Concentration % (by volume)	Specific Gravity	Average d50 Particle Size µm (in)
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.			

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 not recommended to install mechanical shaft seals in pumps with stuffing boxes designed for packing, unless an external injection (API Plan No. 32) is used. Seal chambers for single seals not using an external flush (API Plan No. 11) 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 (API Plan No. 02), 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 smaller 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 12.3.8.3.8 provides a selection guide to the applicability of various single seal types to the type of slurry being sealed when no external injection is used. 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 12.3.8.3.8a shows that

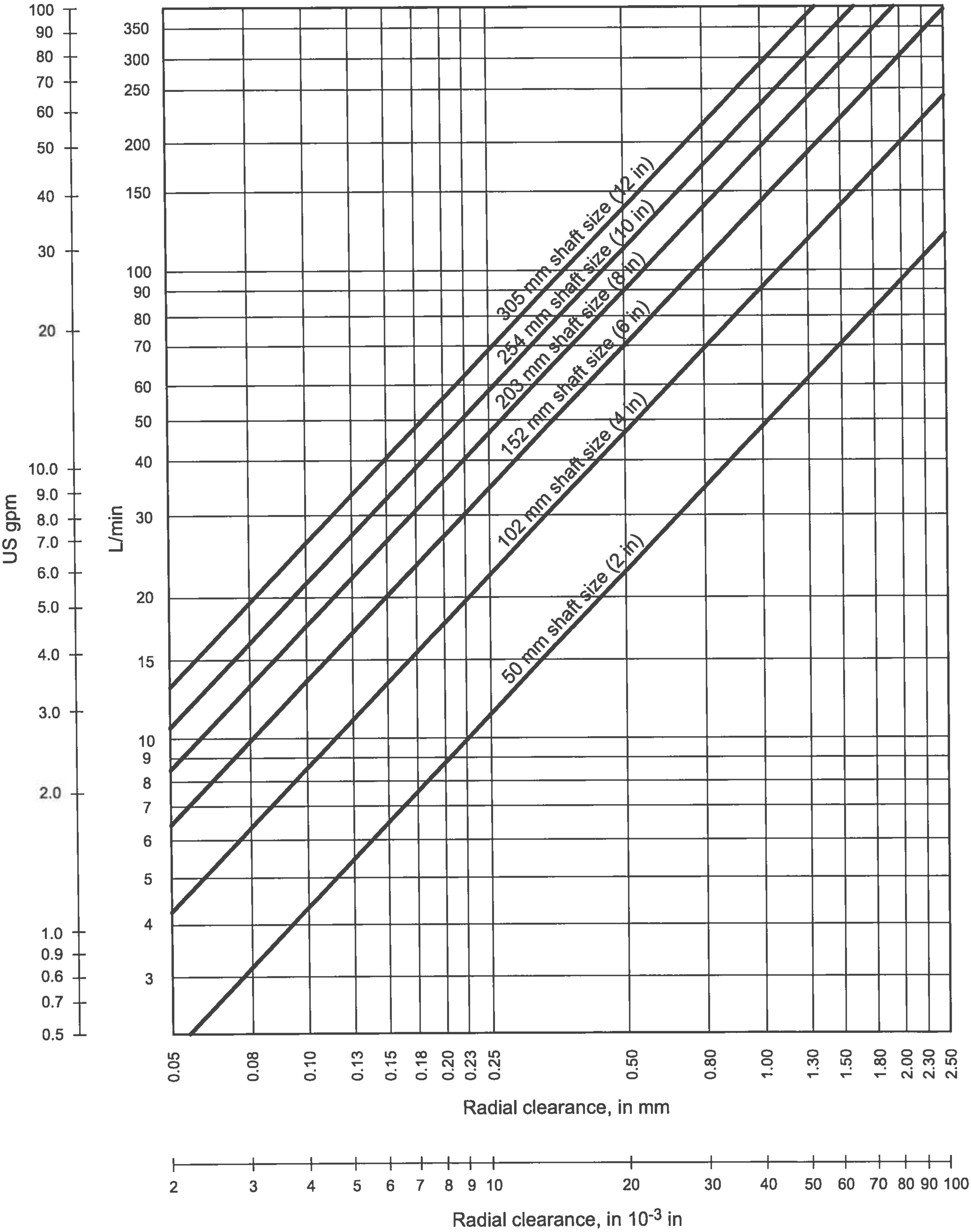


Figure 12.3.8.3.8a — Flow rates required to create 4.6 m/s (15 ft/s) velocity past a bushing

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 0.25 mm (0.010 in) can be accommodated with rotating elastomeric bellows nonpusher seals on pumps with shaft diameters less than 75 mm (3 in) depending on 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 nonconcentricity. Figure 12.3.8.3.8b 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 ensure 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.

Where more positive sealing of the quench fluid is required, dual unpressurized seals can be used, 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 API Plan 53 or 54 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 API Plan 32, 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 12.3.8.3.8c, 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 12.3.8.3.8 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.

12.3.8.3.9 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 (Figures 12.3.8.3.9a and 12.3.8.3.9b). The nonpusher-type elastomer bellows seal has the advantage of free movement of the front section of the bellows and uses a nonclogging single-spring design. *Nonpusher* refers to the movement of the secondary seal along the shaft or sleeve. When mounted as a stationary seal, as shown in Figure 12.3.8.3.9b, the probability of seal hang-up from leakage is decreased.

Heavy-duty pusher designs (Figures 12.3.8.3.9c and 12.3.8.3.9d). 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.

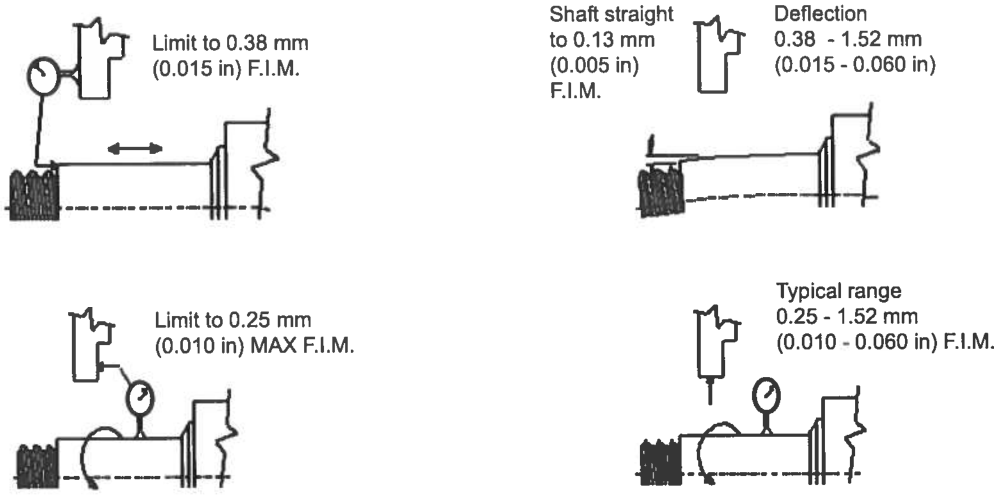


Figure 12.3.8.3.8b — Slurry pump shaft alignment and runout

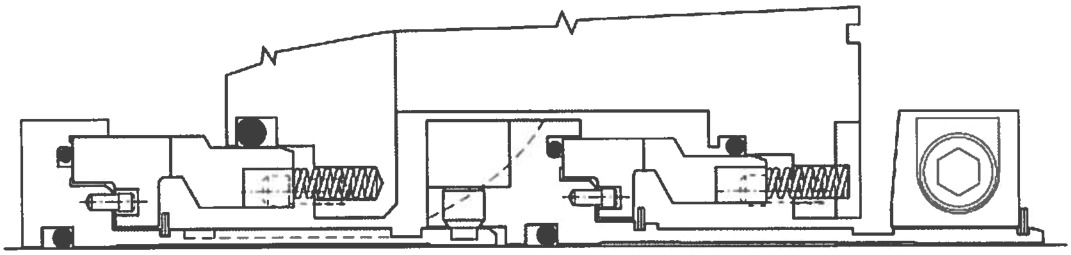


Figure 12.3.8.3.8c — Dual pressurized seal arrangement for slurry applications

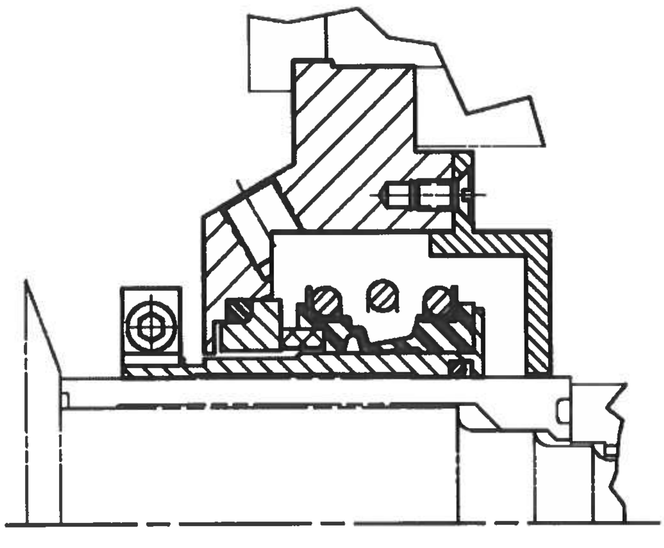


Figure 12.3.8.3.9a — Rotating elastomeric bellows seal

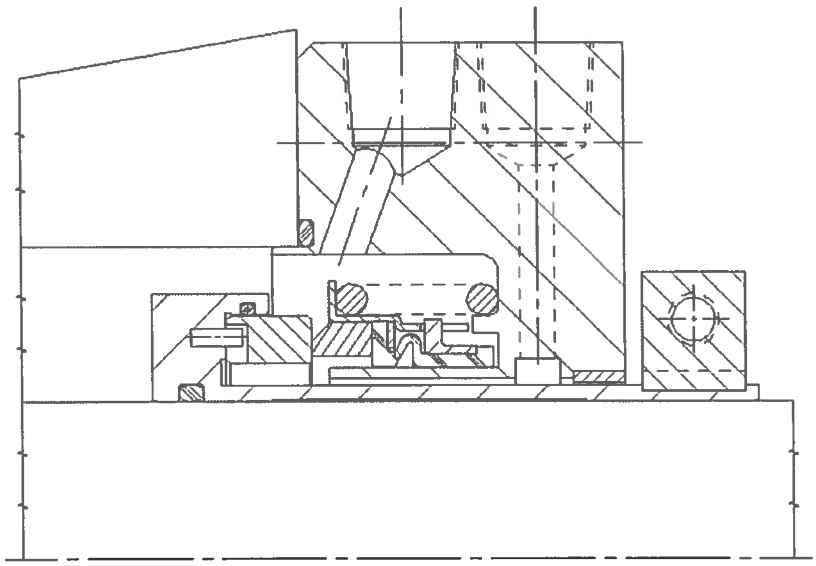


Figure 12.3.8.3.9b — Stationary elastomeric bellows seal

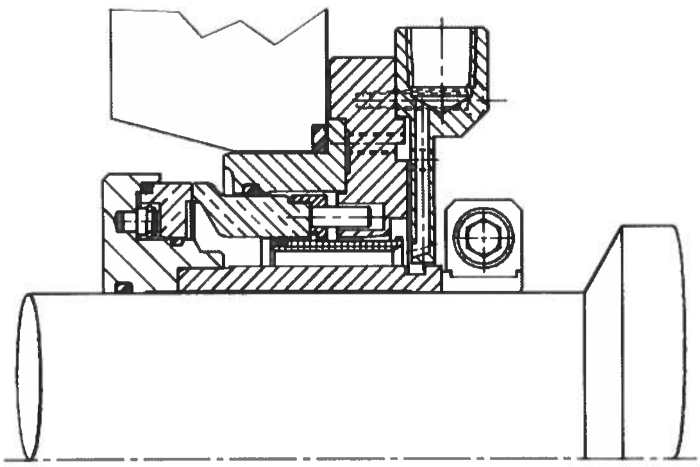


Figure 12.3.8.3.9c — Heavy-duty slurry seal with quench device

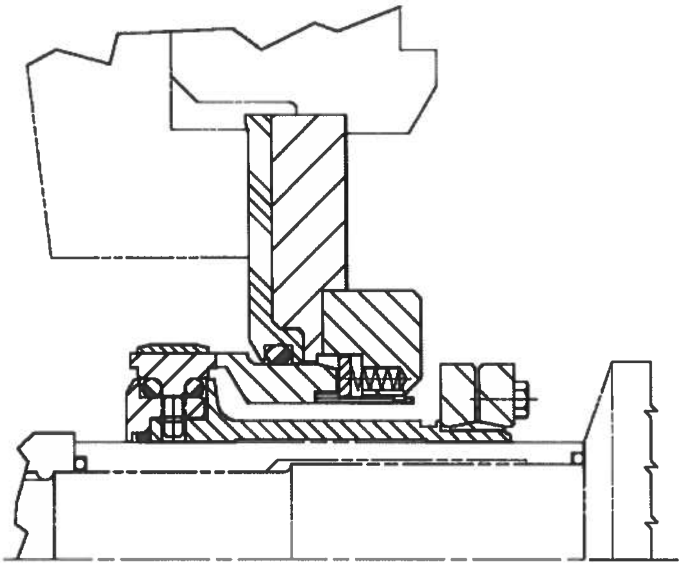


Figure 12.3.8.3.9d — Heavy-duty slurry seal with elastomeric liner

Pusher or nonpusher split seals (Figure 12.3.8.3.9e). 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).

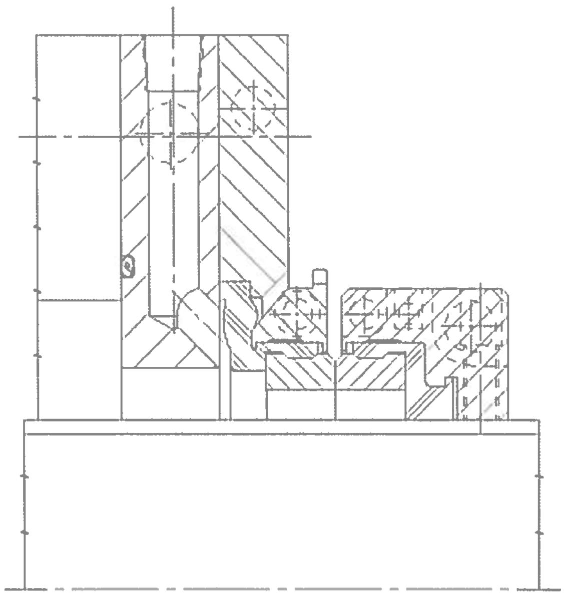


Figure 12.3.8.3.9e — Fully split nonpusher seal

Specialty designs (Figure 12.3.8.3.9f). Typically nonpusher designs using an elastomeric seal that may also act as the spring or use single coil or encapsulated finger-type springs that are nonclogging.

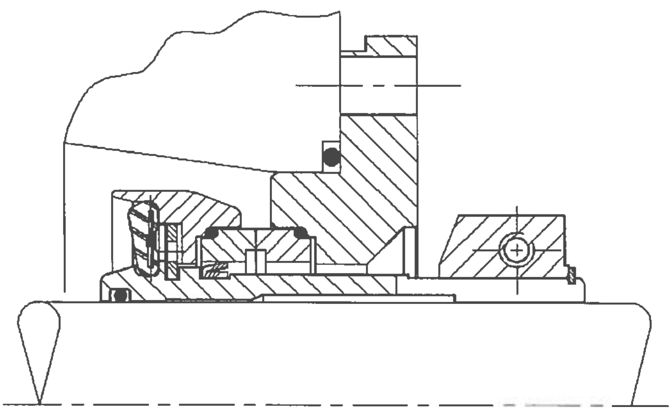


Figure 12.3.8.3.9f — Specialty slurry seal

12.3.8.3.10 Mechanical seal and packing flush arrangements

Flush arrangements used in conjunction with end face mechanical seals vary from no-flush designs (API Plan 02) to elaborate external circulating systems (API Plan 54). Following is a brief description of flush plans that can be used:

API Plan 02. 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.

API Plan 11. 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.

API Plan 32. 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 12.3.8.3.8a. 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.

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 12.3.8.3.10. This is attainable with relatively inexpensive filtration treatment equipment available and will provide for good packing life.

Table 12.3.8.3.10 — Recommended water quality limits

pH	Dissolved Solids	Suspended Solids	Filtration
6 – 8	< 1000 mg/L (< 0.13 oz/gal)	< 100 mg/L (< 0.013 oz/gal)	100% of 60-μm (0.0024-in) or larger particles removed

API Plan 52. 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.

API Plan 53. 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 an increased capital cost.

API Plan 54. 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.

API Plan 62. 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.

12.3.8.3.11 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 an API Plan 32.

- 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 usually should conform to appropriate ANSI, ISO, Japanese Industrial Standards (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 or variable-speed drives to meet the required service conditions. The type of driver used for nonsynchronous speeds normally 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 1.4 *Rotodynamic (Centrifugal) Pumps for Manuals Describing 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

This section includes recommendations for allowable nozzle loads for centrifugal end suction slurry pump types. In the absence of manufacturer's recommendations and when specified by the user, pumps supplied are expected to conform to these requirements.

Movement, or slippage, of the pump relative to the baseplate causes gross misalignment of the pump and motor shaft. Nozzle loads required to cause slippage for slurry pumps are typically less than those that would produce excessive stresses in the nozzle or internal distortion of parts. Based on this assumption, maximum allowable nozzle loads are determined by the ability of properly tightened hold-down bolts to prevent pump slippage and by allowable tensile stress limits in the hold-down bolts.

The orientations of the nozzle loads, as identified in Figure 12.4.2, are for those applied to lined and unlined metal pumps with bases mounted on the machined mounting surface(s) of a carbon-steel substructure or baseplate. Note that the discharge nozzle coordinate system always moves with discharge angle (F_z always in direction of flow).

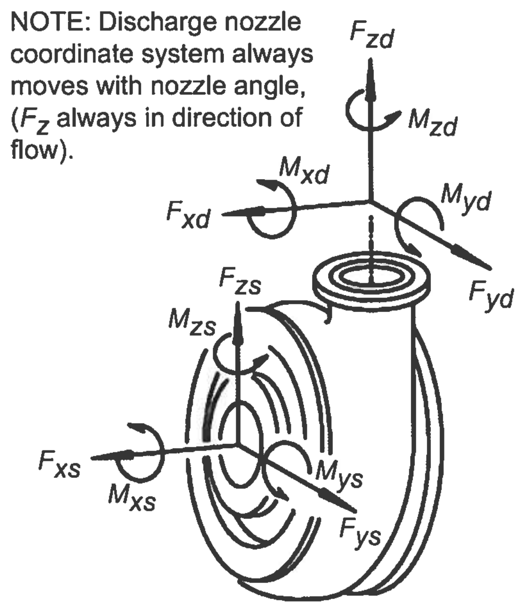


Figure 12.4.2 — Direction of forces and moments being applied to suction and discharge nozzles

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

Review of the effects of the nozzle loads on slurry pumps suggests that the common limiting factors are:

a) Allowable tensile stress of the hold-down bolts (pump casing to the baseplate).

b) Allowable displacement of the drive end of the shaft.

c) Allowable tensile load on the hold-down bolts (embedded in concrete).

d) Allowable bending stress in the nozzles.

For pumps equipped with mechanical seals, internal distortion of parts may cause excessive displacement of stationary and rotating seal components and become a limiting factor.

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. For the purpose of this document, yield strength is assumed at two thirds of ASTM A307 Grade A specified tensile strength of 413.7 MPa (60,000 psi).

The pump shall be bolted to the baseplate and sufficiently tightened to prevent slippage or movement relative to the baseplate. 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.

The criterion for acceptable nozzle loads with respect to casing hold-down bolts is that the forces and moments involved shall not move the pump. Although it is possible that some of the bolts will contact the edge of the bolt hole, the assumption is made that contact does not occur. Therefore the only restraining force exerted on the pump by the casing hold-down bolts is considered to be the friction developed between the pump and base when the casing hold-down bolts are tightened.

12.4.2.4 Assumed effect of nozzle loading

Forces in the x and y directions and moments about the z-axis (see Figure 12.4.2) for pumps with a top vertical discharge orientation result in a shear force between the pump base and the substructure. Movement of the pump occurs when the force overcomes the static horizontal friction force on all feet induced by the torque of the hold-down bolts.

Forces in the z direction and moments about the x- or y-axis result in a tensile load in some of the hold-down fasteners. Failure occurs if the tensile load is sufficient enough to relieve the clamp load allowing movement of the pump base or result in a stress level in the fastener that exceeds the yield stress of the material.

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

Procedure for assessment of combined applied loads for some common types of industrial pumps is described in ANSI/HI 9.6.2 *Rotodynamic Pumps for Assessment of Applied Nozzle Loads.* This procedure was developed to ensure that the various pumps designed for general industrial applications would meet common design standards.

Slurry pumps vary greatly in design and are typically more robust than general industrial pumps, thus adoption of the ANSI/HI 9.6.2 procedure and values may unnecessarily limit their application. However the assumptions and procedures used to determine the allowable nozzle loads are applicable to slurry pumps.

Typical allowable combined nozzle loads for centrifugal end suction slurry pumps are referenced in Section 12.4.2.10. Tables are found in Appendix C.

12.4.2.5 Allowable nozzle load based on hold-down capability criterion

The following four criteria 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 (275.8 MPa).

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

d) The maximum shear stress in the hold-down bolts must not exceed 25% of ASTM A307 Grade A fastener yield strength (275.8 MPa).

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

12.4.2.6 Allowable sideways slide force per bolt

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

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)

F_S = Factor of safety (2.0)

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

$F_a = 6.895 \times A$ (mm^2), N per bolt

$F_a = 1000 \times A$ (in^2), lbf per bolt

Conversion factors:

To convert megapascals (MPa) to pounds per square inch (psi), multiply by 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.

12.4.2.7 Sliding forces and moments

For Figures 12.4.2.7a and b, and 12.4.2.9, the sliding forces are F_{xs}, F_{ys}, F_{xd}, and F_{yd}, and the sliding moments are M_{zs} and M_{zd} for pumps with a top vertical discharge orientation. To keep the analysis simple, moments are summed about the geometric center of the hold-down bolt pattern and equally distributed to the hold-down bolts. The shear force that must be overcome with the clamp load, on a per bolt basis, is the sum of the force per bolt due to the sliding forces and the force per bolt due to sliding moments.

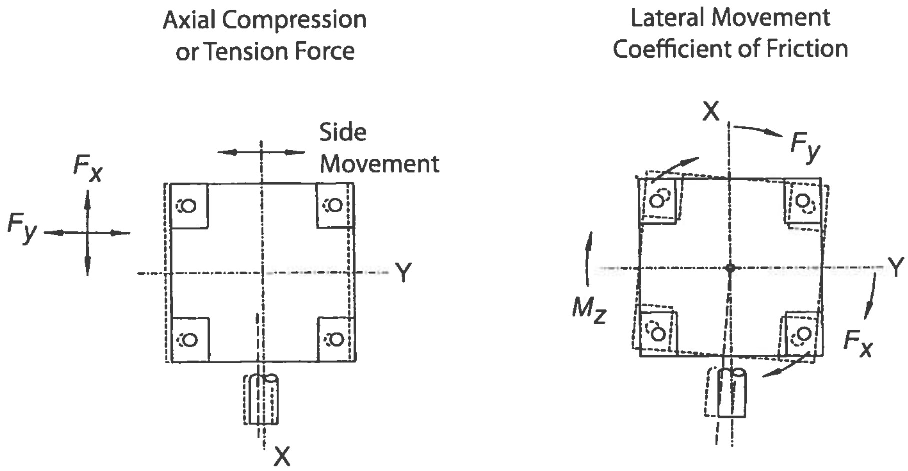


Figure 12.4.2.7a — Sliding movement when forces and moments are transferred to pump feet

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, MPa (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 248.22 MPa (36,000 psi). The total allowable force may be calculated as:

$$F_t = 248.22 \times A \times n, \quad \text{N}$$

or

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

Nozzle Configuration		
Type Figure		
	Connection	
Figure 12.4.2.7b	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

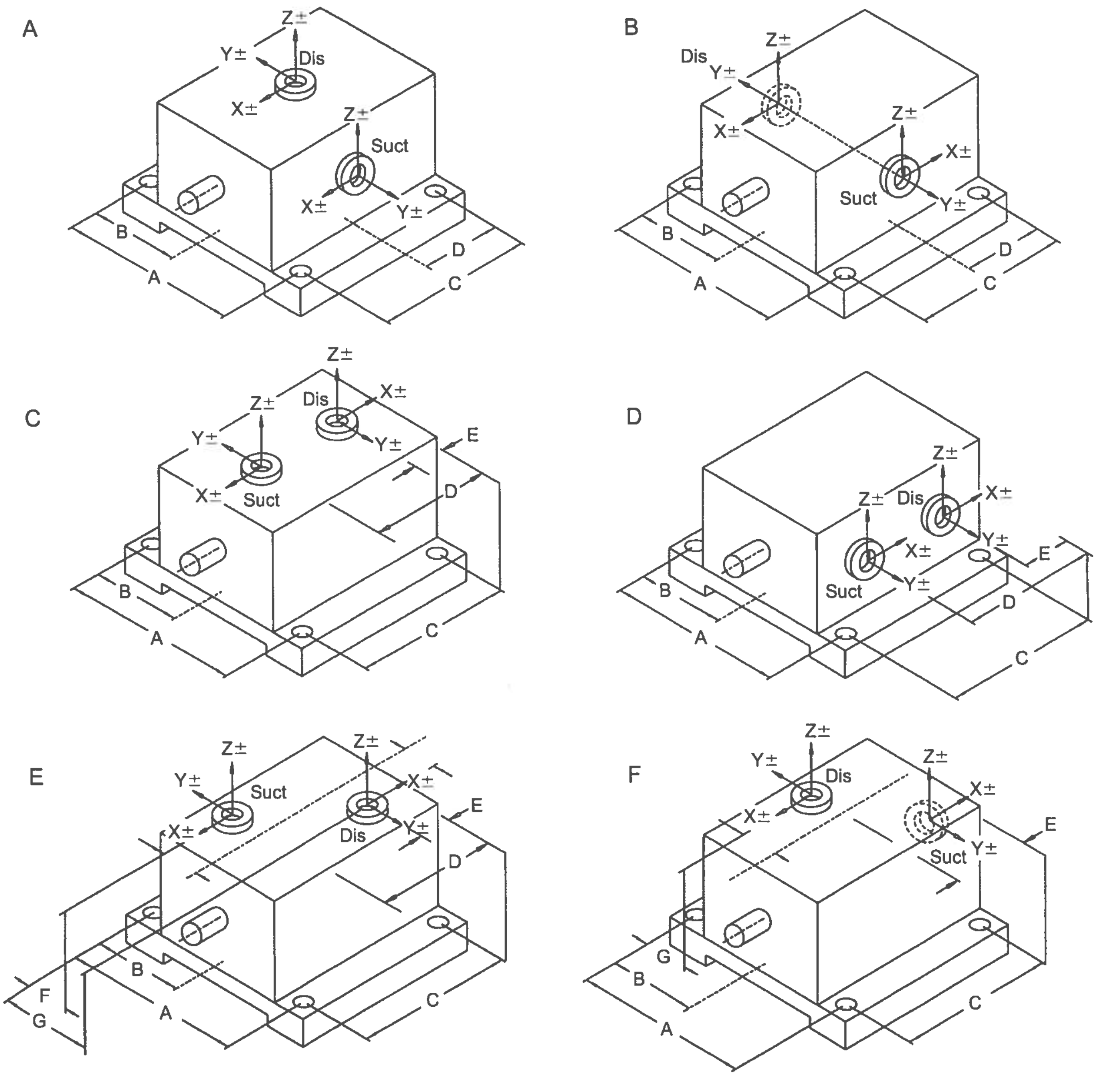


Figure 12.4.2.7b — Forces and moments applied to pumps with various suction and discharge nozzle locations

12.4.2.9 Bolts in tension

Table 12.4.2.9 — Calculation of allowable forces and moments on the suction(s) and discharge(d) connections

Total Allowable Force on the Bolts = F_t						
Figure 12.4.2.7b	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: Forces are in N (lbf) and moments are in N•m (lb•ft). 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 moments in lb•ft.

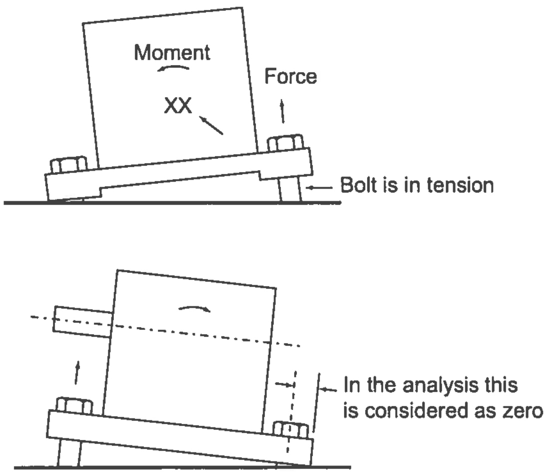


Figure 12.4.2.9 — Bolts in tension when a moment is applied to the pump

12.4.2.10 Precalculated values

The previous sections take into account the different manufacturers' product geometry and should be the main basis for allowable nozzle loads. They will be by necessity different for different geometries.

Values given in Appendix C have been used satisfactorily for some slurry pump configurations. These are provided as a precalculated set of common loads which, if by mutual agreement between the supplier and customer, may be used.

12.4.2.11 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 9.6.6 *Rotodynamic Pumps for Pump Piping* 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 stresses or strains greater than those allowed. 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 *Rotodynamic (Centrifugal) Pumps for Manuals Describing Installation, Operation, and Maintenance* 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.

Means should be provided to flush the pump if solids are allowed to settle inside, preventing impeller from rotation at start-up.

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 9.6.6 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 aboveground 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 sides (at least 45 degrees above horizontal) 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, use 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, then 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 mixture 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, then consider discharge area requirements.

f) If suction is taken from a sump, then 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 Sideliners

Uneven or localized wear is not uncommon. On many pump designs, the sideliners can be rotated to several different positions. When local wear occurs, these liners can be rotated 180 degrees 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 sloughing 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, then a variable-speed motor or drive 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 ANSI/HI 1.6 *Centrifugal Pump Tests*, to be superseded by ANSI/HI 14.6 *Rotodynamic Pumps - Hydraulic Performance Acceptance Tests.*

12.6.1.2 Performance tests

Slurry pumps are not routinely tested in the factory. If facility testing is specified, then it shall be done in accordance with ANSI/HI 1.6 *Centrifugal Pump Tests*, to be superseded by ANSI/HI 14.6 *Rotodynamic Pumps - Hydraulic Performance Acceptance Tests* 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 *Rotodynamic Pumps for Vibration Measurements and Allowable Values.*

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 on 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 on 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, then the pump may be tested at between 50% and 120% 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-µm (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 minimal 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. ASTM G75-95 *Standard Test Method for 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 Flowmeters

Magnetic flowmeters are preferred but a calibrated elbow meter (bend meter) with suitable piezometer flush points also can be acceptable. Wherever possible, the magnetic flowmeter 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, to be superseded by ANSI/HI 14.6, 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.

Appendix A

Equipment data sheets (informative)

This appendix is not part of ANSI/HI 12.1–12.6 and is included for informative purposes only.

Equipment Data Sheet
Rotodynamic Slurry Pump
Metric Units

Insert Company Logo Here

User/ Equipment Descriptive Information

User Name:	
Site/Plant Area:	
Equipment No.(s):	
Equipment Name/Service:	
Document No.:	
Contract/Requisition No.:	
System Description:	

Note: in this document all pressures are gauge pressures except the vapor pressure which is absolute.

System Operating Conditions (User to complete)

Fluid Description:			
Erosive Fluid: No ☐ Yes ☐	Due to:		
Max Particle Size (µm):	Miller number:		
Corrosive Fluid: No ☐ Yes ☐	Due to:		
pH:	Chlorides:		
Liquid (carrier) SG:			
Solids SG:			
Operating Conditions	Rated	Minimum	Maximum
Operating Temp (°C):			
Flow @ Temp (m^3/hr):			
% Time			
Suction Pressure (kPa):			
Discharge Pressure (kPa):			
Solids by weight (Cw %)			
Slurry SG:			
Solids d50 (µm):			
Solids d85 (µm):			
Yield stress (Pa):			
Viscosity (Pa·s)			
Total Differential Head (m):			
Vap Press @ Temp (kPa(abs) @ °C):			
NPSHA @ Flow & Temp (m):			

Supplier Information / Data

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

Construction (Supplier to complete unless specified)

Nozzles	Size	Rating	Facing	Location
Suction				
Discharge				

Jacketed:	Yes ☐	No ☐		
Max. Allowable Working Pressure:			kPa @	°C
Hydrostatic Test Pressure:			kPa @	°C
Connections:	☐ Casing Drain		☐ Casing Vent	
	☐ Gauge		☐ Valved/Plugged	
Impeller Type:	☐ Closed	☐ Semi-Open	☐ Open	☐ Other
Impeller Dia. (mm):	Max.		Supplied	
Impeller Attachment:	☐ Keyed		☐ Threaded	
Casing Mount:	☐ Frame		☐ Centerline	
	☐ Vertical		☐ Inline	
Casing Split:	☐ Axial	☐ Radial		
Thrust Bearing (Mfr/Model):				Location:
Radial Bearing (Mfr/Model):				Location:
Lubrication:	☐ Oil Mist	☐ Grease	☐ Oil	
Bearing	Drive End - Mfr./Model:			Dia. (mm)
Seals	Non-Drive End-Mfr./Model:			Dia. (mm)
Coupling: Mfr/ Model:			Driver Half Mounted By:	

Performance (Supplier to complete)

Proposed Curve No.:	Flow:	(m^3/hr)
No. Pumps:	Slurry Head per Pump (rated)	m
Head Ratio (H_r):	Water Head @ Speed:	m
Water Efficiency (%):	Efficiency Ratio (η_r):	
Rated Speed: RPM	NPSHR (m):	
Max Power Supplied Impeller @ Speed:		kW
Max Head Supplied Impeller @ Speed:		m
Minimum Continuous Flow:		(m^3/hr)
Rotation from Driver Side:	☐ CW	☐ CCW

Shaft Seal (User ☐ or Supplier ☐ to complete, please indicate)			
Stuffing Box Type:	☐ Packing ☐ Centrifugal Seal	☐ Mechanical Seal ☐ Other:	
Packing Type:	Size:	No. of Rows:	
	Required Flush:	L/min @	kPa
Mechanical Seal Mfr:		Flush Plan(s):	
Rotating Face Mat'l:		Stationary Face Mat'l:	
Wetted/Nonwetted Metal Prts:		Secondary Seal Mat'l:	
Required:	☐ Flush:	L/min @	kPa
	☐ Quench:	L/min @	kPa

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

Testing (clear water)

Test Type	Test Standard	Witness
☐ Hydrostatic		☐
☐ Performance		☐
☐ NPSH		☐
☐ Vibration		☐
☐ Sound		☐
☐ Shop Inspection		☐
☐ Dismantle and Inspect after Test		☐

Shipping Weights and Dimensions (Supplier to Complete)	
Pump:	kg
Drive (Gearbox/Coupling or V-belt Drive)	kg
Driver (Motor, Engine, or Other):	kg
Base Plate:	kg
Total Shipping wt.:	kg
Dimensions (LxHxW)	m

Additional Notes (User or Supplier to complete)

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

Materials (Supplier to complete)
Casing:
Impeller:
Casing Liner(s):
Suction Cover Liner (side plate):
Stuffing Box Cover Liner (side plate):
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:
Guard, Coupling:

Insert Outline Drawing of Pump and Pump Driver Here

Equipment Data Sheet
Rotodynamic Slurry Pump
US Customary Units

Insert Company Logo Here

User/ Equipment Descriptive Information

User Name:	
Site/Plant Area:	
Equipment No.(s):	
Equipment Name/Service:	
Document No.:	
Contract/Requisition No.:	
System Description:	

Note: in this document all pressures are gauge pressures except the vapor pressure which is absolute.

System Operating Conditions (User to complete)

Fluid Description:			
Erosive Fluid: No ☐	Yes ☐	Due to:	
Max Particle Size (µm):		Miller number:	
Corrosive Fluid: No ☐	Yes ☐	Due to:	
pH:		Chlorides:	
Liquid (carrier) SG:			
Solids SG:			

Operating Conditions	Rated	Minimum	Maximum
Operating Temp (°F):			
Flow @ Temp (US GPM):			
% Time			
Suction Pressure (psi):			
Discharge Pressure (psi):			
Solids by weight (Cw %)			
Slurry SG:			
Solids d50 (µm):			
Solids d85 (µm):			
Yield stress (psi):			
Viscosity (lb/ft·s)			
Total Differential Head (ft):			
Vap Press @ Temp (psia @ °F):			
NPSHA @ Flow & Temp (ft):			

Supplier Information / Data

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

Construction (Supplier to complete unless specified)

Nozzles	Size	Rating	Facing	Location
Suction				
Discharge				

Jacketed:	Yes ☐	No ☐	
Max. Allowable Working Pressure:		psi @	°F
Hydrostatic Test Pressure:		psi @	°F
Connections:	☐ Casing Drain ☐ Gauge	☐ Casing Vent ☐ Valved/Plugged	
Impeller Type:	☐ Closed	☐ Semi-Open ☐ Open	☐ Other
Impeller Dia. (in):	Max.	Supplied	
Impeller Attachment:	☐ Keyed	☐ Threaded	
Casing Mount:	☐ Frame ☐ Vertical	☐ Centerline ☐ Inline	
Casing Split:	☐ Axial	☐ Radial	
Thrust Bearing (Mfr/Model):		Location:	
Radial Bearing (Mfr/Model):		Location:	
Lubrication:	☐ Oil Mist	☐ Grease	☐ Oil
Bearing	Drive End - Mfr./Model:		Dia. (in)
Seals	Non-Drive End-Mfr./Model:		Dia. (in)
Coupling: Mfr/ Model:		Driver Half Mounted By:	

Performance (Supplier to complete)

Proposed Curve No.:	Flow:	GPM
No. Pumps:	Slurry Head per Pump (rated)	ft
Head Ratio (H_r):	Water Head @ Speed:	ft
Water Efficiency (%):	Efficiency Ratio (η_r):	
Rated Speed: RPM	NPSHR (ft):	
Max Power Supplied Impeller @ Speed:		HP
Max Head Supplied Impeller @ Speed:		ft
Minimum Continuous Flow:		(GPM)
Rotation from Driver Side:	☐ CW	☐ CCW

Shaft Seal (User ☐ or Supplier ☐ to complete, please indicate)			
Stuffing Box Type:	☐ Packing ☐ Centrifugal Seal		☐ Mechanical Seal ☐ Other:
Packing Type:	Required Flush:	Size: GPM @	No. of Rows: psi
Mechanical Seal Mfr: Rotating Face Mat'l: Wetted/Nonwetted Metal Prts:		Flush Plan(s): Stationary Face Mat'l: Secondary Seal Mat'l:	
Required	☐ Flush: ☐ Quench:	GPM @ GPM @	psi psi

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

Testing (clear water)		
Test Type	Test Standard	Witness
☐ Hydrostatic		☐
☐ Performance		☐
☐ NPSH		☐
☐ Vibration		☐
☐ Sound		☐
☐ Shop Inspection		☐
☐ Dismantle and Inspect after Test		☐

Shipping Weights and Dimensions (Supplier to Complete)	
Pump:	lb
Drive (Gearbox/Coupling or V-belt Drive)	lb
Driver (Motor, Engine, or Other):	lb
Base Plate:	lb
Total Shipping wt.:	lb
Dimensions (L×H×W)	in

Additional Notes (User or Supplier to complete)

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

Materials (Supplier to complete)
Casing:
Impeller:
Casing Liner(s):
Suction Cover Liner (side plate):
Stuffing Box Cover Liner (side plate):
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:
Guard, Coupling:

Insert Outline Drawing of Pump and Pump Driver Here

Appendix B

Source material and references (informative)

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

B.1 Source material

American National Standards Institute/Hydraulic Institute Standards ANSI/HI 1.1 – 1.5; ANSI/HI 9.8; ANSI/HI 9.6; ANSI/HI 14.6

1) Addie, Graeme R.; Dunens, Egils; Pagalthivarthi, Krishnan V. and Sharpe, John (2002) Oil Sands Tailings Pump Cost Considerations, *Hydrotransport 15*, Banff, Canada

2) Addie, Graeme R.; Pagalthivarthi, Krishnan V; and Visintainer, Robert J. (1996) Centrifugal Slurry Pump Wear, Technology and Field Experience, *Proc. ASME Fluids Meeting*, San Diego, CA, pp. 703-716

3) Addie, G.R.; Visintainer, R.J.; and Roco, M.C. (1987) Experiences With a Numerical Method of Calculating Slurry Pump Casing Wear, *Proc. 4th International Pump Symposium and Short Courses*

4) Bross, S. and Addie, G.R. (2001) Prediction of Impeller Nose Wear Behavior in Radial Slurry Pumps, *4th International Conference on Multiphase Flow* (ICMF-2001), New Orleans, LA

5) Clift, Roland and Clift, Diana H.M. (1981) Continuous Measurement of the Density of Flowing Slurries, *Int. J. Multiphase Flow*, Vol. 7, No. 5 pp. 555-561

6) Sellgren, Anders and Addie, G.R. (1989) Effect of Solids on Large Centrifugal Pumps Head and Efficiency, *Proc. at CEDA Dredging Day*, Amsterdam, The Netherlands

7) Pagalthivarthi, K.V. and Helmly, H.W. (1990) Application of Materials Wear Testing to Solids Transport via Centrifugal Slurry Pumps, *ASTM Conference*, San Antonio, TX

8) Sellgren, A. and Addie, G. (1993) Solids Effect on the Characteristics of Centrifugal Slurry Pumps, *Proc. 12th Int. Conference on the Hydraulic Transport of Solids in Pipes*, Brugge, Belgium

9) Sellgren, Anders and Addie, Graeme (1997) Effect of Coarse and Heavy Solids Particles on Centrifugal Pumps Head and Efficiency, *Proc. 47th Canadian Chemical Engineering Conference*, Edmonton, AB

10) Walker, C.I. and Goulas, A. (1984) Performance Characteristics of Centrifugal Pumps When Handling non-Newtonian Homogeneous Slurries, *Proc. Inst. Mech. Eng.*, 198A(1), pp. 410-49

11) Wilson, K.C. (1979) Deposition-limit Nomograms for Particles of Various Densities in Pipeline Flow, *Proc. Hydrotransport 6, BHRA Fluid Engineering*, Cranfield, UK, 1-12

12) Wilson, K.C. (1986) Effect of Solids Concentration on Deposit Velocity, *J. Pipelines*, 5, 251-257

13) Wilson, K.C.; Addie, G.R.; Sellgren, A. and Clift R. (1997) Slurry Transport Using Centrifugal Pumps, 2nd edition, *Blackie Academic & Professional*, London

14) Mueller, Thomas (2003) Flange Load Evaluation Procedure (Rev. 02) *GIW Internal Memorandum to Graeme R. Addie* (4/23/03)

15) Materials: Elastomers. Technical bulletin 25, March. Warman International Inc. (1999)

16) Materials: Metals. Range of Metals in Slurry Pumps. Technical bulletin 26, May. Warman International Inc. (1999)

17) Whitlock, L.; Sellgren, A.; Wilson, K. (2001) Net Positive Suction Head Requirement for Centrifugal Slurry Pumps, *Handbook of Conveying and Handling of Particulate Solids*

B.2 References

1. Addie, G. and Bross, S. (2000) Modeling of Slurry Pump Nose Wear, *15th Annual Regional Phosphate Conference*, Lakeland, FL

2. Addie, Graeme R. and Carlson, Richard R. (1998) Tools and Technology for Selecting Slurry Pumps, *CIM* Montreal '98

3. Addie, Graeme R. and Sellgren, Anders (1998) Design, Selection, Sizing and Control Considerations for Cyclone Feed Slurry Pumps, *9th Annual Intl. Freight Pipeline Symposium* – Powder Tech 3824, Monterrey, Mexico

4. Addie, Graeme R.; Wilson, K.C.; Sellgren, Anders (1999) Suction Performance of Centrifugal Pumps and its Effect on Cost of Slurry Transport, *Proc. Hydrotransport 14*, (BHRA) Fluid Engineering, Cranfield, UK

5. Addie, Graeme R. and Sellgren, A. (1998) Effect of Wear on the Performance of Centrifugal Slurry Pumps, *Proc. ASME Fluids Engineering Summer Meeting*, Washington, D.C.

6. ASM International (2000) Worldwide Guide to Equivalent Irons and Steels, 4th edition, *ASM International*, Materials Park, OH. www.asm-intl.com

7. Aude, T.C.; Cowper, N.T.; Thompson, T.L.; and Wasp, E.J. (1971) *Slurry Piping Systems Trends*, Chemical Engineering, June

8. Clift, R.; Wilson, K.C.; Addie, G.R.; and Carstens, M.R. (1982) A Mechanistically Based Method for Scaling Pipeline Tests for Settling Slurries, *Proc. Hydrotransport 8*, (BHRA) Fluid Engineering, Cranfield, UK, pp. 91-101

9. Pagalthivarthi, K.V.; Desai, P.V.; and Addie, G.R. (1991) Quasi-3d Computation of Turbulent Flow in Centrifugal Pump Casings, *Proc. ASME Symposium on Multidiciplinary Application of CFD*

10. Roudnev, A.S. (2004) Slurry pump suction performance considerations, *Proc. 16th Intl. Conference on Hydrotransport*, BHR Group, Santiago, Chile, pp. 137-150

11. Roudnev, A.S. (1999) Some Aspects of Slurry Pump Design, *World Pumps*, Elsevier, UK, Number 389, pp. 58-61

12. Sellgren, A. (1979) Performance of Centrifugal Pump When Pumping Ores and Industrial Minerals, *Proc. 6th International Conference on the Hydraulic Transport of Solids in Pipes*, (BHRA) Fluid Engineering, Cranfield, Bedford, UK, pp. 291-304

13. Sellgren, Anders and Addie, G.R. (1996) Pump and Pipeline Solids Effects of Transporting Sands with Different Size Distributions and Concentrations, *Proc. 13th Int. Conference on the Hydraulic Transport of Solids in Pipes*, (BHRA) Fluids Engineering Johannesburg, South Africa, pp 227-236

14. Sellgren, Anders and Addie, Graeme (1997) Cost-effective Pumping of Coarse Mineral Products Using Fine Sands, *Journal of Powder Technology* 94:191-194 Article: 3345

15. Sellgren, Anders; Addie, G.R.; and Yu, Wei Chung (1997) Effect of Non-Newtonian Mineral Suspensions on the Performance of Centrifugal Pumps, *The Rheology in the Mineral Industry Conference*

16. Sellgren, Anders; Addie, G.R.; and Pagalthivarthi, K.V. (2002) Wear and the Total Cost of Ownership of Slurry Pumps, *ASME Fluids Engineering Division Summer Meeting*

17. Schnarr, P.; Shaeffer, L.; and Weinard, H. (2002) Elastomers in the Mineral Processing Industry, *Proc. SME Mineral Processing Plant Design, Practice, and Control*, Vol. 2, pp. 1932-1952

18. Shook, C.A. and Roco, M.C. (1991) Slurry Flow Principles and Practice, *Butterworth-Heinemann*, USA

19. Sundqvist, Asa; Sellgren, A.; and Addie, G.R. (1996) Pipeline Friction Losses of Coarse Sand Slurries, Comparison with a Design Model. *Powder Technology, VDX, Division of Water Resources Engineering*

20. Van den Berg, C.H.; Vercruijsse, P.M.; and Van den Broeck (1999) The Hydraulic Transport of Highly Concentrated Sand-Water Mixtures Using Large Pumps and Pipeline Diameters, *Proc. Hydrotransport 14*, (BHRA) Fluid Engineering, The Netherlands

21. Walker, C.I.; Wells, P.J.; and Pomat, C. (1992) The Effect of Impeller Geometry on the Performance of Centrifugal Slurry Pumps, *Proc. 4th Int. Conf. On Bulk Materials Handling and*

Transportation, IE Aust. Wollongong, Australia, pp. 97-101

22. Pumping Non-Newtonian Slurries, Technical bulletin 14, Oct. Warman International Inc. (1991)

23. Influence of Slurry on Pump Performance. Technical bulletin 22, Jan. Warman International Inc. (1993)

24. Slurry Pump Manual, 2nd edition. Envirotech Pumpsystems (now Weir Slurry Group, Inc.) (1999)

25. Centrifugal Slurry Pump Equipment Data Sheet (Metric and US) Hydraulic Institute. ANSI/HI 12.1-12.6, 2011, Appendix A. http://www.pumps.org/public/slurrypump/downloads.htm

B.3 Standards organizations

The following list includes Web site 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

Nozzle load tables (informative)

Table C.1 — Typical allowable combined nozzle loads for centrifugal end suction slurry pumps – metric

Branch Size	Discharge						Suction					
	F_{XD} (N)	F_{YD} (N)	F_{ZD} (N)	M_{XD} (N·m)	M_{YD} (N·m)	M_{ZD} (N·m)	F_{XS} (N)	F_{YS} (N)	F_{ZS} (N)	M_{XS} (N·m)	M_{YS} (N·m)	M_{ZS} (N·m)
2	7110	5690	14,450	3570	3570	5420						
3	7840	6270	15,180	3930	3930	5960	15,180	7840	6270	5960	3930	3930
4	8590	6890	15,930	4290	4290	6500	15,930	8590	6890	6500	4290	4290
6	10,110	8090	17,450	4990	4990	7570	17,450	10,110	8090	7570	4990	4990
8	11,700	9340	19,040	5690	5690	8620	19,040	11,700	9340	8620	5690	5690
10	13,390	10,710	20,730	6380	6380	9670	20,730	13,390	10,710	9670	6380	6380
12	15,230	12,180	22,560	7070	7070	10,710	22,560	15,230	12,180	10,710	7070	7070
14	17,300	13,830	24,640	7740	7740	11,730	24,640	17,300	13,830	11,730	7740	7740
16	19,760	15,790	27,100	8410	8410	12,750	27,100	19,760	15,790	12,750	8410	8410
18	22,750	18,190	30,090	9070	9070	13,750	30,090	22,750	18,190	13,750	9070	9070
20	26,240	20,990	33,580	9730	9730	14,740	33,580	26,240	20,990	14,740	9730	9730
22	29,730	23,790	37,070	10,380	10,380	15,720	37,070	29,730	23,790	15,720	10,380	10,380
24	32,720	26,190	40,060	11,020	11,020	16,700	40,060	32,720	26,190	16,700	11,020	11,020
26	35,170	28,150	42,510	11,650	11,650	17,660	42,510	35,170	28,150	17,660	11,650	11,650
28	37,250	29,800	44,590	12,280	12,280	18,600	44,590	37,250	29,800	18,600	12,280	12,280
30	39,090	31,260	46,430	12,900	12,900	19,540	46,430	39,090	31,260	19,540	12,900	12,900
32	40,780	32,640	48,120	13,510	13,510	20,470	48,120	40,780	32,640	20,470	13,510	13,510
34	42,370	33,380	49,710	14,110	14,110	21,390	49,710	42,370	33,890	21,390	14,110	14,110
36	43,890	35,090	51,230	14,710	14,710	22,290	51,230	43,890	35,090	22,290	14,710	14,710

NOTES:

Allowable combined nozzle loads applicable to slurry pumps of both lined and unlined designs.

Based on calculation method described in Section 12.4.2.8.

Coordinate system per Figure 12.4.2.

Values in the table are to be used as a first guide only in absence of the manufacturer's recommendations. Higher allowable loads may be possible; lower allowable loads may exist depending on individual pump configuration and operating conditions.

Table C.2 — Typical allowable combined nozzle loads for centrifugal end suction slurry pumps – US customary units

Branch Size	Discharge						Suction					
	F_{XD} (lb)	F_{YD} (lb)	F_{ZD} (lb)	M_{XD} (ft•lb)	M_{YD} (ft•lb)	M_{ZD} (ft•lb)	F_{XS} (lb)	F_{YS} (lb)	F_{ZS} (lb)	M_{XS} (ft•lb)	M_{YS} (ft•lb)	M_{ZS} (ft•lb)
2	1600	1280	3250	2640	2640	4000						
3	1760	1410	3410	2900	2900	4390	3410	1760	1410	4390	2900	2900
4	1930	1550	3580	3160	3160	4790	3580	1930	1550	4790	3160	3160
6	2270	1820	3920	3680	3680	5580	3920	2270	1820	5580	3680	3680
8	2630	2100	4280	4200	4200	6360	4280	2630	2100	6360	4200	4200
10	3010	2410	4660	4700	4700	7130	4660	3010	2410	7130	4700	4700
12	3420	2740	5070	5210	5210	7900	5070	3420	2740	7900	5210	5210
14	3890	3110	5540	5710	5710	8650	5540	3890	3110	8650	5710	5710
16	4440	3550	6090	6200	6200	9400	6090	4440	3550	9400	6200	6200
18	5110	4090	6760	6690	6690	10,140	6860	5110	4090	10,140	6690	6690
20	5900	4720	7550	7170	7170	10,870	7550	5900	4720	10,870	7170	7170
22	6680	5350	8330	7650	7650	11,600	8330	6680	5350	11,600	7650	7650
24	7350	5890	9000	8120	8120	12,310	9000	7350	5890	12,310	8120	8120
26	7900	6330	9550	8590	8590	13,020	9550	7900	6330	13,020	8590	8590
28	8370	6700	10,020	9050	9050	13,720	10,020	8370	6700	13,720	9050	9050
30	8780	7030	10,430	9510	9510	14,410	10,430	8780	7030	14,410	9510	9510
32	9160	7340	10,810	9960	9960	15,100	10,810	9160	7340	15,100	9960	9960
34	9520	7620	11,170	10,410	10,410	15,770	11,170	9520	7620	15,770	10,410	10,410
36	9860	7890	11,510	10,850	10,850	16,440	11,510	9860	7890	16,440	10,850	10,850

NOTES:

Allowable combined nozzle loads applicable to slurry pumps of both lined and unlined designs.

Based on calculation method described in Section 12.4.2.8.

Coordinate system per Figure 12.4.2.

Values in the table are to be used as a first guide only in absence of the manufacturer's recommendations. Higher allowable loads may be possible; lower allowable loads may exist depending on individual pump configuration and operating conditions.

Appendix D

Materials data (informative)

D.1 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
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

Material Name								
Stainless, Martensitic, CA15 (400 series)				Typical Application: 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

D.2 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
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

Mineral	Minimum	Maximum
Corundum	3.90	4.10
Covellite	4.60	
Cuprite	5.70	6.10
Cyanite	3.50	3.70
Diamond, Black	2.75	3.42
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

Mineral	Minimum	Maximum
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
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 E

Index (informative)

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

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

This page intentionally blank.

B128

Hydraulic Institute
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

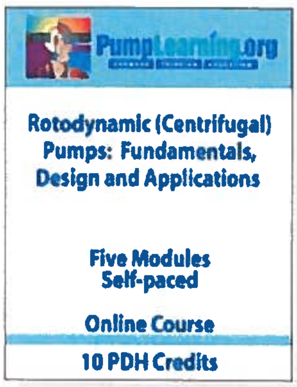
- Hardcopy
- Downloadable

Complete Set — Hardcopy

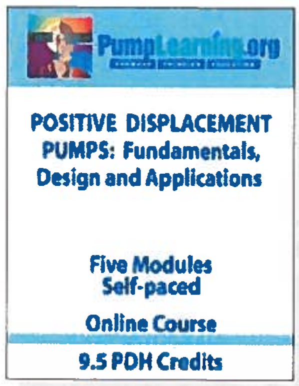
ANSI/HI Pump Standards on CD — complete set

ANSI/HI Pump Standards by Subscription

Pumping System Optimization Course
Benefit from Major Energy Efficiency Improvements and Bottom-line Savings.



Rotodynamic (Centrifugal) Pumps: Fundamentals, Design and Applications Online Course



Positive Displacement Pumps: Fundamentals, Design and Applications Online Course

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

www.Pumps.org | www.PumpSystemsMatter.org | www. PumpLearning.org