American National Standard for

Vertical Pumps

for Nomenclature, Definitions, Application and Operation

ANSI/HI 2.1-2.5-1994

Hydraulic Institute

9 Sylvan Way
Parsippany, New Jersey
07054-3802

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Sponsor

Hydraulic Institute

Approved May 23, 1994

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Contents

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

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

Revisions

The Standards of the Hydraulic Institute are subject to review, and revisions are undertaken whenever it is found necessary because of new developments and progress in the art.

Scope

This Standard applies to vertical centrifugal pumps that are driven by electric motors or engines with right angle gears. it includes types and nomenclature; definitions; design and application; and installation, operation and maintenance.

Units of Measurement

US Customary units of measurement are predominantly used, and, where appropriate, Metric unit equivalents appear in brackets following the US units. Sample calculations are shown with US units only.

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

The following organizations, recognized as having an interest in the standardization of vertical 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.

Agrico Chemical Corporation
American Petroleum Institute
Amer. Society of Heating, Refrigerating & Air-Conditioning Engineers
Amer. Society of Mechanical Engineers
Amoco Oil Company
Aurora Industries
Bechtel Corporation
Black & Veatch
BP America
Brown & Caldwell
Camp Dresser & McKee, Inc.
CH2M Hill
Chiyoda International Corporation
Commonwealth Edison
DeWanti & Stowell
Dexter Corporation
DuPont Engineering
Edison Electrical Institute
Electric Power Research Institute
Florida Power Corporation
Florida Power & Light
Fluor Daniel
Freese and Nichols, Inc.
G.E. Motors
HDR Engineering
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John Crane, Inc.
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Naval Sea Systems
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Newport News Shipbuilding
Pacific Gas & Electric
Raytheon Engineers & Constructors
Riverwood International Georgia, Inc.
San Francisco Bureau of Engineering
Siemens Energy & Automation
Simons-Eastern Consultants
Sordoni-Skanska Construction Co.
Star Enterprises
State Farm Mutual Automobile Ins. Co.
State of California Dept. of Water Res.
Stone & Webster
Summers Engineering, Inc.
T. Hopkins - Consultant
Tennessee Eastman
Union Carbide Chemicals & Plastics Co.
US Bureau of Reclamation

2 Vertical pumps

2.1 Types and nomenclature

2.1.1 Definition of vertical pumps

1) All vertical pumps contain one or more bowls (diffusers);

2) The pumps are equipped with one of the following four types of impellers:

a) radial flow;

b) modified radial flow (turbine pumps);

c) mixed flow;

d) axial flow (propeller pumps).

3) The pumps, particularly the radial flow and modified radial flow types, are usually designed for multistaging, by bolting or threading individual bowls together;

4) The pumping element (bowl assembly) is usually suspended by a column pipe, which also carries the liquid from the bowl to the discharge opening;

5) The driver is mounted either:

a) on the discharge head (lineshaft pumps);

b) directly to the bowl assembly, either above or below (pumps with submersible motors);

c) in a horizontal configuration, such as an electrical motor or engine, driving through a right angle gear.

2.1.2 Types of vertical pumps

See Figure 2.1 Vertical pump types.

2.1.2.1 Deep well (lineshaft)

This type of vertical pump is commonly installed in a drilled and cased well. Its function is to lift liquid (usually water) from the water level in the well to the surface and provide a specified discharge pressure at the surface (see Figure 2.4). The pumping element consists of a single or multistage bowl assembly and is located below the lowest pumping level. The bowl bearings are usually lubricated by the pumped liquid. The column pipe and lineshaft assembly is either an open type product lubricated assembly or enclosed type oil or external liquid lubricated assembly. The column pipe is supported at the surface by a discharge head. The discharge head directs the water from vertical to horizontal flow and also supports a driver. A shaft sealing arrangement is contained within the discharge head. This type of pump is self-priming.

2.1.2.2 Wet pit, short setting or close-coupled (lineshaft)

This type of vertical pump usually is suspended in a wet pit. See Figures 2.4 and 2.7. The pumping element can be fitted with a bowl assembly of any desired specific speed. Normally the bowl assembly bearings are product-lubricated; however, they can be force-lubricated by grease, water or other lubricants. The column pipe assembly supports the bowl assembly and houses a lineshaft. The lineshaft bearings are usually open type, product-lubricated. However, enclosed type lineshaft force-feed lubrication with oil, grease or water may also be supplied. Generally, enclosed type lineshaft, gravity-feed lubrication is not recommended except where the pressure head above discharge head control line is less than 30 ft. (10 m). A shaft sealing arrangement is contained within the discharge head on product-lubricated pumps. This type of pump is self-priming.

2.1.2.3 Barrel or can (lineshaft)

This type of pump is mounted in an enclosed container (barrel or can) and generally is used in booster applications and where inadequate suction pressure conditions exist (see Figure 2.6). The can pump contains the same pumping elements and column pipe as the wet pit type pumps. The lineshaft bearing assembly is almost always product-lubricated. The discharge head performs the same functions as the wet pit head except the base is sealed to atmosphere. Liquids other than water are commonly pumped by this type of pump. This type of pump is very effective where inadequate system NPSH is available. Additional NPSH is created by extending the pump can length and bowl assembly to create additional submergence (suction head).

2.1.2.4 Submersible

This type consists of an electric drive motor coupled directly to the bowl assembly. See Figure 2.5. The driving "submersible type" motor and bowl assembly are designed to be submerged in the fluid pumped. The pumping element usually is of the turbine bowl design; however, mixed flow and propeller types are also available. This type of unit is normally used in wells and occasionally for wet pit or canned booster service.

2.1.3 Classification by configuration

Listed below are the general configurations that describe vertical pumps.

2.1.3.1 Mounting, above and below floor discharge

Vertical pump bowls discharge the pumped liquid into a column, which takes it to the discharge.

There are two basic types of pump discharge configurations. Pumps with above floor discharge (see Figure 2.7) and pumps with below floor discharge (see Figure 2.9). The driver is mounted above the floor in both.

2.1.3.2 Hollow/solid shaft driver

The hollow shaft drivers (see Figures 2.4 and 2.9) have the top section of the head shaft installed inside the tubular hollow driver shaft. The coupling of the head shaft to driver is arranged on top of the motor and has a provision for axial lineshaft adjustment. Standard dimensions for the coupling are shown in Figure 2.11.

The solid shaft driver (see Figures 2.6, 2.7 and 2.8) is coupled to the lineshaft by an axially adjustable rigid coupling. The coupling is installed below the driver on the extended driver shaft.

2.1.3.3 Open/enclosed impeller

A typical semi-open impeller (see Figures 2.4 and 2.10) has a back shroud, with integral impeller vanes, but the vanes are open to the front (no front shroud). The leakage control is adjustable between the impeller vane and seat. This is achieved by positioning the impeller shaft axially for close impeller vane-to-bowl seat clearance.

The enclosed impeller (see Figures 2.4 and 2.10) has both a back shroud and a front shroud. Leakage control is limited.

2.1.3.4 Open/enclosed lineshaft

With open lineshaft pumps (see Figure 2.4), the pump shafting is exposed to the pumped liquid, which also cools and lubricates the lineshaft bearings.

Enclosed lineshaft pumps (see Figure 2.4) have the lineshaft protected from the pumped liquid by the shaft enclosing tube. The lineshaft bearings may be lubricated by fresh water, oil, or some other liquid injected into the enclosing tube at the ground or floor level.

2.1.4 Classification by impeller design

2.1.4.1 Specific speed

Specific speed is a number usually expressed as:

$$NS = \frac{(n)(Q)^{.5}}{(H_{ba})^{.75}}$$

Where:

NS = Pump specific speed;

n = Pump speed in revolutions per minute;

Q = Flow at best efficiency in gallons per minute (cubic meters per hour);

H_{ba} = Bowl assembly head per stage in feet (meters) (full diameter impeller).

The specific speed of an impeller is defined as the revolutions per minute at which a geometrically similar impeller would run if it were of such a size as to discharge one gallon per minute against one foot head (1 m^3/hr versus 1 m head).

Specific speed is indicative of the shape and characteristics of an impeller (see Figure 2.2). Specific speed is useful to the designer in establishing design parameters.

Impeller form and proportions vary with specific speed, as shown in Figure 2.3. It can be seen that there is a gradual change in the profiles from radial to axial flow configuration.

2.1.4.2 Radial flow

Pumps with this type of impeller have very low specific speeds [up to approximately 1,000 (1,160)]. The liquid enters the eye of the impeller and is turned by the impeller vanes and shroud to exit perpendicular to the axis of the pump shaft.

2.1.4.3 Modified radial flow

This type of pump usually has specific speed ranging from around 1,000 to 4,000 (1,160 to 4,640). The impellers are normally single suction. In pumps of this type, the liquid enters the impeller at the eye and exits semi-radially, at about a 60° to 70° angle with shaft axis (see Figure 2.8).

2.1.4.4 Mixed flow

This type of pump has a single inlet impeller with the flow entering axially and discharging about 45° with shaft axis, to the periphery. Pumps of this type usually have a specific speed from 4,000 to 9,000 (4,650 to 10,000) (see Figure 2.8).

2.1.4.5 Axial flow

A pump of this type, also called a propeller pump, has a single inlet impeller with the flow entering axially and discharging nearly axially. Pumps of this type usually have a specific speed above 9,000 (10,000). The axial flow pump propeller does not have a shroud (see Figure 2.9).

2.1.5 General information

2.1.5.1 Duplicate performance pump

A duplicate pump is one in which the performance characteristics are the same as another, within the variations permitted by these standards, and parts are of the same type; but by reason of improved design and/or materials, mounting dimensions and parts are not necessarily interchangeable.

2.1.5.2 Dimensionally interchangeable pump

An interchangeable pump is one in which the mounting dimensions are such that the replacement pump can be mounted on the existing foundation and match existing piping and driver, with hydraulic characteristics and materials to be specified. Interchangeability may involve some variation, not necessarily significant, as a result of manufacturing tolerances.

2.1.5.3 Identical pump

An identical pump is a duplicate of, and in addition is interchangeable with, a specific pump. Where it is intended that a pump is to be identical in all respects including parts, mountings, connecting flange dimensions, and materials, it should be identified as identical with pump number ______, not duplicate. An "identical pump" will duplicate the original pump as closely as manufacturing tolerances allow.

2.1.5.4 Rotation

The normal pump shaft rotation is counterclockwise (CCW) as viewed from the driver end of the pump. Left-hand threaded lineshaft joints will tighten when driven by a CCW driver.

Optional pump shaft rotation is clockwise (CW) as viewed from the driver end of the pump. Right-hand threaded joints will tighten when driven by a CW driver.

2.1.6 Construction

The cross-sectional drawings (Figures 2.4 through 2.10) illustrate commonly used parts in their proper relationship and a few typical construction modifications but do not necessarily represent recommended design.

The figure numbers shown in Figure 2.1 are for convenient cross-reference between tabulated names of parts and cross-sectional representation of standard part numbers in use by any manufacturer.

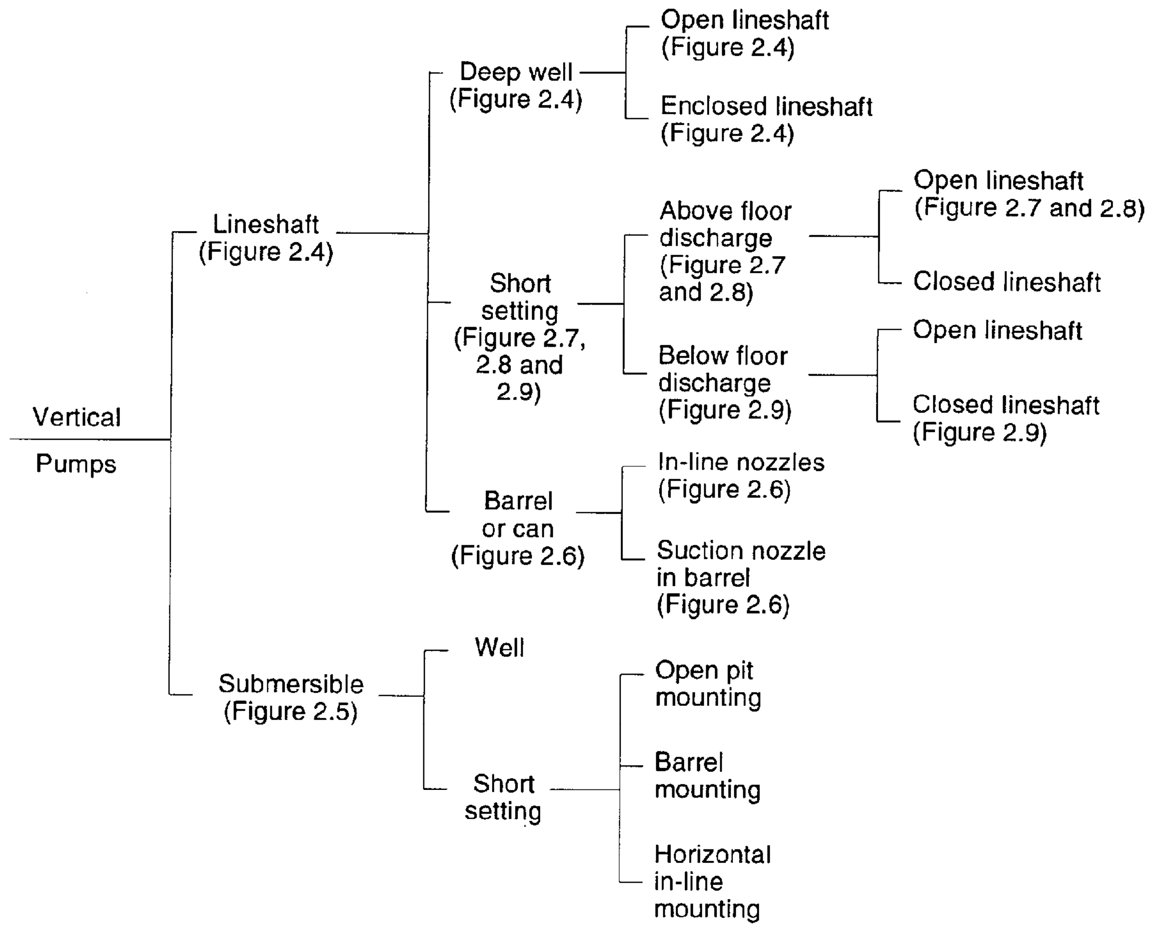


Figure 2.1 — Vertical pump types – single and multistage

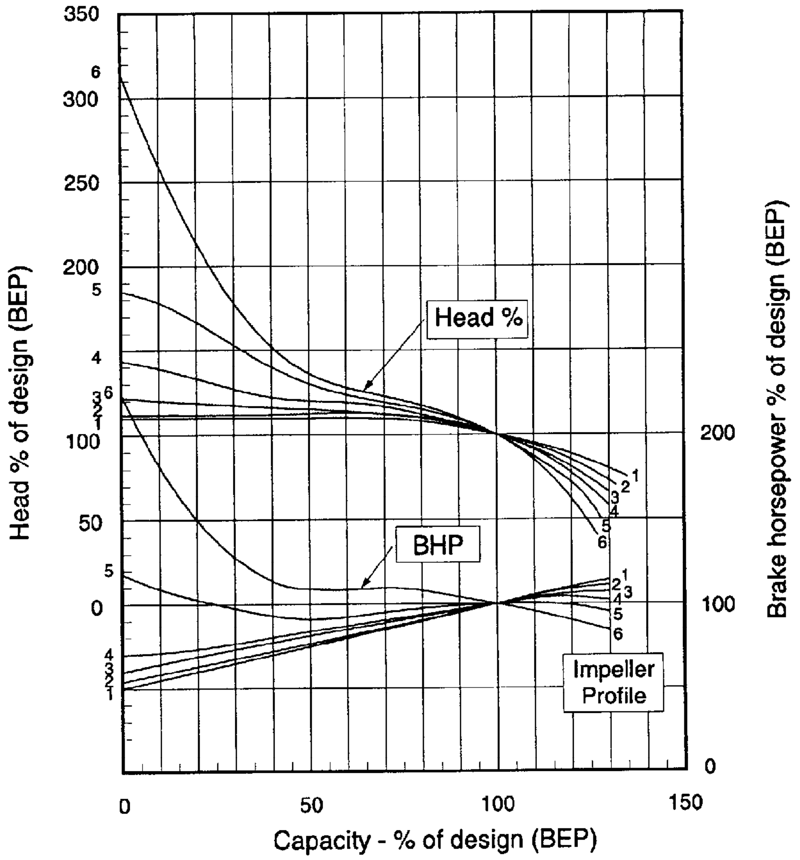


Figure 2.2 — Capacity – % of design

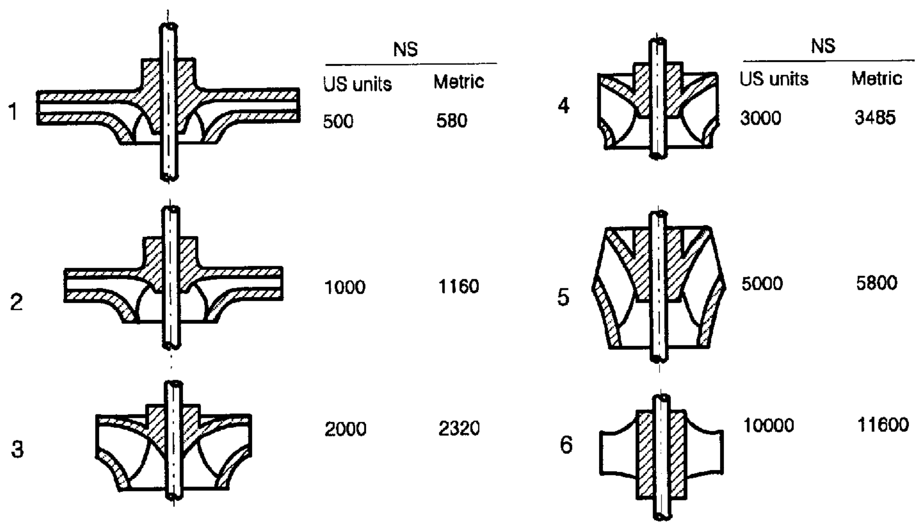
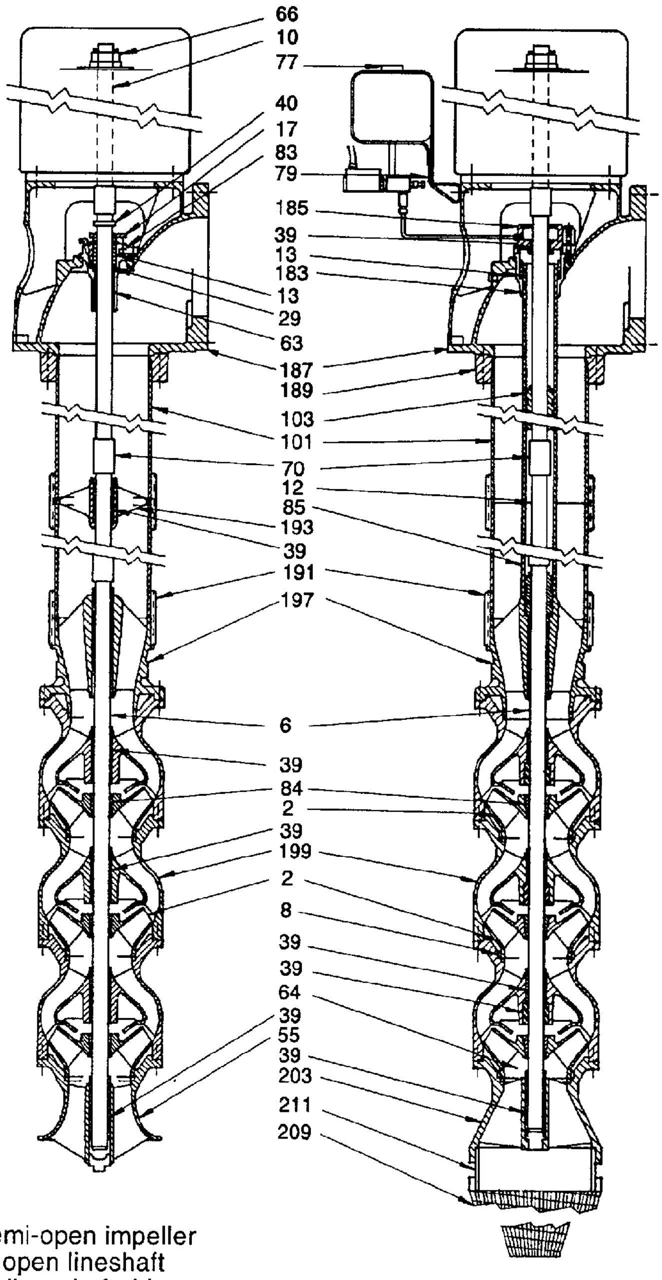


Figure 2.3 — Comparison of impeller profiles for various specific speed designs

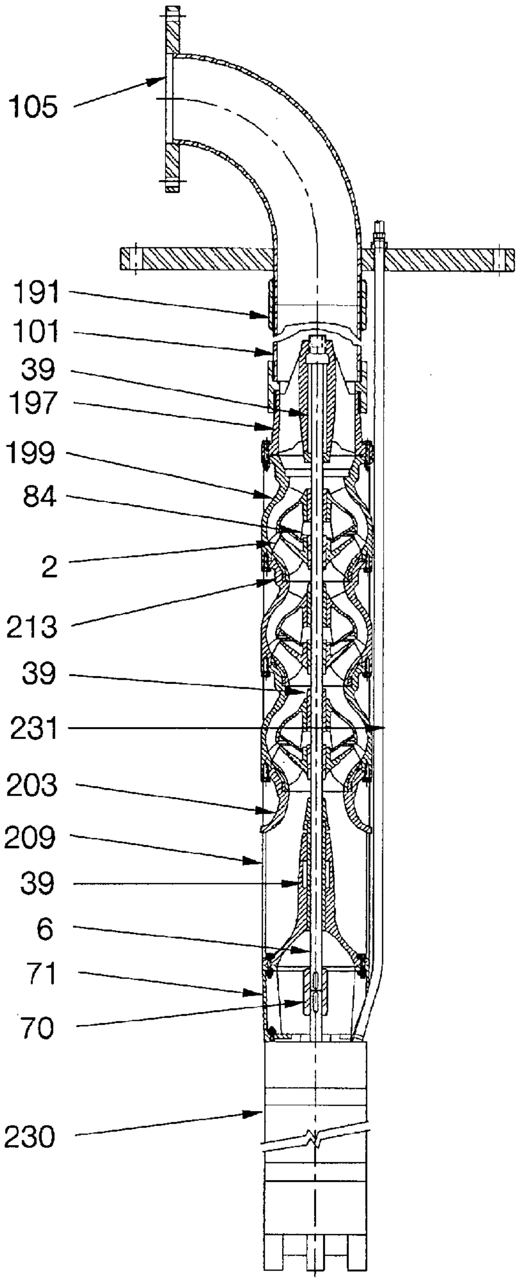
2 Impeller
6 Shaft, pump
8 Ring, impeller
10 Shaft, head
12 Shaft, line
13 Packing
17 Gland
29 Ring, lantern
39 Bushing, bearing
40 Deflector
55 Bell, suction
63 Bushing, stuffing-box
64 Collar, protecting
66 Nut, shaft-adjusting
70 Coupling, shaft
77 Lubricator
79 Bracket, lubricator
83 Stuffing-box
84 Collet, impeller lock
85 Tube, shaft-enclosing
101 Pipe, column
103 Bearing, lineshaft, enclosing
183 Nut, tubing
185 Plate, tension, tube
187 Head, surface discharge
189 Flange, top column
191 Coupling, column pipe
193 Retainer, bearing, open line shaft
197 Case, discharge
199 Bowl, intermediate
203 Case, suction
209 Strainer (optional)
211 Pipe, suction



Semi-open impeller
open lineshaft
hollow shaft driver

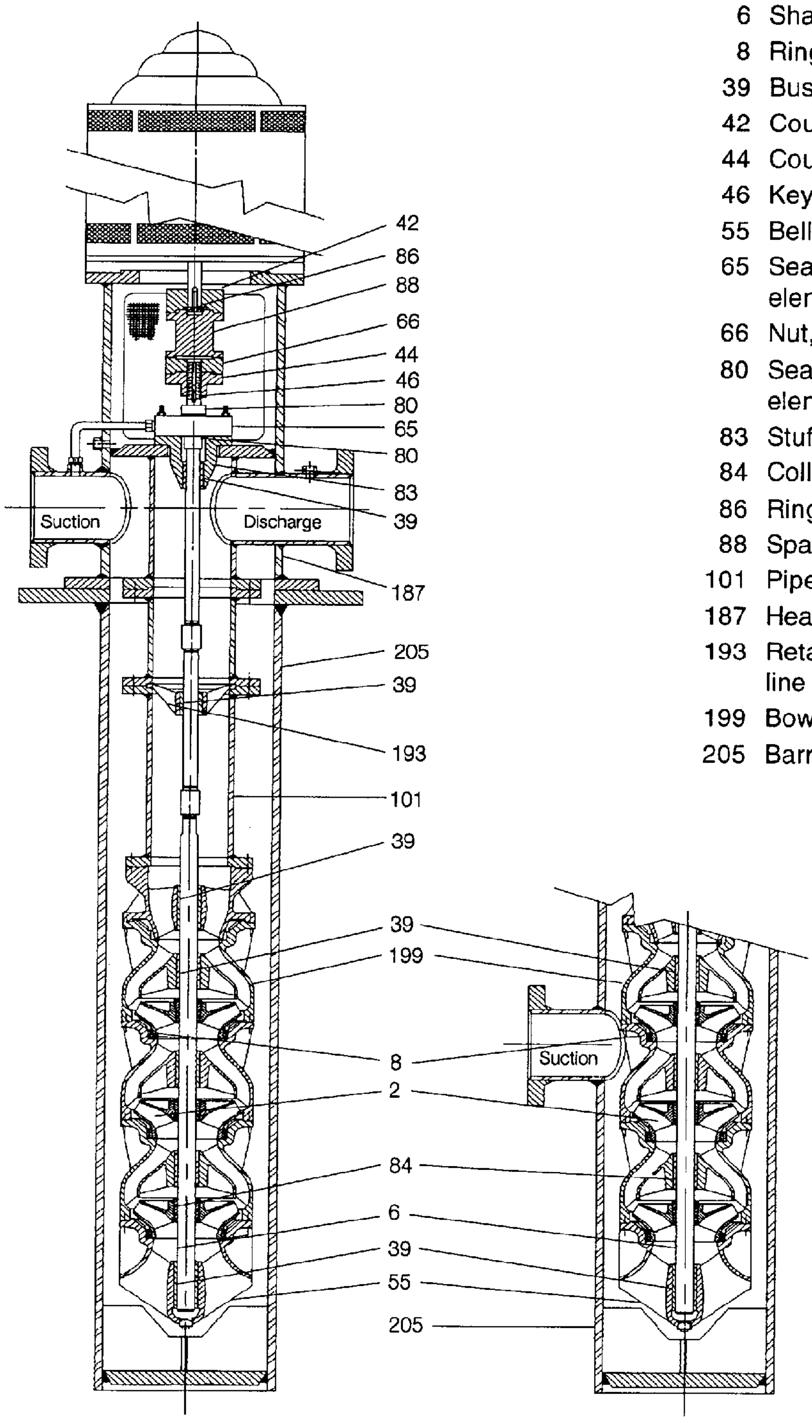
Enclosed impeller
enclosed lineshaft
hollow shaft driver

Figure 2.4 — Deep well pumps



2 Impeller
6 Shaft, pump
39 Bushing, bearing
70 Coupling, shaft
71 Adapter
84 Collet, impeller lock
101 Pipe, column
105 Elbow, discharge
191 Coupling, column pipe
197 Case, discharge
199 Bowl, intermediate
203 Case, suction
209 Strainer
213 Ring, bowl
230 Motor, submersible
231 Electric cable, submersible

Figure 2.5 — Vertical, multistage, submersible pump

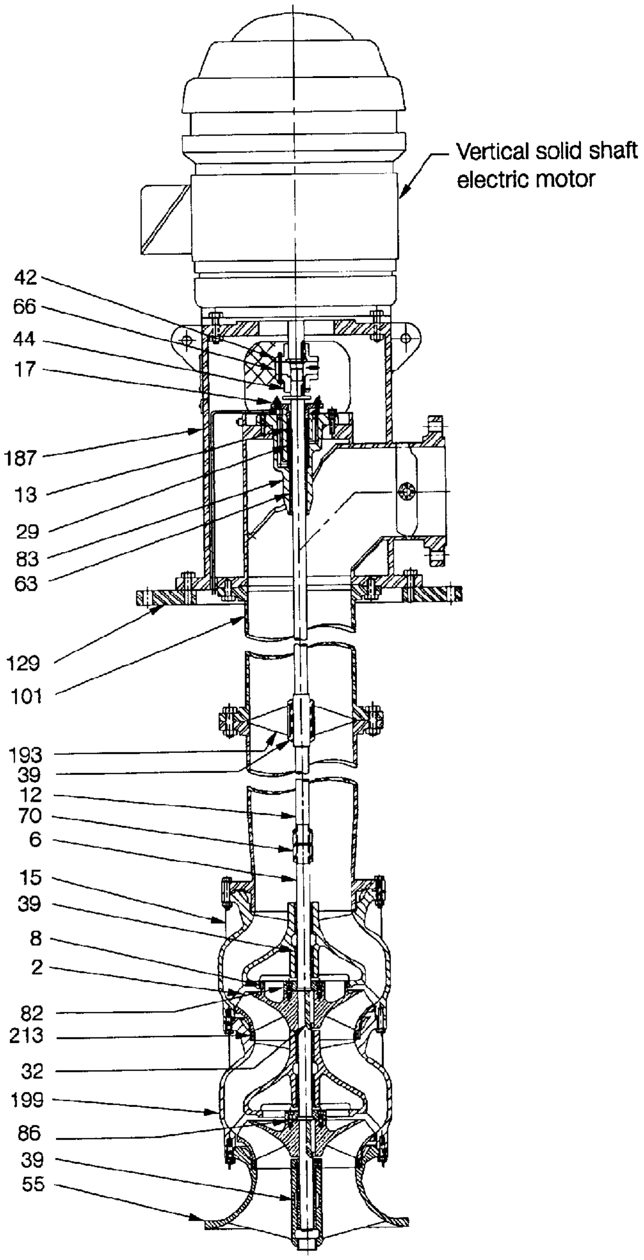


Suction nozzle in discharge head

Suction nozzle in barrel

2 Impeller
6 Shaft, pump
8 Ring, impeller
39 Bushing, bearing
42 Coupling half, driver
44 Coupling half, pump
46 Key, coupling
55 Bell, suction
65 Seal, mechanical, stationary element
66 Nut, shaft, adjusting
80 Seal, mechanical, rotating element
83 Stuffing-box
84 Collet, impeller lock
86 Ring, thrust, split
88 Spacer, coupling
101 Pipe, column
187 Head, surface discharge
193 Retainer, bearing, open line shaft
199 Bowl, intermediate
205 Barrel or can suction

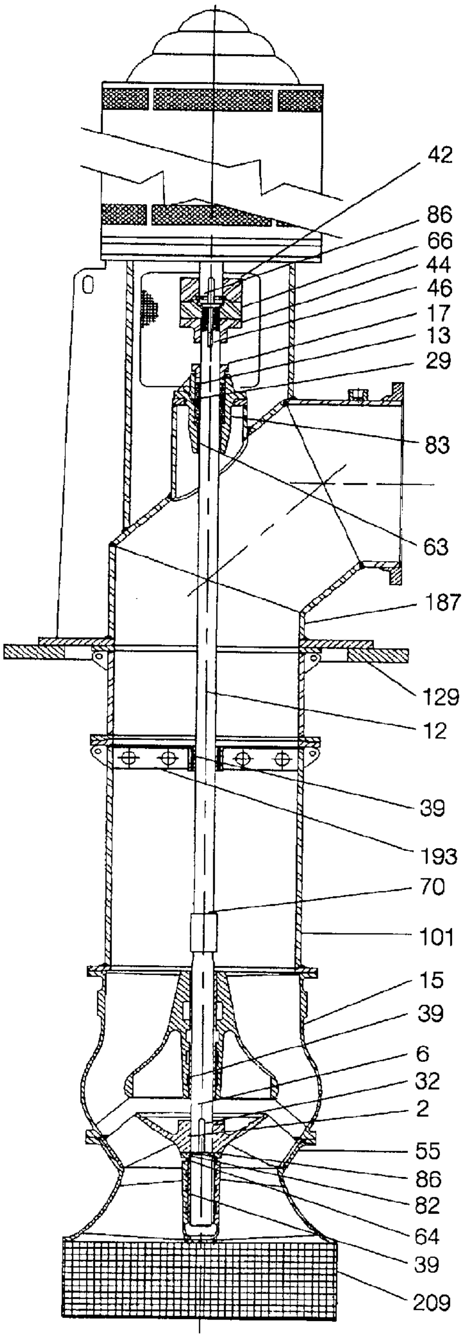
Figure 2.6 — Vertical, single or multistage barrel or can pump



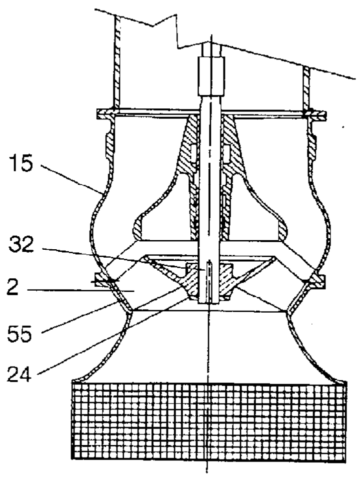
2 Impeller
6 Shaft, pump
8 Ring, impeller
12 Shaft, line
13 Packing
15 Bowl, discharge
17 Gland
29 Ring, lantern
32 Key, impeller
39 Bushing, bearing
42 Coupling half, driver
44 Coupling half, pump
55 Bell, suction
63 Bushing, stuffing-box
66 Nut, shaft adjusting
70 Coupling, shaft
82 Ring, thrust, retainer
83 Stuffing-box
86 Ring, thrust, split
101 Pipe, column
129 Sole plate
187 Head, surface discharge
193 Retainer, bearing, open line shaft
199 Bowl, intermediate
213 Ring, bowl

Figure 2.7 — Vertical single or multistage, short setting, open line shaft

2 Impeller
6 Shaft, pump
12 Shaft, line
13 Packing
15 Bowl, discharge
17 Gland
24 Nut, impeller
29 Ring, lantern
32 Key, impeller
39 Bushing, bearing
42 Coupling half, driver
44 Coupling half, pump
46 Key, coupling
55 Bell, suction
63 Bushing, stuffing-box
64 Collar, protecting
66 Nut, shaft, adjusting
70 Coupling, shaft
82 Ring, thrust, retainer
83 Stuffing-box
86 Ring, thrust, split
101 Pipe, column
129 Sole plate
187 Head, surface discharge
193 Retainer bearing open lineshaft
209 Strainer (optional)

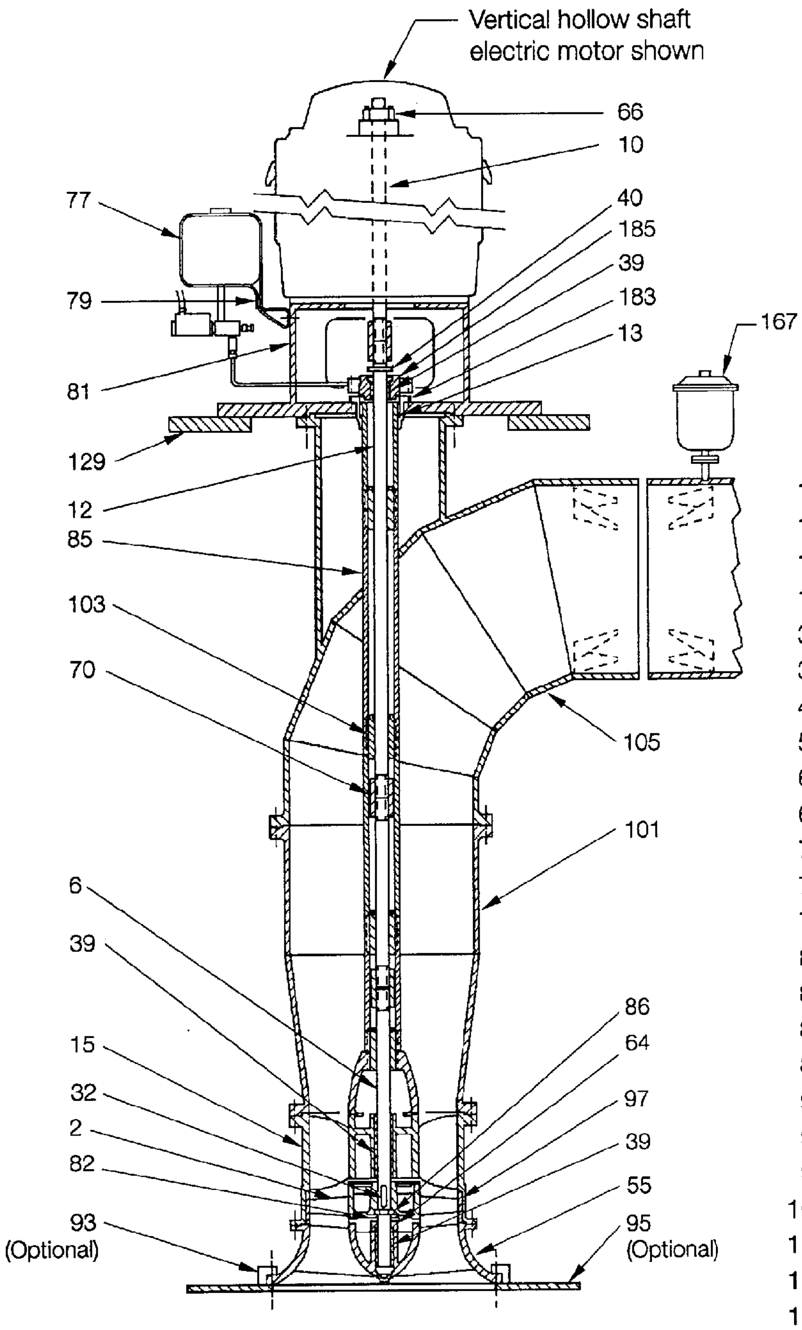


Bearing below impeller



Without bearing below impeller

Figure 2.8 — Mixed flow vertical – open line shaft



2 Impeller (propeller)
6 Shaft, pump
10 Shaft, head
12 Shaft, line
13 Packing
15 Bowl, discharge
32 Key, impeller
39 Bushing, bearing
40 Deflector
55 Bell, suction
64 Collar, protecting
66 Nut, shaft, adjusting
70 Coupling, shaft
77 Lubricator
79 Bracket, lubricator
81 Pedestal, driver
82 Ring, thrust, retainer
85 Tube, shaft enclosing
86 Ring, thrust, split
93 Clamp, umbrella
95 Umbrella, suction
97 Liner, bowl
101 Pipe, column
103 Bearing, lineshaft, enclosed
105 Elbow, discharge
129 Sole plate
165 Pipe, vacuum breaker
167 Valve, air & vacuum release
183 Nut, tubing
185 Plate, tension, tube

Figure 2.9 — Vertical, axial flow impeller (propeller) type (enclosed lineshaft) below floor discharge configuration

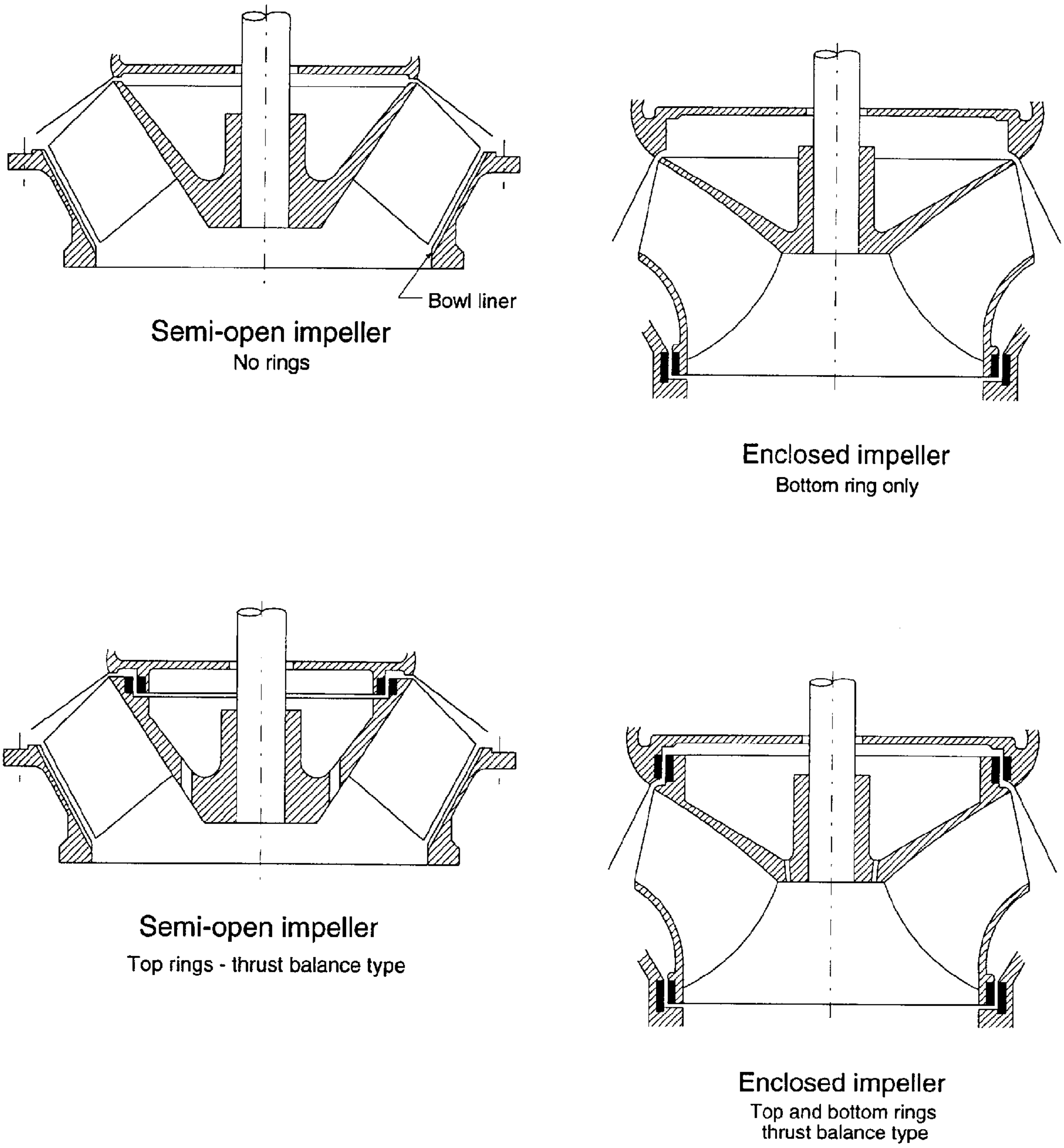
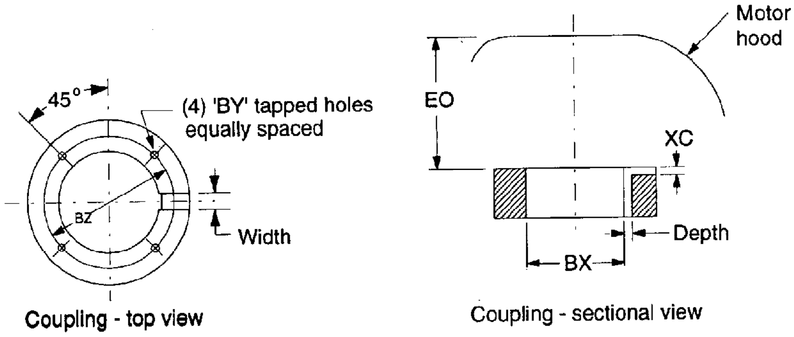


Figure 2.10 — Wear ring arrangements



Top drive coupling

Coupling dimensions (inches)				Keyway (inches)[3]		Hood clearance (inches)
Coupling bore BX[1]	BY	BZ	XC	Width	Depth	EO[2]
.751	10-32	1.375	.38	.187	.109	2.25
.876	10-32	1.375	.38	.187	.109	2.63
1.001	10-32	1.375	.43	.250	.140	3.00
1.188	.250-20	1.750	.43	.250	.140	3.50
1.251	.250-20	1.750	.43	.250	.140	3.75
1.251	.250-20	1.750	.56	.375	.203	3.75
1.438	.250-20	2.125	.56	.375	.203	4.30
1.501	.250-20	2.125	.56	.375	.203	4.50
1.688	.250-20	2.500	.56	.375	.203	5.00
1.751	.250-20	2.500	.56	.375	.203	5.25
1.938	.250-20	2.500	.68	.500	.265	5.80
2.001	.250-20	2.500	.68	.500	.265	6.00
2.188	.375-16	3.250	.68	.500	.265	6.50
2.251	.375-16	3.250	.68	.500	.265	6.75
2.438	.375-16	3.250	.81	.625	.327	7.30
2.501	.375-16	3.250	.81	.625	.327	7.50
2.688	.375-16	3.750	.81	.625	.327	8.00
2.751	.375-16	3.750	.81	.625	.327	8.25
2.938	.375-16	4.250	.94	.750	.390	10.00
3.188	.375-16	4.250	.94	.750	.390	10.00
3.438	.375-16	4.500	1.06	.875	.453	10.00
3.688	.375-16	5.000	1.06	.875	.453	10.00
3.938	.375-16	5.000	1.06	.875	.453	10.00

NOTES

1 Tolerances for the "BX" dimension are +0.001 inch, -0.000 inch, up to and including 1.5 inch diameter, and +0.002 inch, -0.000 inch for larger diameters.

2 The "EO" dimension, which is clearance from coupling top to inside of hood, is based upon a minimum dimension of 3 times the BX dimension for shaft diameters 2.75 and smaller and 10" for shaft diameters 2.94 thru 3.94.

3 American Standard, Gib-Head, Taper Stock and Square type keys fit the above dimensions.

Figure 2.11 — Vertical hollow shaft driver coupling dimensions

Table 2.1 — Alphabetical part name listing

Part Name	Item #	Abbreviation	Definition
Adapter	71	Adpt	A machined piece used to permit assembly of two other parts or for a spacer
Adapter, tube	195	Adpt tube	A cyclindrical piece used to connect discharge case to enclosing tube
Barrel or can, suction	205	Bl/can suc	A receptacle for conveying the liquid to the pump
Base plate	23	Base Pl	A metal member on which the pump and its driver are mounted
Bearing, inboard	16	Brg inbd	The bearing nearest the coupling
Bearing, lineshaft enclosed	103	Brg linesht encl	A bearing which also serves to couple portions of the shaft enclosing tube
Bearing, outboard	18	Brg outbd	The bearing most distant from the coupling
Bearing, sleeve	39	Brg slv	A replaceable, cylindrical bearing secured within a stationary member
Bell, suction	55	Bel suct	A flared tubular section for directing the flow of liquid into the pump
Bowl, discharge	15	Bowl disch	A diffuser of an axial flow or mixed flow or turbine pump
Bowl, intermediate	199	Bowl intmd	An enclosure within which the impeller rotates and which serves as a guide for the flow from one impeller to the next
Bracket, lubricator	79	Bkt lubr	A means of attaching the lubricator to the pumping unit
Bushing, stuffing-box	63	Bush stfg box	A replaceable sleeve or ring placed in the end of the stuffing-box opposite the gland
Case, discharge	197	Case disch	A guide for liquid flow from bowl to pump column
Case, suction	203	Case suct	A device used to receive the liquid and guide it to the first impeller

(continued)

Table 2.1 — Alphabetical part name listing *(continued)*

Part Name	Item #	Abbreviation	Definition
Clamp, umbrella	93	Clp umbla	A fastening used to attach the suction umbrella to suction bowl
Collar, protecting	64	Clr protg	A rotating member for preventing the entrance of contaminating material to bearings of vertical pumps
Collar, shaft	68	Clr sft	A ring used on a shaft to establish a shoulder for a ball bearing
Collet, impeller lock	84	Cllt imp lock	A tapered collar used to secure the impeller to the pump shaft
Coupling, column pipe	191	Cplg col pipe	A threaded sleeve used to couple sections of column pipe
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 shaft	70	Cplg sft	A mechanism used to transmit power from the lineshaft to the pump shaft or to connect two pieces of shaft
Cover, bearing, inboard	35	Cov brg inbd	An enclosing plate for either end of an inboard bearing
Cover, bearing, outboard	37	Cov brg outbd	An enclosing plate for either end of the outboard bearing
Deflector	40	Defl	A flange or collar around a shaft and rotating with it to prevent passage of liquid, grease, oil or heat along the shaft
Elbow	57	Ell	A curved water passage, usually 90 degrees, attached to the pump inlet or discharge
Elbow, discharge	105	Ell disch	An elbow in an axial flow, mixed flow, or turbine pump by which the liquid leaves the pump
Electrical cable, subm	231	El cab subm	

(continued)

Table 2.1 — Alphabetical part name listing *(continued)*

Part Name	Item #	Abbreviation	Definition
Fitting, discharge	161	Ftg disch	A body to which may be assembled various fire pump fittings such as relief valve, hose valve, manifold, etc.
Flange, top column	189	Flg top col	A device used to couple column to discharge head
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 of proper shape and characteristics for use in joints between parts to prevent leakage
Gland	17	Gld	A follower which compresses packing in a stuffing-box or retains the stationary element of a mechanical seal
Head, surface discharge	187	Hd surf disch	A support for driver, pump column and a means by which the liquid leaves the pump
Housing, bearing	99	Hsg brg	A body in which the bearing is mounted
Impeller (propeller)	2	Imp	The bladed member of the rotating assembly of the pump which imparts the principal force to the liquid pumped. Called a "Propeller" for axial flow
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, bowl	97	Lnr bowl	A replaceable cylindrical piece mounted on the discharge bowl and within which the propeller rotates
Locknut, bearing	22	Lknut brg	A fastener which locks a ball bearing on the shaft

(continued)

Table 2.1 — Alphabetical part name listing *(continued)*

Part Name	Item #	Abbreviation	Definition
Lockwasher	69	Lkwash	A device to prevent loosening of a nut
Lubricator	77	Lubr	A device for applying a lubricant to the point of use
Motor, submersible	230	Mot subm	An electrical motor for submerged-in-liquid operation
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, tube	183	Nut tube	A device for sealing and locking shaft enclosing tube
Packing	13	Pkg	A pliable lubricated material used to provide a seal around that portion of the shaft located in the stuffing-box
Pedestal, driver	81	Ped drvr	A metal support for the driver of a vertical pump
Pipe, column	101	Pipe col	A vertical pipe by which the pumping element is suspended
Pipe, suction	211	Pipe suct	A device for conveying the liquid to the pump's suction
Plate, tension, tube	185	Pl tens tube	A device for maintaining tension on shaft-enclosing tube
Seal, mechanical, rotating element	80	Seal mech rot elem	A device flexibly mounted on the shaft in or on the stuffing-box and having a smooth, flat seal face held against the stationary sealing face
Seal, mechanical, stationary element	65	Seal mech sta elem	A sub assembly consisting of one or more parts mounted in or on a stuffing-box and having a smooth flat sealing face
Shaft, head	10	Sft hd	The upper shaft in a vertical pump which transmits power from the driver to the drive shaft

(continued)

Table 2.1 — Alphabetical part name listing *(continued)*

Part Name	Item #	Abbreviation	Definition
Shaft, line	12	Sft ln	The shaft which transmits power from the head shaft or driver to the pump shaft
Shaft, pump	6	Sft pump	The shaft on which the impeller is mounted and through which power is transmitted to the impeller
Sole plate	129	Sole pl	A metallic pad, usually imbedded in concrete, on which the pump base is mounted
Spacer, coupling	88	Spcr cplg	A cylindrical piece used to provide axial space for the removal of the mechanical seal without removing the driver
Strainer	209	Str	A device used to prevent large objects from entering the pump
Stuffing-box	83	Stfg box	A portion of the casing through which the shaft extends and in which packing and a gland or a mechanical seal is placed to prevent leakage
Tube, shaft-enclosing	85	Tube sft encl	A cylinder used to protect the drive shaft and to provide a means for mounting bearings
Umbrella, suction	95	Umbla suct	A formed piece attached to the suction bowl to reduce disturbance at pump inlet and reduce submergence required
Valve, air and vacuum relief	167	Val air vac rel	A means of releasing air during start-up and releasing vacuum during shutdown

2.2 Definitions, terminology and symbols

The purpose of this section is to define terms used in pump applications. Symbols, terms and units are shown in Table 2.2 and subscripts in Table 2.3.

2.2.1 Capacity (Q)

The capacity of a pump is the total volume throughput per unit of time at suction conditions. It includes both liquid and any dissolved or entrained gases at the stated operating conditions.

2.2.2 Speed (n)

The number of revolutions of the shaft in a given unit of time. Speed is expressed as revolutions per minute.

2.2.3 Head (h)

Head is the expression of the energy content of the liquid referred to any arbitrary datum. It is expressed in units of energy per unit weight of liquid. The measuring unit for head is feet (meter) of liquid.

2.2.3.1 Gauge head (h_g)

The energy of the liquid due to its pressure above atmospheric as determined by a pressure gauge or other pressure measuring device.

$$\text{(US units) } h_g = \frac{2.31p}{s}$$

2.2.3.2 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 = dividing the flow by the cross-section area at the point of gauge connection.

2.2.3.3 Elevation head (Z)

The potential energy of the liquid due to its elevation relative to datum level measured to the center of the pressure gauge or liquid level.

2.2.3.4 Datum

The pump's datum is a horizontal plane which serves as the reference for head measurements taken during test. Vertical pumps are usually tested in an open pit with the suction flooded. The datum is then the eye of the first stage impeller (see Figure 2.12).

Optional tests can be performed with the pump mounted in a suction can. Irrespective of pump mounting, the pump's datum is maintained at the eye of the first stage impeller.

2.2.3.5 Total suction head (h_s), open suction

For open suction (wet pit) installations, the first stage impeller of the bowl assembly is submerged in a pit. The total suction head (h_s) at datum is the submergence in feet of water (Z_w). The average

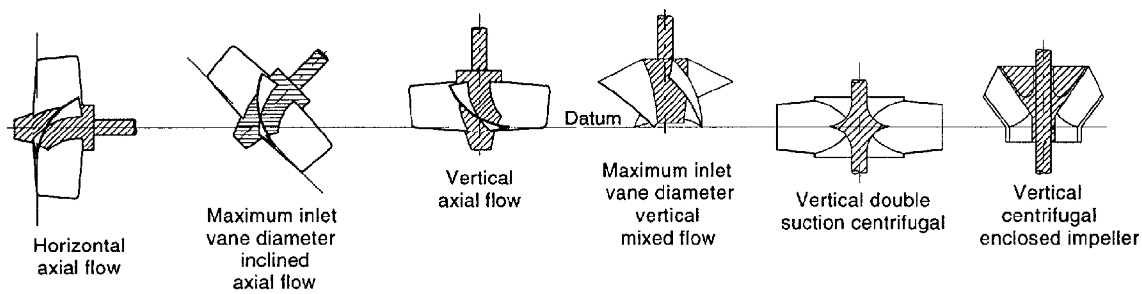


Figure 2.12 — Datum elevations for various pump designs

Table 2.2 – Symbols and terminology

Symbol	Term	US Customary Unit	Abbreviation	Metric unit	Abbreviation	Conversion factor [1]
A	Area	square inches	in^2	square millimeter	mm^2	645.2
β (beta)	Meter or orifice ratio	dimensionless	—	dimensionless	—	1
D	Diameter	inches	in	millimeter	mm	25.4
Δ (delta)	Difference	dimensionless	—	dimensionless	—	1
η (eta)	Efficiency	percent	%	percent	%	1
g	Gravitational acceleration	feet/second/second	ft/sec^2	meter/second/second	m/s^2	0.3048
γ (gamma)	Specific weight	pounds/cubic foot	lb/ft^3	kiloNewton/cubic meter	kN/m^3	0.1571
h	Head	feet	ft	meter	m	0.3048
H	Total head	feet	ft	meter	m	0.3048
l	Static lift	feet	ft	meter	m	0.3048
n	Speed	revolutions/minute	rpm	revolutions/minute	rpm	1
NPSHA	Net positive suction head available	feet	ft	meter	m	0.3048
NPSHR	Net positive suction head required	feet	ft	meter	m	0.3048
NS	Specific speed $NS = nQ^{1/2}/H^{3/4}$	dimensionless	—	dimensionless	—	1.162
ν (nu)	Kinematic viscosity	feet squared/second	ft^2/sec	millimeter squared/sec	mm^2/s	92903
π	pi = 3.1416	dimensionless	—	dimensionless	—	1
p	Pressure	pounds/square inch	psi	kilopascal	kPa	6.895
P	Power	horsepower	hp	kilowatt	kW	0.7457
q	Capacity	cubic feet/second	ft^3/sec	cubic meter/hour	m^3/h	101.94
Q	Capacity	US gallons/minute	gpm	cubic meter/hour	m^3/h	0.2271
RM	Linear model ratio	dimensionless	—	dimensionless	—	—
ρ (rho)	Density	pound mass/cubic foot	lbm/ft^3	kilogram/cubic meter	kg/m^3	16.02
S	Suction specific speed	dimensionless	—	dimensionless	—	1.162
s	Specific gravity	dimensionless	—	dimensionless	—	1
t	Temperature	degrees Fahrenheit	°F	degrees Celcius	°C	(°F–32) x 5⁄9
τ (tau)	Torque	pound-feet	lb-ft	Newton – meter	N•m	1.356
v	Velocity	feet/second	ft/sec	meter/second	m/s	0.3048
x	Exponent	none	none	none	none	1
Z	Elevation gauge distance above or below datum	feet	ft	meter	m	0.3048

[1] Conversion factor x US units = metric units.

Table 2.3 – Subscripts

Subscript	Term	Subscript	Term
1	Test condition or model	min	Minimum
2	Specific condition or prototype	mot	Motor
a	Absolute	ot	Operating temperature
atm	Atmospheric	OA	Overall unit
b	Barometric	p	Pump
ba	Bowl assembly	s	Suction
d	Discharge	t	Theoretical
dvr	Driver	v	Velocity
g	Gauge	vp	Vapor pressure
im	Intermediate mechanism	w	Water
max	Maximum		

velocity head of the flow in the pit is small enough to be neglected:

$$h_s = Z_w$$

Where:

Zw = Vertical distance in feet (meters) from free water surface to datum.

2.2.3.6 Total suction head (h_s), closed suction test

For closed suction installations, the pump suction nozzle may be located either above or below grade level.

The total suction head (h_s), referred 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 (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 an amount exceeding the sum of the elevation head and the velocity head.

2.2.3.7 Pump total discharge head (h_d)

The total discharge head (h_d) is the sum of the discharge gauge head (h_{gd}) measured after the discharge elbow plus the velocity head (h_{vd}) at the point of gauge attachment plus the elevation (Z_d) from the discharge gauge centerline to the pump datum.

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

2.2.3.8 Pump total head (H)

This is the measure of energy increase per unit weight of the liquid, imparted to the liquid by the pump, and is the difference between the total discharge head and the total suction head.

This is the head normally specified for pumping applications, since the complete characteristics of a system determine the total head required.

2.2.3.9 Bowl assembly total head (H_{ba})

The bowl assembly head (H_{ba}) is the gauge head (h_{gd}) measured at a gauge connection located on the column pipe downstream from the bowl assembly, plus the velocity head (h_v) at point of gauge connection, plus the vertical distance (Z_d) from datum to the pressure gauge centerline, minus the submergence Z_w, which is the vertical distance from datum to the liquid level.

$$H_{ba} = h_{gd} + h_{vd} + Z_d - Z_w$$

2.2.3.10 Atmospheric head (h_{atm})

Local atmospheric pressure expressed in feet (meters).

2.2.3.11 Friction head (h_f)

Friction head is the hydraulic energy required to overcome frictional resistance of a piping system to liquid flow.

2.2.4 Condition points

2.2.4.1 Rated condition point

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

2.2.4.2 Specified condition point

Specified condition point is synonymous with rated condition point.

2.2.4.3 Normal condition point

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

2.2.4.4 Best efficiency point (BEP)

The capacity and head at which the pump efficiency is a maximum.

2.2.4.5 Shutoff

The condition of zero flow where no liquid is flowing through the pump.

2.2.4.6 Allowable operating range

This is the flow range at the specified speeds with the impeller supplied, as limited by cavitation, heating, vibration, noise, shaft deflection, fatigue and other similar criteria. This range is to be specified by the manufacturer.

2.2.5 Suction conditions

2.2.5.1 Submerged suction

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

2.2.5.2 Static suction lift (l_s)

Static suction lift is a hydraulic pressure below atmospheric at the intake port of the pump.

2.2.5.3 Net positive suction head available (NPSHA)

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

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

Where:

h_{sa} = Total suction head absolute = $h_{atm} + h_s$

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

In can pumps (see Figure 2.7), NPSHA is often determined at the suction flange. Since NPSHR is determined at the first stage impeller, the NPSHA value must be adjusted to the first stage impeller by adding the difference in elevation and subtracting the losses in the can (see Paragraph 2.3.2.15).

2.2.5.4 Net positive suction head required (NPSHR)

The amount of suction head, over vapor pressure, required to prevent more than 3% loss in total head to the first stage of the pump at a specific capacity.

2.2.5.5 Maximum suction pressure

This is the highest suction pressure to which the pump will be subjected during operation.

2.2.6 Power

2.2.6.1 Electric motor input power (P_{mot})

The electrical input power to the motor.

$$P_{mot} = \frac{kW}{0.746}$$

2.2.6.2 Pump input power (P_p)

The power needed to drive the complete pump assembly including bowl assembly input power, lineshaft power loss, stuffing-box loss and thrust bearing loss. With pumps having built-in thrust bearing, the power delivered to the pump shaft coupling is equal to the pump input power. With pumps that rely on the driver thrust bearing, the thrust bearing loss shall be added to the power delivered to the pump shaft. It is also called brake horsepower.

2.2.6.3 Bowl assembly input power (P_{ba})

The power delivered to the bowl assembly shaft.

2.2.6.4 Pump output power (P_w)

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

$$\text{(US units) } P_w = \frac{Q \times H \times s}{3960}$$

$$\text{(Metric) } P_w = \frac{Q \times H \times s}{366}$$

2.2.6.5 High-energy pump

Vertical turbine pumps with heads greater than 250 ft/stage and/or requiring more than 300 hp/stage.

2.2.6.6 Overall efficiency (η_{OA})

This is the ratio of the energy imparted to the liquid (P_W) by the pump to the energy supplied to the driver (P_{dvr}); that is, the ratio of the water horsepower to the power input to the primary driver expressed in percent.

$$\eta_{OA} = \frac{P_W}{P_{dvr}} \times 100$$

2.2.6.7 Pump efficiency (η_p)

The ratio of the pump output power (P_W) to the pump input power (P_p); that is, the ratio of the water horsepower to the brake horsepower expressed in percent.

$$\eta_p = \frac{P_W}{P_p} \times 100$$

2.2.6.8 Bowl assembly efficiency (η_{ba})

This is the efficiency obtained from the bowl assembly, excluding all hydraulic and mechanical losses within other pump components. This is the efficiency usually shown on published performance curves.

2.2.7 Pump pressures

2.2.7.1 Working pressure (p_d)

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

2.2.7.2 Maximum allowable casing working pressure

This is the highest pressure at the specified pumping temperature for which the pump casing is designed. This pressure shall be equal to or greater than the maximum discharge pressure. In the case of double casing can pumps, the maximum allowable casing working pressure on the suction side may be different from that on the discharge side.

2.2.7.3 Maximum discharge pressure

The highest discharge pressure to which the pump will be subjected during operation.

2.2.7.4 Field test pressure

The maximum static test pressure to be used for leak testing a closed pumping system in the field if the pumps are not isolated. Generally this is taken as 125% of the maximum allowable casing working pressure. Where mechanical seals are used, this pressure may be limited by the pressure-containing capabilities of the seal.

NOTE – See Paragraph 2.2.7.2 Maximum allowable casing working pressure. Consideration of which may limit the field test pressure of the pump to 125% of the maximum allowable casing working pressure on the suction side of double casing can type pumps and certain other pump types.

2.2.8 Impeller balancing

2.2.8.1 Single plane balancing (also called static balancing)

Correction of residual unbalance to a specified maximum limit by removing or adding weight in one correction plane only. Can be accomplished statically using balance rails or by spinning.

2.2.8.2 Two plane balancing (also called dynamic balancing)

Correction of residual unbalance to a specified limit by removing or adding weight in two correction planes. Accomplished by spinning on appropriate balancing machines.

2.3 Design and application

The purpose of this section is to provide a guide for the application of vertical pumps for various services. No attempt has been made to cover all phases of vertical pump application, but an endeavor has been made to point out some of the principal features of this type of pump and the precautions which should be taken in their use. In general, there are several advantages to vertical pumps.

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

- *First:* Pumps can be designed for any desired value in the complete range of specific speeds;
- *Second:* Multistage pumps have characteristically steep head - capacity curves. The steep curve features less flow change when head conditions change;
- *Third:* The maximum pump horsepower usually coincides with the recommended operating range. Generally for pumps with specific speeds of 5000 or less, changes in operating conditions do not cause driver overload;
- *Fourth:* It is easy to change the staging on the pump; that is, adding or subtracting to existing equipment or changing impellers in the pump, if it becomes necessary to change the hydraulic characteristics of the system;
- *Fifth:* The length of the pump column can be easily selected so that the NPSH available exceeds the NPSH required at all times;
- *Sixth:* The pumping element is normally submerged, which eliminates the need for priming devices, enhancing unattended reliable service;
- *Seventh:* Minimum floor space is required.

In summary, the advantages to the system designer in particular is that using a vertical pump provides flexible hydraulic characteristics and dimensional adaptability.

Vertical pumps often make optimum solutions to pumping problems possible.

2.3.1 Typical applications

2.3.1.1 Booster service (open suction, closed suction, can pumps)

Vertical pumps in booster service are of two basic types:

1) Open suction, wet pit which takes suction from a free surface source and discharges into an enclosed pipe (see Figure 2.7).

2) Closed suction with the integral suction chamber. Commonly called can or barrel pumps (see Figure 2.6).

Wet pit type pumps are commonly used where liquids are contained in a sump open to the atmosphere. They are also used in settling basins where sand and solids settle out before entering the pump's suction.

Vertical can pumps are particularly effective for booster service in surface piping systems due to the in-line arrangement of suction and discharge connections. The drive, normally a vertical induction motor, is mounted directly on top of the pump. Either a vertical hollow shaft or solid shaft motor may be used, with solid shafts preferred for this type of application. Horizontal drivers such as engines or steam turbines may be used in conjunction with a right angle gear drive to transmit torque to the pump.

Various materials of construction are available to handle the liquids pumped. It is important that careful consideration be exercised by the specifying engineer and user to select materials of construction which are compatible with the corrosive and erosive qualities of the liquid pumped.

For a successful installation special emphasis should be placed on proper sump and can design, along with appropriate materials of construction. Additional information on sump design may be found in the intake design section of this standard.

Vertical can or barrel pumps come in a variety of sizes and shapes. Many of the designs are conceived, either based on very general knowledge by the architect or contractor, or simply to fit into a piping arrangement without properly considering flow patterns at the can inlet or in the can itself.

It is difficult to generalize on correct or satisfactory designs because there are many interrelated factors to be considered (see Figure 2.13). Some of these factors are inlet velocities at (A), velocity patterns at (C), the configuration at (D), location

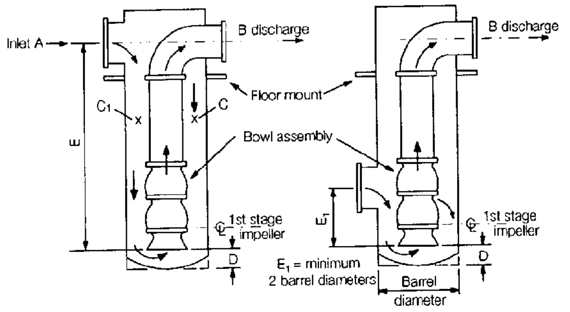


Figure 2.13 — Vertical can or barrel pumps

of can inlet with respect to pump suction bell (E) and pump energy level and liquid being pumped, as shown in Figure 2.13.

In order to avoid cavitation, submerged vortexing, surges or other flow instabilities, it is recommended that reference be made to the pump manufacturer's published standard dimensions, where possible (see Figures 2.26 through 2.33). Additional information may be found in the intake design section of this standard.

Particular caution is needed when oversizing the barrel diametrically for future system flow increases. Flow stabilizers attached to the barrel or pumping assembly may be recommended. Also, extended barrel length for future addition of stages must be carefully reviewed for hydraulic performance.

2.3.1.2 Process service (chemical, petrochemical)

The vertical process service pump usually has an integral suction. Lubrication of the pump bearings is usually by the product pumped, thus no external lubrication is necessary. The pumped liquid should not contain extensive amounts of solids or abrasive particles. Vertical in-line pumps are not covered here.

The Vertical Chemical Process Pump is a pump designed for pumping corrosive liquids. The materials of construction for the parts in contact with the liquid, including stuffing-boxes or seals, must be selected to offer maximum resistance to corrosion at the pumping temperature with due consideration to economy. Each application must be carefully scrutinized to determine the severity of corrosion or abrasion, the viscosities at extreme pumping temperatures, the hazard involved with the liquids to be pumped, changes in the composition of the liquid, vapor pressure, NPSH, prolonged operation at or near shut-off, or any other pertinent characteristic of the liquid or application.

The physical and chemical properties of materials and the available forms and methods of fabrication must be considered in the design of satisfactory equipment.

Special seals and/or stuffing-boxes may be required. The unit should be designed for easy and quick disassembly for inspection, cleaning, or repair. Since the pumped liquid usually is the bearing lubricant, special consideration is required for the compatibility of liquid to the design and materials of the stuffing-box, bearings, bowl and lineshaft.

The manufacturer's instruction for installation procedures should be carefully followed. Established schedules for periodic inspection and maintenance are essential.

The Vertical Petrochemical Process Pump usually refers to a unit pumping volatile hydrocarbons. The determination of the net positive suction head available (NPSHA) for pumps handling volatile, multicomponent liquids such as gasoline and kerosene should be based on the true vapor pressure of the particular liquid at the most critical pumping temperature.

The NPSH required by a pump at a given flow is a function of the individual pump characteristics. The NPSH available must exceed the NPSH required by the pump and can be established correctly only when the true vapor pressure is known.

The suction piping should be arranged to avoid any accumulation of vapor. Provisions should be made at the highest point on the suction side of the pump for venting of vapors to prevent vapor lock.

Since the liquid pumped is usually flammable or toxic, special consideration should be given to the shaft sealing area. Frequently, tandem or double mechanical seals are used, which can be instrumented for alarms when seal leakage occurs.

The materials of construction should be selected with regard to the corrosive action of the liquid pumped.

The driver, normally a vertical explosion-proof induction motor, is mounted directly on top of the pump. A vertical solid shaft driver with a rigid type adjustable spacer coupling is usually recommended. Horizontal drivers such as engines or steam turbines may be used in conjunction with a vertical right angle gear drive.

The pump normally is of the open lineshaft type (product-lubricated).

2.3.1.3 Transfer service (pipeline, open sump)

The vertical transfer pump is usually a part of a process or system where liquid requires transport. This type of pump can be constructed in different configurations to suit the application. Examples are:

- the can pump, Figure 2.6;
- short setting pump, Figure 2.7;
- mixed flow pump, Figure 2.8;
- axial flow (propeller) pump, Figure 2.9.

Material of construction can be selected to suit the liquid and service. Various types of construction are available, either open type lineshaft (product-lubricated) or enclosed type lineshaft construction.

2.3.1.4 Dewatering service (mine, flood control, etc.)

The vertical dewatering pump may take suction from a pit, bore hole, well, or mine shaft. The pumped liquid is lifted to the surface for discharge. See:

- Figure 2.4 for well type;
- Figure 2.5 for submersible type;
- Figure 2.7 for short setting type;
- Figures 2.8 and 2.9 for flood control types.

Frequently, mine liquid must be pumped from great depths and may be hot, corrosive and abrasive. A mine pump should be of heavy-duty construction with ample corrosion allowances and liberal margins of safety for pressure containments and heavily stressed areas in the pump. The design should allow for easy renewal of parts subject to corrosion or wear.

Various types of construction are available; either the open type lineshaft (product-lubricated) or the enclosed type lineshaft construction may be used.

For a successful installation, great care is necessary in properly defining all the pump's requirements for the pump manufacturer.

Flood control vertical dewatering pumps are usually of the mixed flow or propeller type (see Figures 2.8 and 2.9). They are normally short setting units capable of high-capacity pumping. This type of pump is particularly suitable for protecting large areas from flooding. It is completely self-priming with the pumping element submerged during flood stage.

The enclosed type lineshaft with oil-lubricated construction protects the lineshaft bearings from abrasives in the liquid pumped and allows lubrication of the bearings independent from the liquid pumped.

The vertical flood control pump can handle silt-laden flood waters effectively. A well-designed sump is necessary. On large pumping stations, it is economically feasible and strongly recommended to perform model sump tests for assurance of sump design integrity.

Many times during flood stages, the potential loss of electrical power mandates a diesel engine driver in lieu of an electrical motor driver. The diesel engine is coupled to a right angle gear which mounts on top of the vertical pump.

The materials of construction for the pump are generally coated steel fabrications with cast iron bowls and bronze impellers. However, when more corrosive flood water such as sea water is encountered, materials of construction with greater corrosion resistance are recommended.

2.3.1.5 Well service

The vertical well pump consists of a pump placed in a circular well (see Figures 2.4 and 2.5). As a minimum, the pump's first stage impeller is submerged in well water to allow pumping to the surface elevation.

When selecting a pump for a specific well, there are a number of considerations for a successful application. The well's inside diameter and straightness determines what diameter bowl assembly can be safely installed. If the well is suspected to be crooked, a dummy bowl assembly can be inserted to check diametrical clearances.

The well static water level (SWL) and pumping water level (PWL) must be determined prior to selecting the pump length (setting). Both the SWL and PWL are subject to fluctuations due to seasonal changes and/or usage patterns. Therefore, it is important to consult with the local well driller, pump installer, or the owner for his setting recommendations.

The well pump's intended use for agricultural, municipal, or industrial applications determines the type of well pump selected. Usually, agricultural installations use an enclosed lineshaft (oil-lubricated) type pump on settings deeper than 100 feet (30 meters) or open lineshaft with special bearing material or special prelubrication flush. Shallower settings may use an open lineshaft (product-lubricated) pump.

Municipal applications tend to use submersible pumps, open lineshaft pumps and occasionally enclosed lineshaft (oil-lubricated) pumps.

Water soluble lubricants are often used to avoid water contamination.

Industrial well pumps usually are selected for a specific use. There are also many well pump applications where the local success with a certain type of pump dictates the construction.

Submersible type well pumps are more commonly used on deeper settings. Advantages include elimination of lineshafts and lineshaft bearings, ability to operate at higher speed and reduced pump and well size. Also, they are easier to install and remove from a well. Disadvantages include a less efficient motor, additional clearance between bowl assembly and well casing for the motor cable and the removal of unit from the well to service the motor.

This type of pump also requires a submerged electrical cable splice and special care in installation to prevent damage to the cable.

There are several well pump construction considerations which must be reviewed for a successful installation. The hydraulic thrust generated by the impellers is transmitted through the lineshaft to the driver thrust bearing. The thrust elongates the lineshaft during operation, which requires that the pump impellers have sufficient axial clearances.

Both foot valves on lineshaft pumps and check valves for submersible pumps must be applied with caution, as high forces can be generated by water hammer, sand locking, etc. Submersible pumps may be supplied with a check valve. However, the valve location in relation to the well water level and the surface valving must be carefully considered.

A non-reverse ratchet is a device which prevents the lineshaft pump from reverse rotation when the pump is shut off and the water in the column pipe returns to the well. Non-reverse ratchets are used on open lineshaft pumps to prevent backspin and thus prevent bearing failure due to lack of lubrication as the column pipe goes dry. The motor and pump manufacturer should be consulted regarding the application of non-reverse ratchets for deeper setting pumps. A post-shutdown lubrication system may be the better solution.

Normally, the industry's standard materials of construction are sufficient for average well water. Special materials of construction are available for corrosive and sandy well waters.

2.3.1.6 Irrigation service

Vertical irrigation pumps are available in several different types. See the following Figures:

- Figure 2.4 for vertical type multistage deep well;
- Figure 2.5 for deep well submersibles;
- Figure 2.7 for vertical type multistage short setting;
- Figure 2.8 for the mixed flow;
- Figure 2.9 for the axial flow propeller type.

Pumps in this service provide water to open irrigation ditches for gravity flow, for surface irrigation to pressurized conduits for sprinkler or drip irrigation, or for lifting water from one elevation to another. The suction supply varies; it can be a well, a pond, a lake, a river, stream or canal.

The deep well submersible and vertical type multistage pumps are for lifting water from wells. These pumps provide sufficient well head pressure for sprinkler or drip irrigation.

The vertical type, multistage short setting pump is used for lifting water from a pond or lake and pressurizing a closed discharge conduit.

The mixed flow and axial flow propeller type pumps are used for low-pressure, high-capacity applications. Generally, the pump sump is custom- designed to fit the application. Larger mixed flow and propeller type pumps, above 2000-3000 GPM (450 – 680 m^3/hr), are used for canal lift stations in the irrigation system.

A well-designed sump is necessary. On large pumping stations, it is strongly recommended to perform model sump tests for assurance of sump design. Pump plugging or clogging with debris is a common problem in irrigation systems. Therefore, large pumps taking water from sumps, lakes, rivers, etc. should have trash racks designed into the sump intake structure. Smaller pumps may get by with suction strainers. Where water is to be pressurized for sprinkler or drip irrigation, it is recommended to use additional strainers beyond the pump to protect the system. Refer to Figures 2.26 through 2.31 for recommendations on sump design. These recommendations are general in nature; the pump manufacturer should be contacted for specific applications.

Standard irrigation service pumps are well-defined with regard to design and materials of construction by the many manufacturers in the industry. Generally, the materials of construction are steel fabrications with cast iron bowls and bronze impellers.

Where electrical power is available, vertical induction motors usually drive the pumps. When internal combustion engine drivers are used, they provide power to a right angle gear drive which mounts on the pump.

There are many successful applications of vertical pumps in irrigation service which are custom-made and not necessarily to industry standards.

2.3.1.7 Utility service (condenser circulating, cooling tower)

Pumps handling cooling water from a source such as a river, lake, ocean, or cooling tower and which pump it through a condenser to condense the steam coming from the turbine generator are commonly termed Condenser Circulating pumps or Cooling Tower pumps. Pumps for this application are of the vertical wet pit, vertical single suction volute, or axially split-case type with two to four pumps normally used per generating unit. When selecting the type of pump, the consulting engineer and end user must consider factors such as suction water level variations, cost of excavation, cost of pumps and their drivers, ease of maintenance, reliability and, of course, past experience. The Vertical wet pit type is discussed in this section.

Vertical wet pit pumps normally take suction from an open sump and discharge into a pipe either above or below ground level. The driver, normally an induction or synchronous motor, is mounted on top of the pump above high water level.

Field problems are often caused by improper sump design, inadequate material selection or inadequate lube water system design. Additional information may be found in the intake design section of this standard. Special emphasis should be placed on these subjects as well as properly matching the pumping equipment to the system head requirements for the entire range of anticipated operation.

2.3.1.8 Condensate and condenser hot well service

Pumps handling condensed steam from a condenser, or other form of surface heat exchange equipment, are commonly termed Condensate pumps. Pumps for this service are usually of the vertical can or the axially split-case type. Since condensate systems normally have very limited NPSH available, the vertical can type is frequently used by excavating the necessary vertical depth to provide adequate NPSH to the pump.

Most condensate pumps have a special first stage impeller design including low inlet velocities, which result in a lower NPSH required. This, in turn, results in shorter pump settings and less excavation.

Stuffing-boxes or mechanical seals operating under vacuum should be provided with a dependable supply of seal water.

The highest point of the suction side of the pump must be vented back to the vapor side of the condenser.

Condensate pumps handle unbuffered water, but due to the low temperatures involved, bronze-fitted pumps usually perform satisfactorily.

Heater drip and drain pumps usually handle condensed steam at high pressures and, therefore, may be considered as condensate pumps.

Material selection is similar to boiler feed pumps for equivalent water conditions.

2.3.1.9 Make-up and service water pumps

Make-up pumps provide replacement water for use in cooling towers and cooling ponds where evaporation and system leakage losses must be made up.

Service water pumps provide station water for auxiliary cooling, lube water systems and other miscellaneous uses.

The make-up pump is normally located in an open sump on a river or lake and provides water to the generating site.

The materials of construction are similar to those of the condenser circulating water pump for equivalent water conditions.

2.3.1.10 Fire protection service

Vertical pumps approved for fire protection service are shown in the following Figures:

- Figure 2.4, the multistage deep well;
- Figure 2.6, the multistage can pump;
- Figure 2.7, the multistage short setting type.

The National Fire Protection Association (NFPA) issues a standard for the installation of vertical fire pumps. This standard is published in their pamphlet No. 20. Always refer to the latest edition of this standard. The NFPA does not approve, inspect, or certify any installation, procedures, equipment, or materials.

The Factory Mutual System (FM) and Underwriters Laboratories, Inc. (UL) approve and list fire pumps which must be designed, manufactured and tested in accordance with their standards. When applying fire pumps, it is necessary to identify the authority having jurisdiction and to obtain the governing standards from that authority. The authority having jurisdiction may be a representative of the federal, state, or local government, the insurance company, or the owner and his designated agent. The use of listed or approved fire pumps is usually mandatory.

2.3.1.11 Pumps used as hydraulic turbines

Vertical pumps of all sizes, types, and specific speeds may be operated in reverse rotation as hydraulic turbines. The most common applications are power recovery from high pressure process and electric power generation from river flow or dams.

While running in the turbine mode, the performance characteristics of a PAT (pump as turbine) differs significantly from pump operation. Special care should be taken in PAT applications to ensure that the mechanical design of the unit is of sufficient proportion to allow safe operation. Frequently, these applications subject the PAT to increased torque and speed levels beyond original pump design values.

Additionally, the head-capacity characteristic is such that torsional stresses increase with increasing capacity. All pumps applied as turbines should be subject to careful calculation of stresses in the shafts. Pumps with semi-open impellers may need special attention to impeller axial clearance.

Precautions should also be taken to insure that the PAT will operate without cavitation. The turbine industry typically uses the terminology TREH (total required exhaust head) and TAEH (total available exhaust head) in place of NPSH. Total exhaust head is defined as the total fluid energy at the runner eye less the vapor pressure of the fluid. Some of the other factors which affect the use of pumps as turbines are:

- runaway speed;
- fluid flow at runaway speed;
- required solids passage;
- blockage from debris.

2.3.2 Selection criteria

2.3.2.1 Performance system requirements

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

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

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

2.3.2.2 Pump versus system curves

A typical simple system and pump curve are shown in Figure 2.14. Note that the pump always operates at the intersection point of the pump and system characteristics curves.

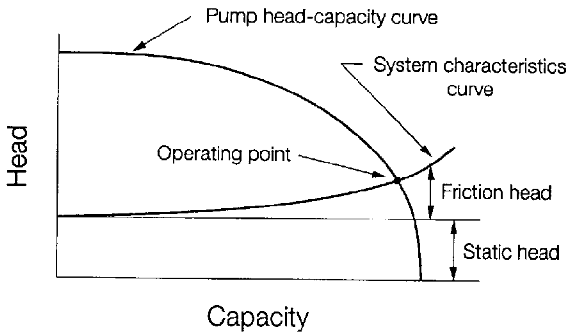


Figure 2.14 — Pump versus system

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

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

It should be noted that most manufacturers' ratings curves are based on the bowl assembly performance. The pump column losses and discharge elbow loss (see Paragraph 2.3.2.23) should be included in the system curve.

2.3.2.3 System pressure limitation

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

The possibility of pressure surges in the system must also be considered, as discussed in more detail in Paragraph 2.3.2.5.

2.3.2.4 Reverse runaway speed

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

If the pump is driven by a prime mover offering little resistance while running backwards, the reverse speed may approach its maximum consistent with zero torque. This speed is called runaway speed. The runaway speed may exceed that corresponding to normal pump operation. This excess speed may impose high mechanical stresses on the rotating parts both of the pump and the prime mover and, therefore, knowledge of this speed during initial design work is essential to safeguard the equipment from possible damage. Refer to Figure 2.42 of the Installation section, Paragraph 2.4.7.8 for more details.

2.3.2.5 Water (hydraulic) hammer analysis

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

The head due to water hammer in excess of normal static head is a function of the destroyed velocity, the time of closure, the size and length of the pipe, and the velocity of pressure wave along the pipe. The value of water hammer can be calculated with a fair degree of accuracy by an engineer thoroughly experienced in this work, provided all of the factors influencing water hammer are known.

Water hammer may be controlled by regulating valve closure time or by application of relief valves, surge chambers and other means. However, vertical pumps whose length from discharge centerline to sump liquid level is greater than atmospheric pressure, typically 34 feet (10 meters), will always produce a vapor or air pocket in the column that upon restart will usually result in water hammer. This will happen regardless of whether the discharge valve is a slow-opening gate valve or a check valve.

The installation of an air-vacuum release valve may be necessary when a vertical pump will have to drive air out of the column when starting. When the pump stops, the air-vacuum release valve allows the water in the column to flow backward through the pump. If the pump discharge is open to the atmosphere, an air-vacuum release valve is not necessary.

It is recommended that specialized engineering services be engaged for such calculations, since few pump users or pump manufacturers have the knowledge and experience necessary for this work.

2.3.2.6 Start-up and shut-down analysis

During start-up, the driver must provide adequate accelerating torque (the additional torque required above normal operating torque) to assure a successful start in a reasonable amount of time. Since torque is directly related to power, the shape of the pump BHP curve becomes very important. If maximum BHP occurs in the normal operating range, a driver sized for the operating range is normally capable of starting the pump. However, if maximum BHP occurs at shut-off, which is common for high specific speed applications, the starting requirements become critical for proper driver selection. In a system with one pump or when starting the first of several units in parallel, the unit can be started with the discharge valves open, or partially open, to avoid operation at shut-off. For high specific speed pumps operating in parallel, there are several options when starting the second and succeeding units. Possible starting procedures range from starting against an open discharge valve at full reverse speed to starting against a closed discharge valve. Either extreme may require a larger or special high torque driver. Normally, an between starting procedure is selected, where the discharge valve is synchronized to start opening prior to, at the same time as, or after the driver is started. The problem then becomes selection of the optimum procedure and time interval for valve opening.

Some of the parameters to consider are:

a) The required valve opening time;

b) The length and diameter of the piping system to determine the effect of the water in the system;

c) The WR^2 of the pump and driver;

d) The four quadrant performance characteristics (Karman-Knapp diagram) of the pump;

e) The speed-torque, speed-time, speed-current and safe time-current characteristics of the motor (if motor driven);

f) The system head curve, the number of pumps in parallel and the available starting voltage.

The resulting analysis will not only ensure that the driver is capable of starting the pump but can result in a lower initial capital investment and higher operating efficiency.

When vertical pumps are started with discharge valves closed, a provision should be made to rapidly vent the column and the head to make sure the lineshaft bearings are lubricated and air is not compressed and then suddenly allowed to expand.

Avoidance of water hammer is the primary concern during the shutdown of a pump, especially in installations with long piping. Gradual closing of the discharge valve is one way to eliminate or reduce water hammer. A mathematical system analysis may be required in some installations to compute the severity of the water hammer.

The possibility of the pump running in reverse direction after a shutdown must also be considered. Where the system makes it a likely occurrence, the driver and the pump must be designed for the maximum reverse rotation speed. Provisions may be made in the driver (anti-rotation device) or in the system (check valves, rapidly closing shutoff valves, siphon breakers) to prevent reverse rotation.

2.3.2.7 Pump and motor speed torque curves

A plot of speed versus torque requirements during the starting phase of a pump is sometimes checked against the speed versus torque curve of the driving motor. The driver must be capable of supplying more torque at each speed than that required by the pump in order to bring the pump up to rated speed. This condition is generally easily attainable with standard induction or synchronous motors, but under certain conditions, such as high specific speed pumps or reduced voltage starting, a motor with high starting torque may be required.

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

2.3.2.8 Determining operating range, series and parallel operation

Many vertical pumps are built in multistage arrangements to produce pressures that a single impeller cannot produce.

The head is additive, while the capacity remains the same. Thus, for example, when a single stage pump at a given speed produces 1000 gpm at 75 feet of head, a four stage pump would produce 1000 gpm at 300 feet of head (4 x 75 feet). This is basically series operation. Separate multistage, vertical pumps can be put in series where it is required for ultimate high pressure or where individual pumps can take care of part of a process. Whatever the system, when operating in series the capacity remains the same and the pressure is additive (see Figure 2.15).

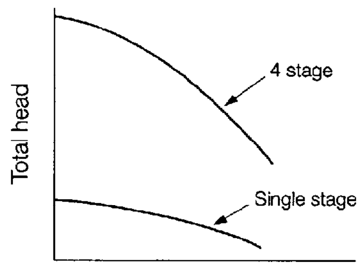


Figure 2.15 — Series operation

Pumps operating in parallel produce a capacity which is additive at the head at which they would run individually. Thus, a system with a pump running at 1000 gpm and 100 feet put in parallel with a pump running at 1500 gpm and 100 feet would pump a total of 2500 gpm at 100 feet if the system consists of only static head. (see Figure 2.16)

2.3.2.9 Continuous, intermittent and cyclic service

Pumps are designed to operate either continuously, intermittently or in cycle duty. Each application requires that the pump and driver be carefully selected for the service specified. If the

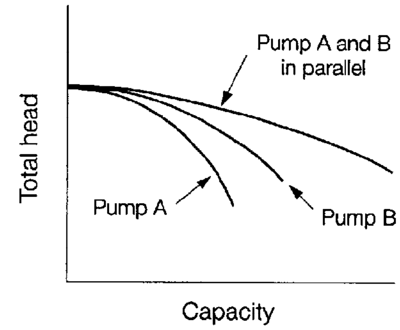


Figure 2.16 — Parallel operation

operating conditions are known, the pump can be designed to meet those conditions. In addition, the driver must also be carefully selected. For instance, an electric motor may be limited to only a few starts per hour to prevent overheating.

2.3.2.10 Range of operation

To determine the operating range, one must determine the minimum and maximum capacity at which the pump must operate continuously. To find these points, one must determine the intersection of the head-capacity curves with the system head curves. Figure 2.17 is an example of two pumps in parallel. The maximum flow always occurs when the pumps are operating at high water

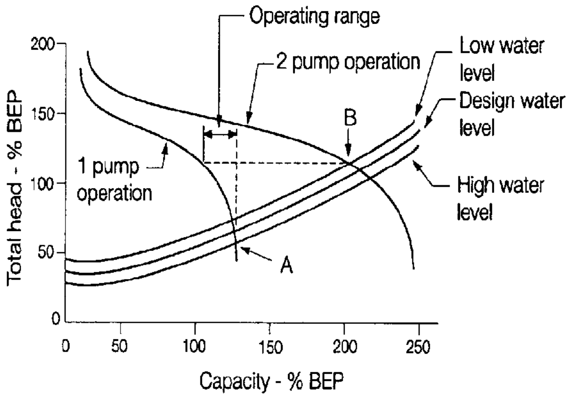


Figure 2.17 — Operating range

level. This means that the pump could operate continuously anywhere between minimum and maximum flow, depending on the suction water level and the number of pumps in operation.

2.3.2.11 Operation away from BEP/minimum flow

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

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

Some causes of problems at reduced capacities are as follows:

- *Temperature rise:* Absorption of the input power into the pumped liquid raises the liquid temperature. Generally, the temperature rise across the pump should be limited to 15°F (8°C) and with a safe margin against flashing;
- *Suction recirculation:* Circulatory flow in the impeller eye at flow operation can cause localized pitting and mechanical damage. The recirculation onset depends on the impeller inlet design. Continuous operation with suction recirculation should be avoided in high-energy pumps;
- *Discharge recirculation:* Circulatory flow in the discharge area of impellers can cause large forces on impeller shrouds, resulting in random axial unbalance of forces and high thrust. Mechanical vibration and bearing failures can occur. The problem is most severe in high-energy pumps. Generally, vertical turbine pumps with heads greater than 250 feet (75 m) per stage and/or more than 300 horsepower (225 kw) per stage are considered high-energy.

  The energy level can be an important consideration for minimum continuous flow, since the destructive forces are greater at high energy levels. Limitations as high as 70 percent of BEP may be required in specialized pump applications of high energy levels. Conversely, for normal energy levels, the required flow for continuous operation may be as low as 20 percent of BEP. Consult the pump manufacturer for recommended minimum flow requirements on any specific application;
- *Net positive suction head:* In some designs, the NPSH required by the impeller increases at low flows and noise, impeller pitting and other symptoms of cavitation can occur. The pump manufacturer's performance curve should be checked for NPSH requirements.

When a pump operates out on the head-capacity curve, problems can occur due to one or more of the following causes:

- *Net positive suction head:* NPSH required by the impeller increases with flow, and noise, impeller pitting, and other symptoms of cavitation can occur if the NPSH available is inadequate. The pump manufacturer's performance curve should be checked for NPSH requirements;
- *Vortexing:* For a given intake structure or sump design, the tendency to vortex increases with flow. A sump design should be evaluated for adequacy at flows greater than the rated flow in order to avoid surface and submerged vortices. Vortices can affect hydraulic performance and create unsteady bearing loads, causing increased wear and vibration. Submerged vortices can cause localized inlet cavitation in various parts of the suction area when the absolute static head in the vortex core reaches the liquid vapor pressure;
- *Flow separation:* Cavitation pitting due to flow separation can occur at flows higher than normal where the angle of attack of the liquid differs significantly from normal;
- *Vibration:* Vertical pumps operate best at the BEP condition. Increased vibration is to be expected at capacities greater than BEP;
- *Upthrust:* Some vertical pump designs experience upthrust to the right of the BEP condition. Care should be taken to determine the upthrust characteristic of the pump

and that the pump has adequate upthrust capacity.

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

2.3.2.12 Noise levels

See Paragraph 2.4.9 in the Installation section 2.4.

2.3.2.13 Suction conditions

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

The suction bell must be well below the water surface and the intake or sump must be of a functionally correct design. This is true for all specific speed designs. Additional information on intake design may be found in Paragraph 2.3.5.

2.3.2.14 Submergence

Submergence is a term used to relate the setting of an immersed pump to the surface level of the suction liquid.

The submergence of a pump is the vertical distance from the suction water level to the lip of the suction bell. It is a linear dimension partially describing a system and cannot be substituted for a dynamic term such as NPSHA. Minimum submergence is often specified by the pump manufacturer to help prevent vortices. Please refer to Figures 2.18 through Figure 2.31 for recommendations.

2.3.2.15 Net positive suction head available (NPSHA)

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

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

Where:

h_{sa} = total suction head in feet absolute

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

Then:

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

Where:

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

Z_s = static elevation of the liquid above the datum point of the pump expressed in feet (meters). If the liquid level is below the pump datum, Z_s is a negative value;

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

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

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

Where:

p_a = Absolute pressure expressed in psia. In an open system, p_a equals atmospheric pressure expressed in psia;

p_{vp} = vapor pressure expressed in psia;

γ = specific weight of liquid at the pumping temperature in pounds per cubic feet.

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

$$NPSHA = Z_s - h_f$$

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

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

Where:

h_{atm} = atmospheric pressure, expressed in feet (meters) absolute;

h_g = gauge head at the suction of the pump expressed in feet (meters) of liquid. h_g is a negative value if it is below atmospheric pressure;

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

Z_s = distance between suction datum and suction gauge.

2.3.2.16 NPSHA corrections for temperature and elevation

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

Applications with water in an open system at sea level with a pumping temperature of 85°F are common. Given γ = 62.4 lbs, per cu ft.

$p_a = 14.7\ psi$, $p_{vp} = 0.6\ psi$, $Z_s = 10.0$ ft *and* $h_f = 0$

We find:

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

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

$$= 42.5 \text{ ft}$$

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

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

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

$$= 27.1 \text{ ft}$$

To find the NPSHA for 180°F water at 5,000 feet elevation p_a = 12.25 psi), proceed as follows:

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

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

$$= 21.3 \text{ ft}$$

NOTE – The correction for elevation is approximately one foot (0.3 meters) per 1,000 feet (300 meters) of elevation.

2.3.2.17 NPSH margin considerations

Any system must be designed such that the net positive suction head available (NPSHA) is equal to, or exceeds, the net positive suction head required (NPSHR) by the pump throughout the range of operation. Margin is the amount by which NPSHA exceeds NPSHR. The amount of margin required varies depending on the pump design, the application and the materials of construction.

NPSH Required (NPSHR) is defined as the NPSH at which the pump total head (first stage head in multistage pumps) has decreased (dropped) by 3% due to low suction head and resulting cavitation within the pump.

Practical experience over many years has shown that, for the majority of pump applications and designs, NPSHR can be used as the lower limit for the NPSH available. However, for high-energy pumps providing NPSHA equal to the NPSHR may not be sufficient. Therefore, the purchaser should consider an appropriate margin for NPSHA over NPSHR for high-energy pumps which is sufficient at all flows to protect the pump from damage caused by cavitation.

Cavitation begins to develop in a pump as very small harmless vapor bubbles, substantially before any degradation in the developed head can be detected. This is called the point of incipient cavitation. It can take from 2 to 20 times the NPSHR to fully suppress incipient cavitation, depending on the impeller design and operating capacity.

Some studies on high-energy applications show that cavitation damage with NPSHA greater than the NPSHR can be substantial. In fact, there are studies on pumps which show the maximum damage to occur at NPSHA values somewhere between 0% and 1% head drop (or 2 to 3 times the NPSHR), especially for high suction pressures as required by pumps with high impeller eye peripheral speeds.There is no universally accepted relationship between the percent head drop and the damage due to cavitation. There are too many variables in the specific pump design and materials, properties of the liquid and system.

The pump manufacturer should be consulted about NPSH margins for the specific pump type and intended service on high-energy, low-NPSHA applications. Based on a Hydraulic Institute study of data contributed by pump manufacturers, there is no correlation between specific speed, suction specific speed, or any other simple variable and the shape of the NPSH curve break-off. The design variables and manufacturing variables are too great. This means that there is no standard relationship between 3%, 2%, 1% or 0% head drop. The ratio between the NPSHR for 0% head drop and the NPSHR for 3% head drop is not a constant but generally varies over a range from 1.05 to 2.5. NPSHR for 0%, 1%, or 2% head drop cannot be predicted by calculation, given NPSHR.

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

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

The consistency of results which have been obtained on tests which have been conducted with both water and hydrocarbon fluids suggests that the NPSH required by a vertical pump may be reduced when handling any liquid having relatively high vapor pressure at pumping temperature. However, since available data are limited to the liquids for which temperature and vapor pressure relationships are shown on Figure 2.18, application of this chart to liquids other than hydrocarbons and water is not recommended except where it is on an experimental basis.

2.3.2.19 Effects of entrained air or gas

Under a number of different circumstances such as an improperly designed intake or sump, vertical pumps may be required to handle a mixture of air and water or similar mixtures. It is known that this reduces the head, capacity and efficiency, even when relatively small percentages of air or gas are present.

Deterioration of performance for a given percentage of air or gas by volume varies from pump to pump, depending on rotating speed, specific speed, pump size, suction pressure, discharge pressure, number of stages and various special design features (see Paragraph 2.4.2.4 in the Installation section). These mixtures may also have a detrimental effect on the mechanical operation of the pump.

2.3.2.20 Effects of handling viscous liquids

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

Figures 2.19 and 2.20 provide a means of determining the performance of a conventional vertical pump handling a viscous liquid when its performance on water is known. Figures 2.19 and 2.20 can also be used as an aid in selecting a pump for a given application. The values shown in Figure 2.20 are averaged from tests of conventional single stage pumps of 2- to 8-inch (50 to 400 mm) size handling petroleum oils. The values shown in Figure 2.19 were prepared from other tests on several smaller pumps [1 inch (25 mm) and below]. The correction curves are, therefore, not exact for any particular pump.

When accurate information is essential, performance tests should be conducted with the particular viscous liquid to be handled.

2.3.2.20.1 Limitations on use of viscous liquid performance correction chart

Reference is made to Figures 2.19 and 2.20. Since these charts are based on empirical rather than theoretical considerations, extrapolation beyond the limits shown would go outside the experience range which these charts cover and is not recommended.

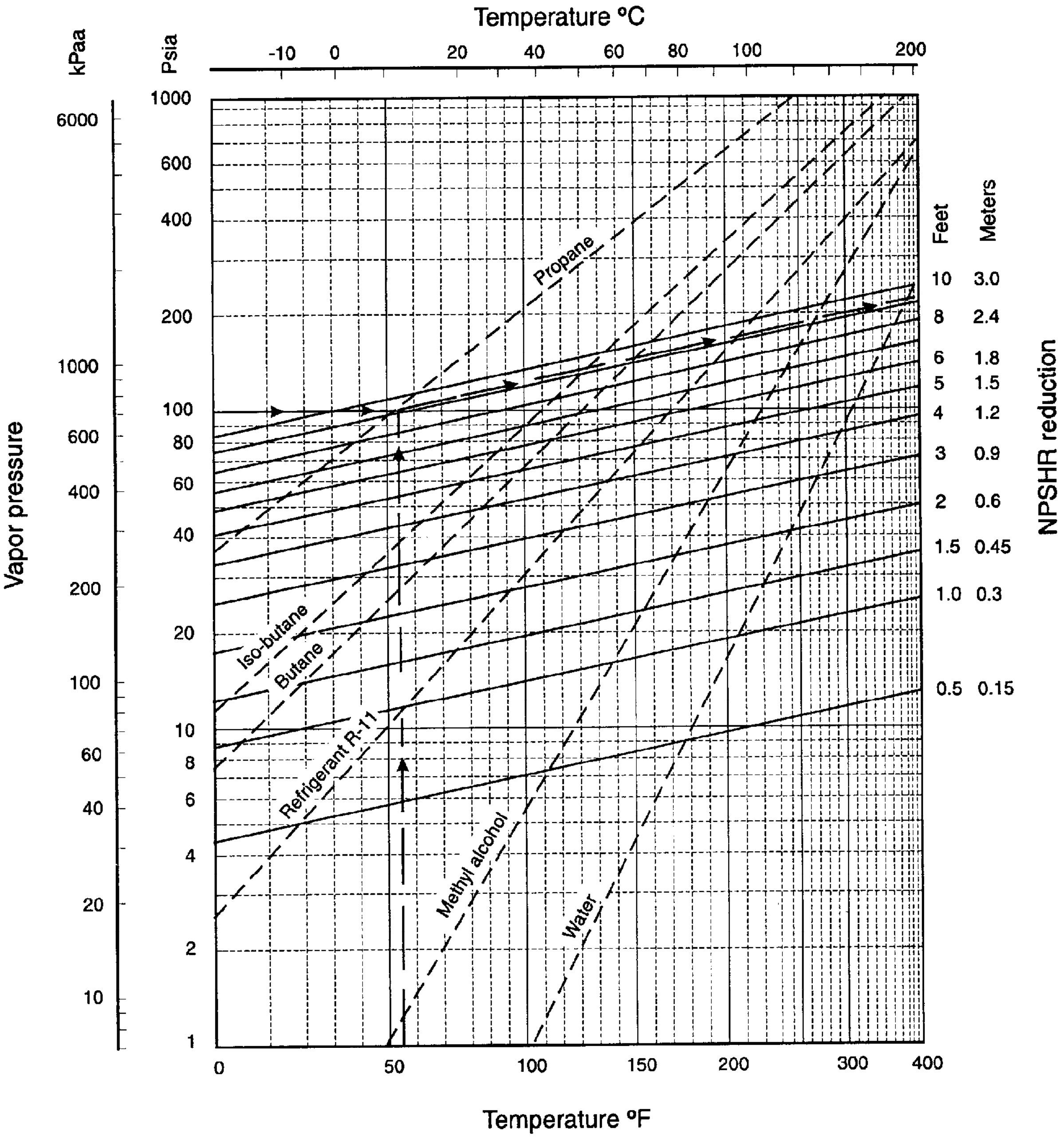


Figure 2.18 — NPSHR reduction for pumps handling hydrocarbon liquids and high-temperature water

Use only for pumps of conventional hydraulic design, in the normal operating range, with open or closed impellers. Do not use for mixed flow or axial flow pumps or for pumps of special hydraulic design for either viscous or non-uniform liquids.

Use only where adequate NPSH is available to avoid cavitation.

Use only on Newtonian (uniform) liquids. Gas, slurries, paper stock and other non-uniform liquids may produce widely varying results, depending on the particular characteristics of the liquids.

2.3.2.20.2 Symbols and definitions used in determination of pump performance when handling viscous liquids

Q_{vis} = Viscous capacity in gpm. The capacity when pumping a viscous liquid;

H_{vis} = Viscous head in feet. The head when pumping a viscous liquid;

η_{vis} = Viscous efficiency in percent. The efficiency when pumping a viscous liquid;

P_{vis} = Viscous brake horsepower. The horsepower required by the pump for the viscous conditions;

Q_W = Water capacity in gpm. The capacity when pumping water;

H_W = Water head in feet. The head when pumping water;

η_W = Water efficiency in percent. The efficiency when pumping water;

s = Specific gravity;

C_Q = Capacity correction factor;

C_H = Head correction factor;

C_η = Efficiency correction factor;

Q_{BEP_W} = Water capacity at which maximum efficiency is obtained.

The following equations are used for determining the viscous performance when the water performance of the pump is known:

$$Q_{vis} = C_Q \times Q_W$$

$$H_{vis} = C_H \times H_W$$

$$\eta_{vis} = C_\eta \times \eta_W$$

$$P_{vis} = \frac{Q_{vis} \times H_{vis} \times s}{3960 \times \eta_{vis}}$$

C_Q, C_H and C_η are determined from Figure 2.19 and Figure 2.20 which are based on water performance. Figure 2.19 is to be used for small pumps having capacity at best efficiency point of less than 100 gpm (25 m^3/hr)(water performance).

The following equations are used for approximating the water performance when the desired viscous capacity and head are given and the values of C_Q and C_H must be estimated from Figure 2.19 and Figure 2.20 using Q_{vis} and H_{vis} as:

$$Q_W \text{ (approx.)} = \frac{Q_{vis}}{C_Q}$$

$$H_W \text{ (approx.)} = \frac{H_{vis}}{C_H}$$

2.3.2.20.3 Instructions for preliminary selection of a pump for a given head-capacity-viscosity condition

Given the desired capacity and head of the viscous liquid to be pumped, and the viscosity and specific gravity at the pumping temperature, Figures 2.19 or 2.20 can be used to find approximate equivalent capacity and head when pumping water.

In the appropriate chart, enter the desired viscous capacity (Q_{vis}) at the bottom and proceed upward to the desired viscous head (H_{vis}) in feet of liquid. For multistage pumps, use head per stage. Proceed horizontally (either right or left) to the fluid viscosity, and then go upward to the correction curves. Divide the viscous capacity (Q_{vis}) by the correction factor (C_Q) to get the approximate equivalent water capacity (Q_W approximately). Divide the viscous head (H_{vis}) by the head correction factor (C_H) from the curve marked: 1.0 x Q_{BEP} to get the approximate equivalent water head (H_W approximately).

Using this new equivalent water head-capacity point, select a pump in the usual manner. The viscous efficiency and the viscous brake horsepower may then be calculated.

This procedure is approximate, as the scales for capacity and head on the lower half of Figure 2.19 or Figure 2.20 are based on water performance. However, the procedure has sufficient accuracy for most pump selection purposes. Where the corrections are appreciable, it is desirable to check the selection by the method described below.

Example: Select a pump to deliver 750 gpm at 100 feet total head of a liquid having a viscosity of 1000 SSU and a specific gravity of 0.90 at the pumping temperature. Enter the chart (Figure 2.20) with 750 gpm, go up to 100 feet head, over to 1000 SSU, and then up to the correction factors:

$C_Q = 0.95;$

$C_H = 0.92$ (for $1.0\ Q_{BEP_W}$);

$C\eta = 0.635;$

$$Q_W = \frac{750}{0.95} = 790 \text{ gpm};$$

$$H_W = \frac{100}{0.92} = 108.8 = 109 \text{ feet head.}$$

Select a pump for a water capacity of 790 gpm at 109 feet head. The selection should be at or close to the maximum efficiency point for water performance. If the pump selected has an efficiency on water of 81 per cent at 790 gpm, then the efficiency for the viscous liquid will be as follows:

$$\eta_{vis} = 0.635 \times 81\% = 51.5\%$$

The brake horsepower for pumping the viscous liquid will be:

$$P_{vis} = \frac{750 \times 100 \times 0.90}{3960 \times 0.515} = 33.1$$

For performance curves of the pump selected, correct the water performance as discussed below.

2.3.2.20.4 Instructions for determining pump performance on a viscous liquid when performance on water is known

Given the complete performance characteristics of a pump handling water, determine the performance for a liquid of a specified viscosity.

From the efficiency curve, locate the water capacity ($1.0 \times Q_{BEP_W}$) at which maximum efficiency is obtained.

For this capacity, determine the capacities ($0.6 \times Q_{BEP_W}$), ($0.8 \times Q_{BEP_W}$) and ($1.2 \times Q_{BEP_W}$).

Enter the chart at the bottom with the capacity at best efficiency ($1.0 \times Q_{BEP_W}$), go upward to the head developed (in one stage) (H_W) at this capacity, then horizontally (either left or right) to the desired viscosity, and then proceed upward to the various correction curves.

Read the values of ($C\eta$), (C_Q) and (C_H) for all four aspects.

Multiply each head by its corresponding head correction factor to obtain the corrected heads. Multiply each efficiency value by ($C\eta$) to obtain the corrected efficiency values which apply at the corresponding corrected capacities.

Plot corrected head and corrected efficiency against corrected capacity. Draw smooth curves through these points. The head at shut-off can be taken as approximately the same as that for water. Calculate the viscous brake horsepower (P_{vis}) from the formula given in Paragraph 2.3.2.20.2.

Plot these points and draw a smooth curve through them which should be similar to and approximately parallel to the brake horsepower (P_p) curve for water.

Example: Given the performance of a pump (Figure 2.21) obtained by test on water, plot the performance of this pump when handling oil with a specific gravity of 0.90 and a viscosity of 1000 SSU at pumping temperature.

On the performance curve (Figure 2.21) locate the best efficiency point which determines Q_{BEP_W}. In this example, it is 750 gpm. Tabulate capacity, head and efficiency for (0.6 x 750), (0.8 x 750), (1.0 x 750) and (1.2 x 750).

Using 750 gpm, 100 feet head and 1000 SSU, enter the chart and determine the correction factors. These are tabulated in Table 2.4. Multiply each value of head, capacity and efficiency by its correction factor to get the corrected values. Using the corrected values and the specific gravity, calculate brake horsepower. These calculations are shown in Table 2.4. Calculated points are plotted in Figure 2.21, and corrected performance is represented by dashed curves.

Figure 2.19 is used in the same manner as Figure 2.20 except that only the best efficiency point corrected performance is obtained. Through the corrected head-capacity point, draw a curve similar in shape to the curve for water performance and having the same head at shutoff. The corrected efficiency point represents the peak of

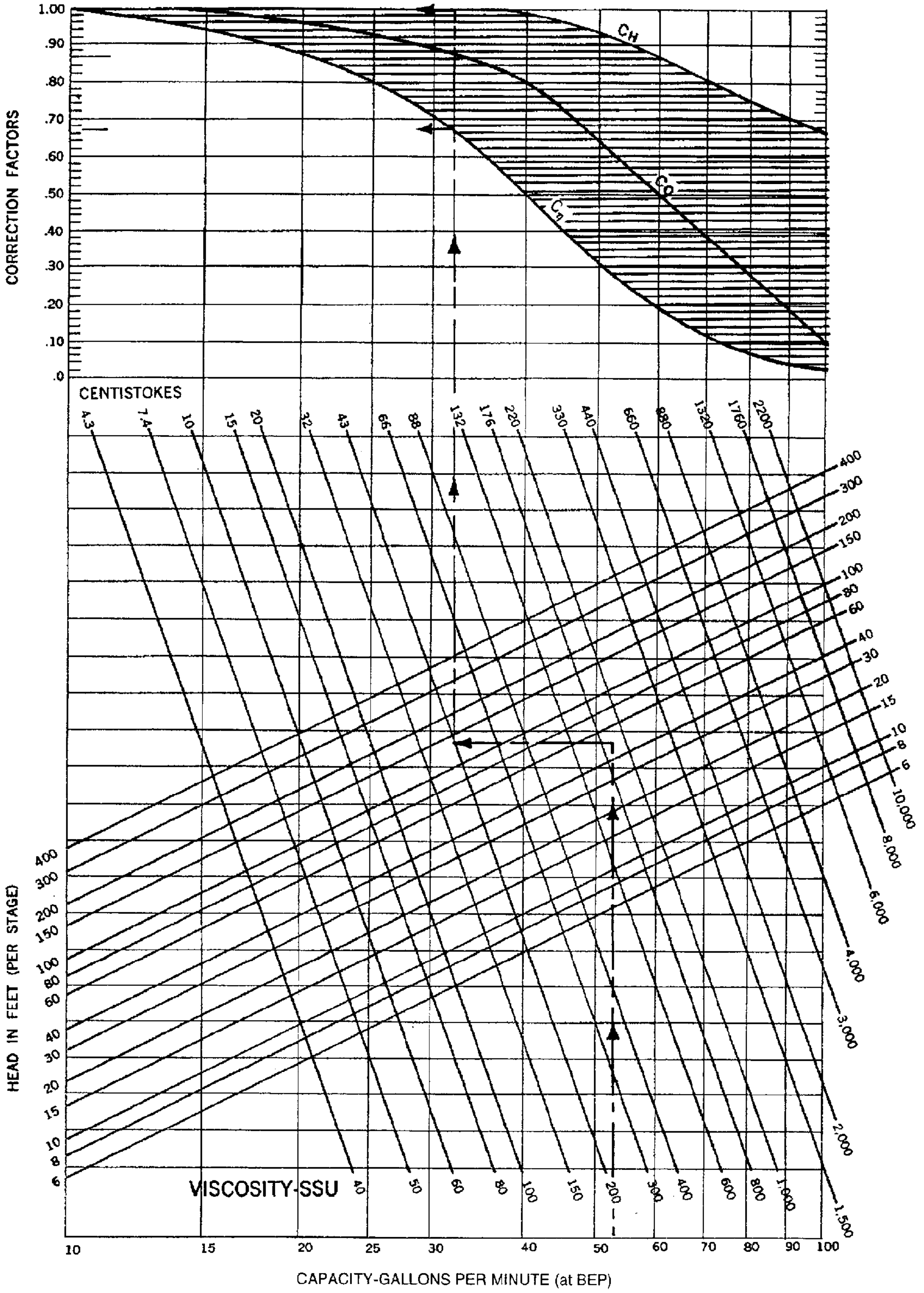


Figure 2.19 — Performance correction chart for viscous liquids

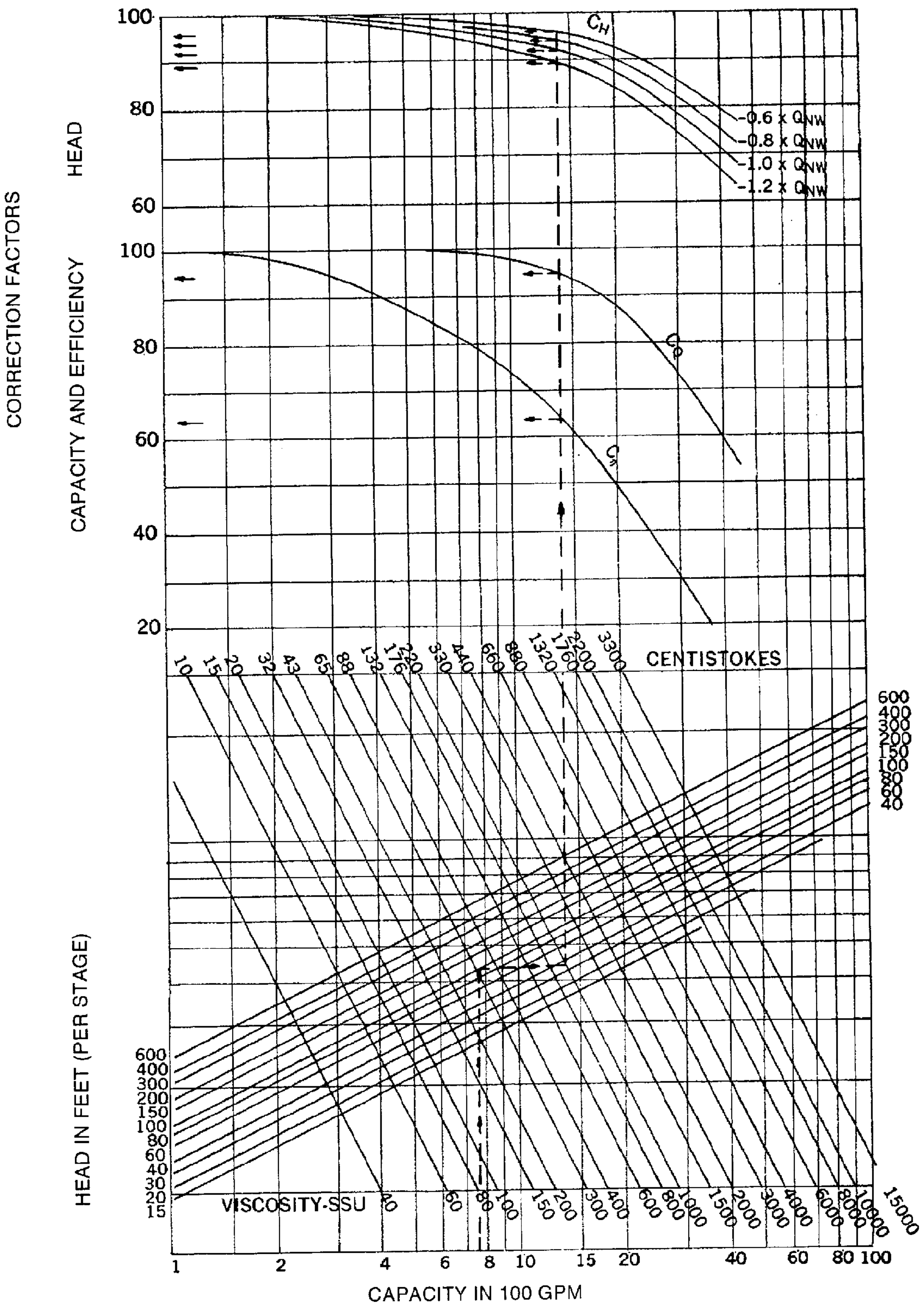


Figure 2.20 — Performance correction chart for viscous liquids

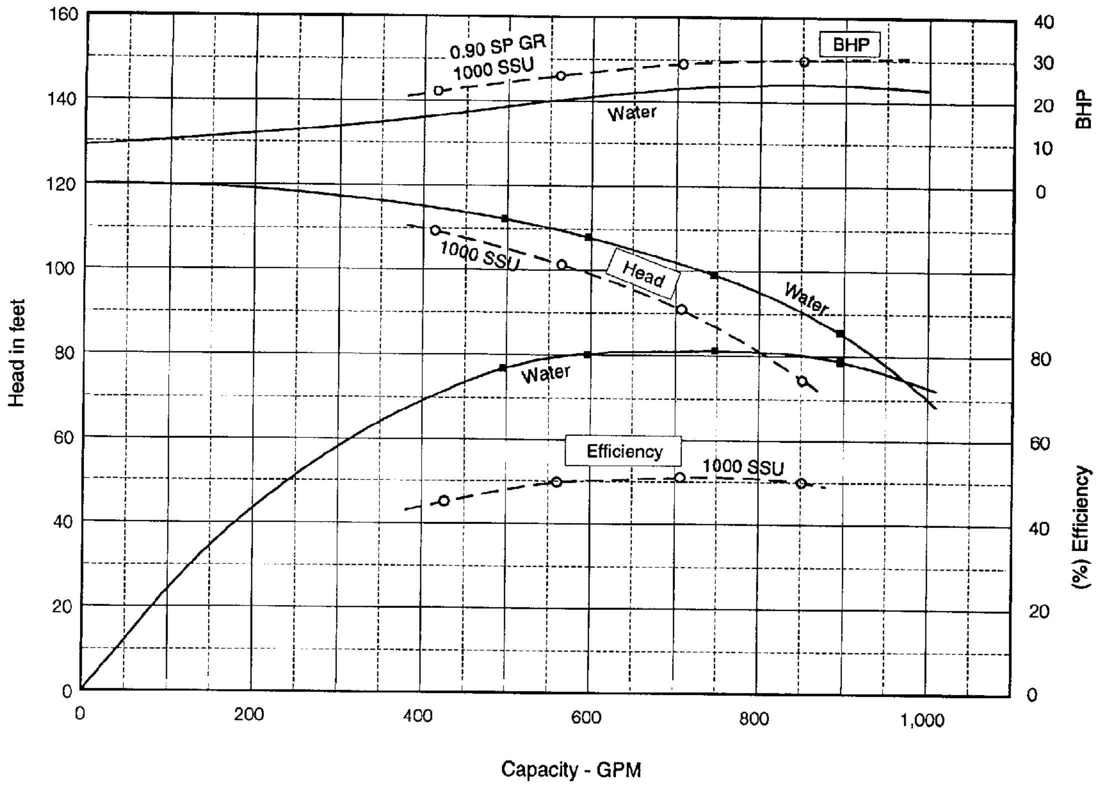


Figure 2.21 — Sample performance chart

Table 2.4 — Sample calculations

	0.6 x Q_{BEP_W}	0.8 x Q_{BEP_W}	1.0 x Q_{BEP_W}	1.2 x Q_{BEP_W}
Water capacity (Q_W)	450	600	750	900
Water head in feet (H_W)	114	108	100	86
Water efficiency (η_W) (%)	72.5	80	82	79.5
Viscosity of liquid	1000 SSU	1000 SSU	1000 SSU	1000 SSU
C_Q from chart	0.95	0.95	0.95	0.95
C_H from chart	0.96	0.94	0.92	0.89
C_η from chart	0.635	0.635	0.635	0.635
Viscous capacity: $Q_W \times C_Q$	427	570	712	855
Viscous head: $H_W \times C_H$	109.5	101.5	92	76.5
Viscous efficiency: $\eta_W \times C_\eta$	46.0	50.8	52.1	50.5
Specific gravity of liquid	0.90	0.90	0.90	0.90
bhp viscous	23.1	25.9	28.6	29.4

the corrected efficiency curve, which is similar in shape to that for water. The corrected brake horsepower curves are generally parallel to that for water.

2.3.2.21 Suction specific speed

Suction specific speed (S), is an index number descriptive of the suction characteristics of a given pump. It is defined as:

$$S = \frac{n\sqrt{Q}}{(NPSHR)^{3/4}}$$

Where:

S = Suction specific speed (for multistage pumps, S is referenced to the first stage);

n = Rotative speed in revolutions per minute;

Q = Flow in gallons per minute at optimum efficiency (use half of the total flow for double suction pumps);

NPSHR = Net positive suction head required in feet, based on a head drop of 3% (see Paragraph 2.3.2.17).

The numerical value of S is mainly a function of the impeller inlet design. Higher numerical values of S are associated with better NPSH capabilities. For pumps of normal design, values of S vary from 6,000 to 12,000. In special designs, including inducers, higher values can be obtained.

2.3.2.22 Rotative speed limitations

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

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

$$n = \frac{S(NPSHA)^{3/4}}{(Q)^{1/2}}$$

The curve presented on Figure 2.22 of this Standard is based on a suction specific speed of 8,500 while operating at or near best efficiency. This represents a practical value for a typical pump handling cold water and fluids with similar properties. Obviously, operating speeds may be lower than the ones shown.

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

Example: Given a capacity of 90,000 gpm and NPSHA of 50 feet, what is the rpm limit for 8,500 suction specific speed?

$$n = \frac{S(NPSHA)^{3/4}}{Q^{1/2}}$$

$$n = \frac{8500(50)^{3/4}}{90{,}000^{1/2}}$$

$$n = 533$$

Therefore, the recommended maximum operating rpm is 533.

From Figure 2.21, note that the intersection of the vertical line for 90,000 gpm and the horizontal line for 50 feet of NPSHA corresponds to 533 rpm.

2.3.2.23 Losses

In determining the overall efficiency of the pump, one must consider the losses within the pump from the suction bell to the discharge elbow.

The suction bell is a flared tubular section for directing the flow of the liquid into the impeller. The losses at the suction bell are a combination of entrance and friction losses.

Bowl losses are unique to each pump design and are obtained through performance testing. The bowl efficiencies are usually published by the pump manufacturer.

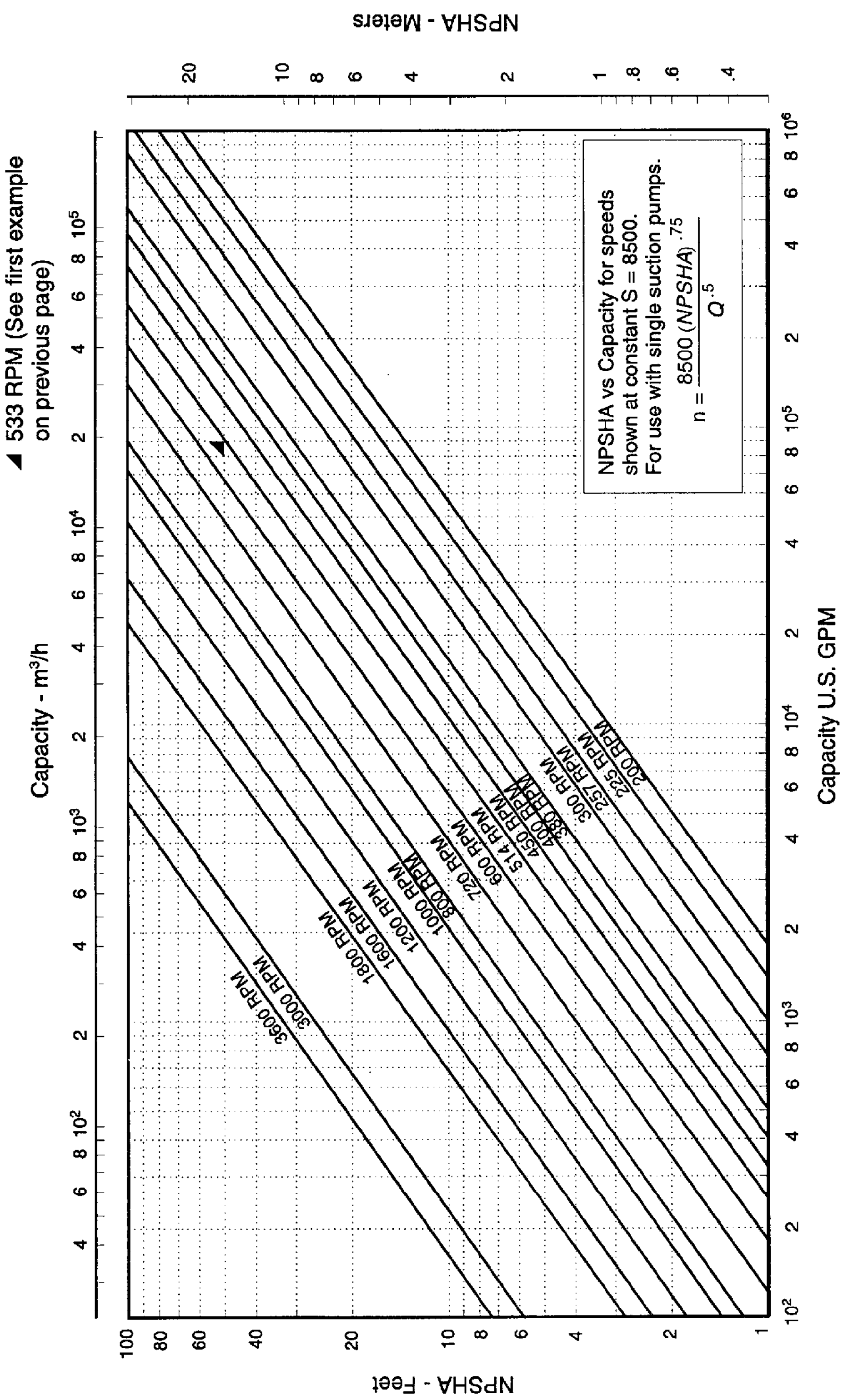


Figure 2.22 — Recommended maximum operating speeds for single suction pumps

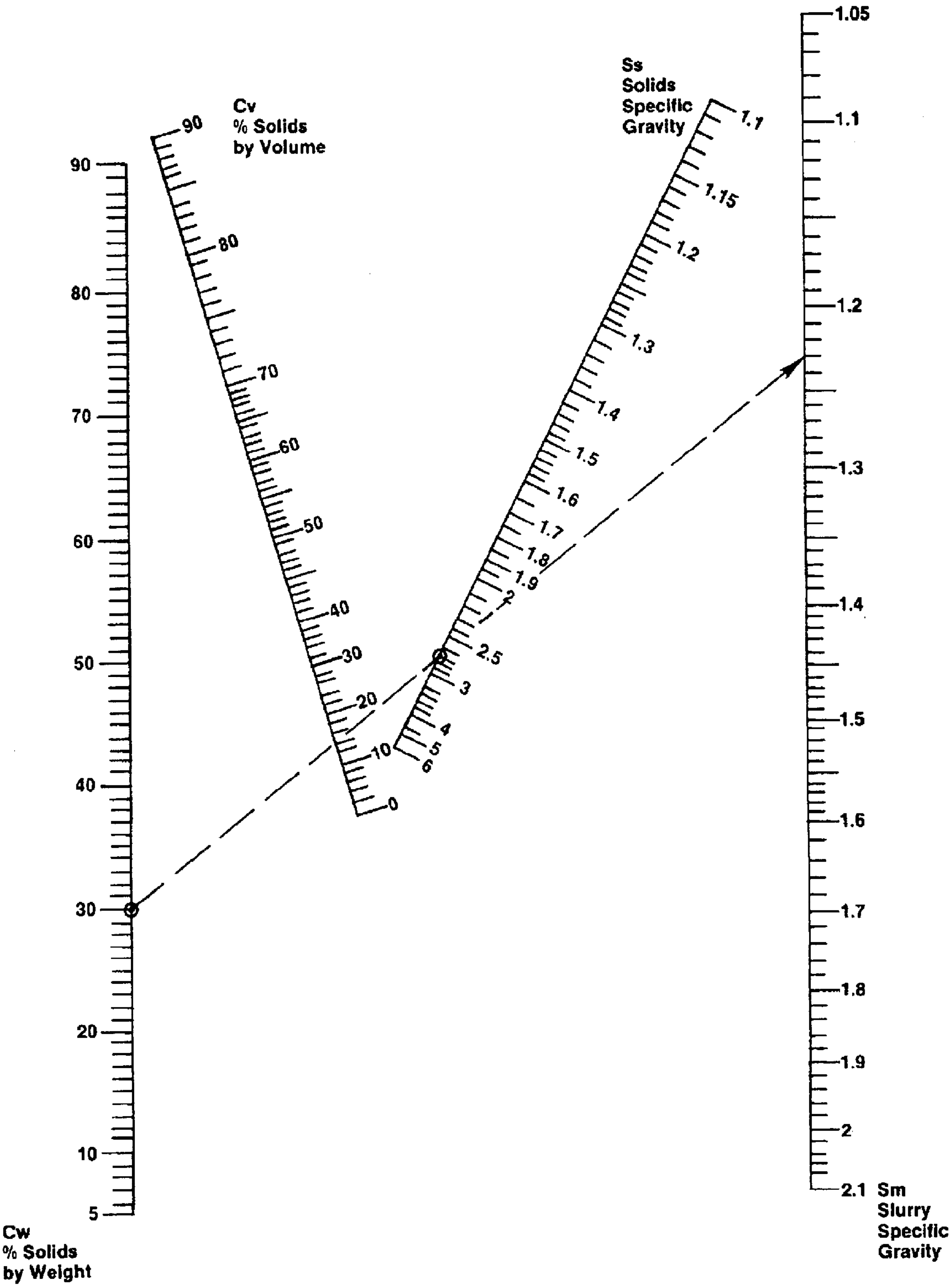


Figure 2.23 — Nomograph of the relationship of concentration to specific gravity in aqueous slurries

The losses in the column sections are due to friction on the column and shaft. An estimate of these losses can be obtained from the *HI Engineering Data Book, Second Edition.* Refer to Figure IIIB-3(a) for column hydraulic loss and Chart VE for lineshaft horsepower losses.

The losses in the discharge head or elbow are a function of each design. If test data is not available, refer to Figure IIIB-3(b) of the *HI Engineering Data Book, Second Edition* for an estimate.

Bearing spiders are used to support the lineshaft bearings and transmit the radial forces from the shaft to the outer column. The friction loss in bearings and spiders varies for different pump manufacturers. In addition, the method of lubrication affects the bearing losses. These losses must be supplied by the pump manufacturer.

2.3.2.24 Effects of handling slurry liquids

Vertical slurry pumps may be used for both inplant process and pipeline applications where heads are not high enough to warrant the use of reciprocating or rotary pumps. The other factors which affect the selection of vertical slurry pumps are:

- Capacity;
- Pressure;
- Abrasiveness (i.e. particle size, density, concentration, shape, hardness);
- Pump performance (i.e. particle size, density, concentration, carrier, viscosity).

Vertical slurry pumps are commonly applied for capacities from 10 GPM to 20,000 GPM with heads up to 300 feet per stage. Pumps may be installed in series for high head and severely abrasive applications.

There are many different slurry pump designs available to accommodate various industrial applications. Those applications include the pumping of solids encountered in mineral ore treatment, dredging, sewage handling, land reclamation, paper manufacture, solids transportation and chemical processing.

2.3.2.24.1 Performance changes

The characteristic performance curve of a vertical pump differs from its clear water performance when solids are included and the flow becomes two phase, i.e., the head and efficiency decrease. The magnitude of the reduction and the shape of the characteristic curve will depend mainly on solids size, volumetric concentration and density. The pump horsepower increases directly with the slurry specific gravity. A nomograph relationship between concentration and specific gravity for aqueous slurries is shown in Figure 2.23.

The pump manufacturer will make allowances in the pump selection for head and efficiency reduction, provided the slurry characteristics are defined.

2.3.2.24.2 Non-settling slurries

Slurries with a narrow band distribution of small particles where the average size is usually less than 100 microns are "non-settling" and behave as Newtonian liquids. Standard viscosity correction procedures can be used, provided the apparent viscosity of the slurry is known. See Figure 2.24 for typical performance characteristics.

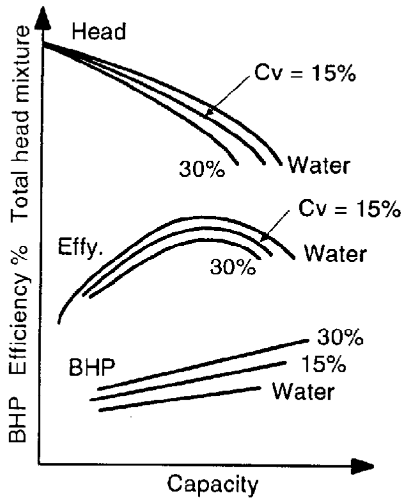


Figure 2.24 — Typical performance characteristics of non-settling slurries

Non-settling slurries which have higher apparent viscosities, such as pastes, filter cakes, etc., should be pumped at lower velocities to minimize friction losses in the system.

2.3.2.24.3 Settling slurries

Slurries with a distribution of larger particles are "settling", and the particles and the liquid exhibit their own characteristics, since energy is dissipated due to liquid drag. See Figure 2.25 for typical performance characteristics.

The critical factor governing a system handling a watery slurry in which the solids have a much

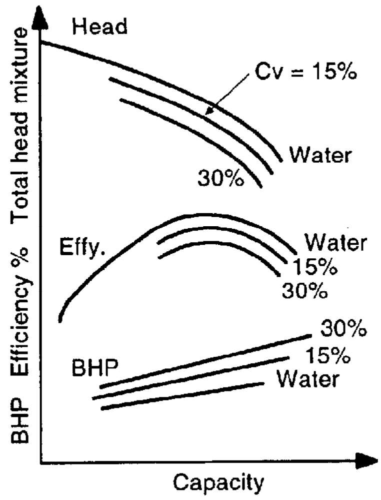


Figure 2.25 — Typical performance characteristics of settling slurries

higher specific gravity than the carrier liquid is the settling characteristic. Coarse solids with high settling rate can only be carried in a vertical pump with many precautions to prevent plugging, draining and squeezeout.

In applying vertical slurry pumps to handle settling slurries, one must be certain that the head requirements of the system above the critical carrying velocity would be met by the pump. If the head produced is insufficient, the capacity is reduced and the solids will settle in the line. Since the head-capacity curve of most slurry pumps has little slope, such an increase can greatly reduce the volume pumped, further reducing the flow velocity and leading to plugging in the system pipe. This situation can usually be avoided by using conservative values for the slurry critical carrying velocity.

2.3.2.24.4 Materials of construction

Pumps designed to resist abrasion are normally made of hard metals (abrasion-resistant cast irons and steels), elastomers, or ceramics. As a general guideline, hard metals are often used in applications characterized by large, sharp-edged solids, elastomers for small round-edged solids. Either high-chrome irons or elastomers are used for their corrosion resistance. In special applications with low head requirements, solid ceramic lined pumps are used for liquids containing fine material.

Some pump design techniques to minimize wear are:

- Utilize pumping elements which are harder than the hardest slurry particles;
- Utilize pumping elements which combine soft and hard materials in such a fashion as to reduce abrasion and provide resiliency;
- Increase material thicknesses in areas of high wear;
- Utilize hydraulic designs with specific speeds of 1400 (1600) or less.

2.3.2.24.5 Rotative Speed

Speed is one of several contributors to wear rate. With abrasive solids, wear rate is generally proportional to relative velocity between the slurry and the pump elements to the power of two or three. While the relative impeller velocity is dependent on the total head developed, it has also been demonstrated that for the same total head, lower-speed pumps exhibit longer life than higher-speed pumps.

2.3.3 Features

2.3.3.1 Types of bearings and spacing

Normally, the bowl and the lineshaft bearings are fluted rubber or of bronze construction, but they can be of other material such as carbon, Teflon, or tungsten carbide.

For vertical turbine pumps, most lineshaft bearings are spaced at intervals not exceeding ten feet.

The number of lineshaft bearings and spacing for large or custom-type pumps is typically determined by critical speed analysis.

2.3.3.2 Lubrication systems

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

Vertical pumps with open lineshaft may require prelubrication of any bearing not submerged prior to pump start-up. A manual or automatic system may be used, providing a compatible liquid is from an external source. An automatic system normally incorporates a solenoid valve in the lubrication line and a time delay relay in the control equipment.

Vertical pumps with enclosed lineshaft must have provisions for lubrication from an external source. This applies to water, grease, or oil lubrication. Again, prelubrication action can be manual or automatic.

The shaft-enclosing tube is provided to protect the shaft and bearings from the liquid being pumped and to provide a means for abrasive-free water, grease, or oil to lubricate the bearings. For some applications, lubrication of the tail bearing and bowl bearing through the use of rifle-drilled pump shafts may be used.

2.3.3.3 Shafting

The lineshaft shall be of a material and size that will transmit the torque from the driver to the impellers and support the maximum thrust load with a proper factor of safety. They should be straight, with the ends faced square to the axis and incorporate a center relief.

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

On larger diameter shafts where threaded connections would be unmanageable, a coupling connection built to accommodate keys to transmit torque and split thrust rings or other means to transmit thrust should be used.

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

2.3.3.4 Impeller types

An enclosed impeller has the vanes fully enclosed with a suction shroud on the inlet side and hub shroud on the back. It is commonly used for all pump types in the low- and medium-specific speed range.

A semi-open impeller has the vanes only enclosed on the back with a hub shroud, with the exposed vanes on the inlet side running in close proximity to a matching case wall, liner or cone. This type of impeller is commonly used over the full specific speed range and particularly for services where moderate amounts of stringy material are present in the pumped fluid.

An open or axial flow impeller has a single inlet with the flow entering axially and discharging nearly axially. Impellers of this type are sometimes called propellers and do not have a shroud. This type of impeller is typically used for low-head applications and has a high specific speed number.

2.3.3.5 System piping and foundation

The foundation should be designed to absorb vibration and to form a permanent, rigid support for the pump base or sole plate.

The system piping should not exert excessive forces and moments on the pump discharge head. Pipe strains are a common cause of misalignment, hot or worn bearings and vibration.

See Paragraph 2.4.1.7 in the Installation section 2.4 entitled "Foundation requirements" and Paragraph 2.4.3 entitled "Suction and discharge piping" for details.

2.3.4 Drivers

In addition to being able to start the pump, the driver shall be sized to meet the load requirements of the driven equipment throughout the normal operating range of the pump. These load requirements must consider torque, thrust and inertia and must also provide for any additional requirements of accessory equipment. The type of driver will be specified by the purchaser.

2.3.4.1 Electric motors

Electric motor drivers must be sized so that the horsepower required by the pump does not exceed the power available from the motor with due consideration for its service factor.

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

2.3.4.2 Variable speed drives

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

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

Variable speed drivers can add to the complexity of the system and may dictate that a thorough analysis of the pump/driver assembly be performed. Some special items of possible concern are the potential for a reverse torque from variable frequency drives which could unscrew the lineshaft couplings; the mass that may be added to the top of the discharge head and its effect on structural vibrations; lineshaft critical speed considerations; and torsional pulsations that could excite a rotor critical speed. For additional information, see Paragraph 2.4.8.

2.3.4.3 Gears

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

2.3.4.4 Deceleration devices

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

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

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

Flywheel applications should he carefully analyzed to match need and performance before they are installed.

2.3.4.5 Thrust bearings

Vertical pump bowl assembly units are not typically designed to balance the unequal pressure forces acting on the impellers. Although this maximizes efficiency, it also leaves a high resultant axial force which must be supported. The exact value of this axial force is dependent on pump design and size, as well as the conditions of service, such as pressure and flow. The pump manufacturer normally supplies the value of this axial thrust which must be supported by the thrust bearings in the driver, a separate thrust bearing in or above the discharge head of the pump, or a separate thrust bearing above or below the bowl assembly. Thrust bearing construction is normally of angular contact ball, spherical roller, or tilting pad hydrodynamic type.

Life expectancy, maintenance and design considerations normally dictate the type, size and location of the thrust bearing used. The pump manufacturer should be consulted for his recommendations.

2.3.4.6 Non-reverse ratchets

As described in Paragraph 2.4.6.4 of the Installation section, vertical pumps are often subject to flow reversals and the resultant reverse rotation of the pump. Reverse rotation may impose high mechanical stresses on the rotating parts of both the pump and the prime mover. One common method to reduce the starting torque is to provide a non-reverse ratchet in the prime mover. However, non-reverse ratchets will not relieve the stress due to the flow reversal. Care should be taken in properly sizing the prime mover and its associated non-reverse ratchet.

2.3.4.7 Pump-to-driver shafting

Vertical pump drivers are either of hollow shaft or solid shaft construction. In the hollow shaft configuration, the top pump shaft, called the head shaft, extends through the driver shaft, which is hollow, and is normally keyed at the top and held axially with the adjusting nut. This permits the shaft to be adjusted to compensate for the tolerance stack-up of the pump rotor and casing components and to provide the desired axial running clearance for the impellers. This clearance is normally specified by the manufacturer and is determined by mechanical and efficiency considerations, as well as thermal and pressure elongation expectations of the column and shafting. Further, a bottom steady bushing option is normally offered with hollow shaft drivers to provide added shaft support. This bottom bushing is often recommended with two piece head shafts, with long one piece head shafts, or to solve head shaft vibration problems.

Solid shaft drivers are connected to the pump through either a rigid or a flexible coupling. Rigid couplings must transmit torsional and axial loads, maintain shaft alignment and permit the same axial adjustment of the shaft as detailed above for a hollow shaft driver. Flexible couplings are normally only used with solid shaft drivers when a separate thrust bearing is provided in or above the pump discharge head and below the flexible coupling. The flexible coupling then only transmits the torque, with the separate thrust bearing providing the axial and radial shaft support and allowing for the required axial adjustment of the rotating element.

2.3.5 Intake system design

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

Calculated low average velocity is not the sole basis for judging the suitability of an intake structure. High velocities in currents and swirls may be present in structures which have very low average velocity.

The ideal approach is a straight channel coming directly to the pump or suction pipe. Turns and obstructions are detrimental, since they may cause eddy currents and tend to initiate deep-cored vortices.

Water should not flow past one pump suction bell, suction pipe, or other intake to reach the next. If the intakes must be placed in line with the flow, it may prove necessary to construct an open front cell around each intake or to put turning vanes under the intake to deflect the water upward. Streamlining should be used to reduce alternating vortices in the wake of an intake or other obstructions in the stream flow.

The amount of submergence available is only one factor affecting vortex-free operation. It is possible to have adequate submergence and still have submerged vortices that may have an adverse effect on pump operation. Successful, vortex-free operation will depend greatly on the approach upstream of the sump.

While specific intake structure design is beyond the scope of the pump manufacturer's responsibility, the pump manufacturer may comment on the preliminary layout given.

The sump dimensional suggestions shown in this Standard are valid for both dry pit pump suction pipes and all types of wet pit pumps. Care should be exercised in order not to use a portion of the suggested dimensions with the remaining dimensions altered in such a fashion as to render the sump design inadequate.

Complete analysis of intake structures can only be accurately accomplished by scale model tests. Model testing is especially recommended for larger pumping units.

These guidelines apply when handling clear liquids. Requirements of sumps for solids-handling pumps are discussed in Paragraph 2.3.5.4.

2.3.5.1 General data information

Subject to the qualifications of the foregoing statements, Figures 2.26 through 2.33 have been constructed for single and multiple intake arrangements to provide guidelines for basic sump dimensions.

Since these values are composite averages for many pump types and cover the entire range of specific speeds, they are not absolute values but typical values subject to variations. For pumps operating at capacities below approximately 3,000 gpm, refer to sump or wet pit designs (small pump), Paragraph 2.3.5.6.

All of the dimensions in Figures 2.26 through 2.29 are based on the rated capacity of the pump. If operation at an increased capacity is to be undertaken for extended periods of time, the maximum capacity should be used for obtaining sump dimensions.

The dimension C in Figures 2.26 and 2.28 is an average, based on analysis of many pumps. Its final value should be specified by the pump manufacturer.

Dimension B in Figures 2.26 through 2.29 is a suggested maximum dimension which may be less depending on actual suction bell or bowl diameters. The edge of the pump or suction pipe bell should be close to the back wall of the sump. If the position of the back wall is determined structurally, dimension B may become excessive and a false back wall should be installed.

Dimension W in Figures 2.27 and 2.29 is a minimum for the sump width for a single pump installation.

Dimension S in Figures 2.26 and 2.28 is a minimum value based on the normal low water level at the pump or suction pipe bell, taking into consideration friction losses through the inlet screen and approach channel. Note that this dimension represents submergence at the intake, or the physical height of the water level above the intake relating to the prevention of eddy formations and vortexing. This submergence value should not be confused with the submergence required to provide adequate NPSHA, which must be considered separately. NPSH considerations may require the water level to be greater than that obtained by Figure 2.28. It should be noted that in checking for adequate NPSHA, all NPSH values are referred to the datum elevation as defined in Paragraph 2.2.3.4.

The channel floor should be level for at least a distance Y (see Figures 2.26 through 2.29) upstream before any slope begins. The screen or gate widths should not be substantially less than W, and heights should not be less than the maximum anticipated water level to avoid overflow. Depending on the approach conditions before the sump, it may be necessary to construct straightening vanes in the approach channel, increase dimension A and/or conduct an intake model test to work out some other combination of these factors.

Dimension W is the width of an individual pump cell or the center-to-center distance of two pumps if no dividing wall is used.

On multiple intake installations, the recommended dimensions in Figures 2.26 and 2.27 apply as noted above, and the following additional factors should be considered.

As shown in Figure 2.29 (A), low velocity and straight in-line flow to all units simultaneously is a primary recommendation. Velocities in the sump should be approximately one foot per second, but velocities of two feet per second may prove satisfactory. This is particularly true when the design is based on a model study. Not recommended would be an abrupt change in the size of the inlet pipe to the sump or the inlet from one side introducing eddying.

In many cases, as shown in Figure 2.29 (B), pumps operate satisfactorily without separating walls. If walls must be used for structural purposes or some pumps operate intermittently, then the walls should extend from the rear wall approximately ten times the C dimension given in Figure 2.26.

If walls are used, increase dimension W by the thickness of the wall for correct centerline spacing and use round or ogive ends of walls. Not recommended is the placement of a number of pumps or suction pipes around the sides of a sump with or without dividing walls.

Abrupt changes in size, as shown in Figure 2.29 (C), from inlet pipe or channel to the sump are not desirable. Connection of a pipe to a sump is best accomplished using a gradually increasing taper section. The angle should be as small as possible, preferably not more than 10 degrees. With this arrangement, sump velocities less than one foot per second are desirable.

Specifically not recommended is a pipe directly connected to a sump with suction intakes close to the sump inlet, since this results in an abrupt change in the flow direction. Centering pumps or suction pipes in the sump leaves large vortex areas behind the intake which will cause operational trouble.

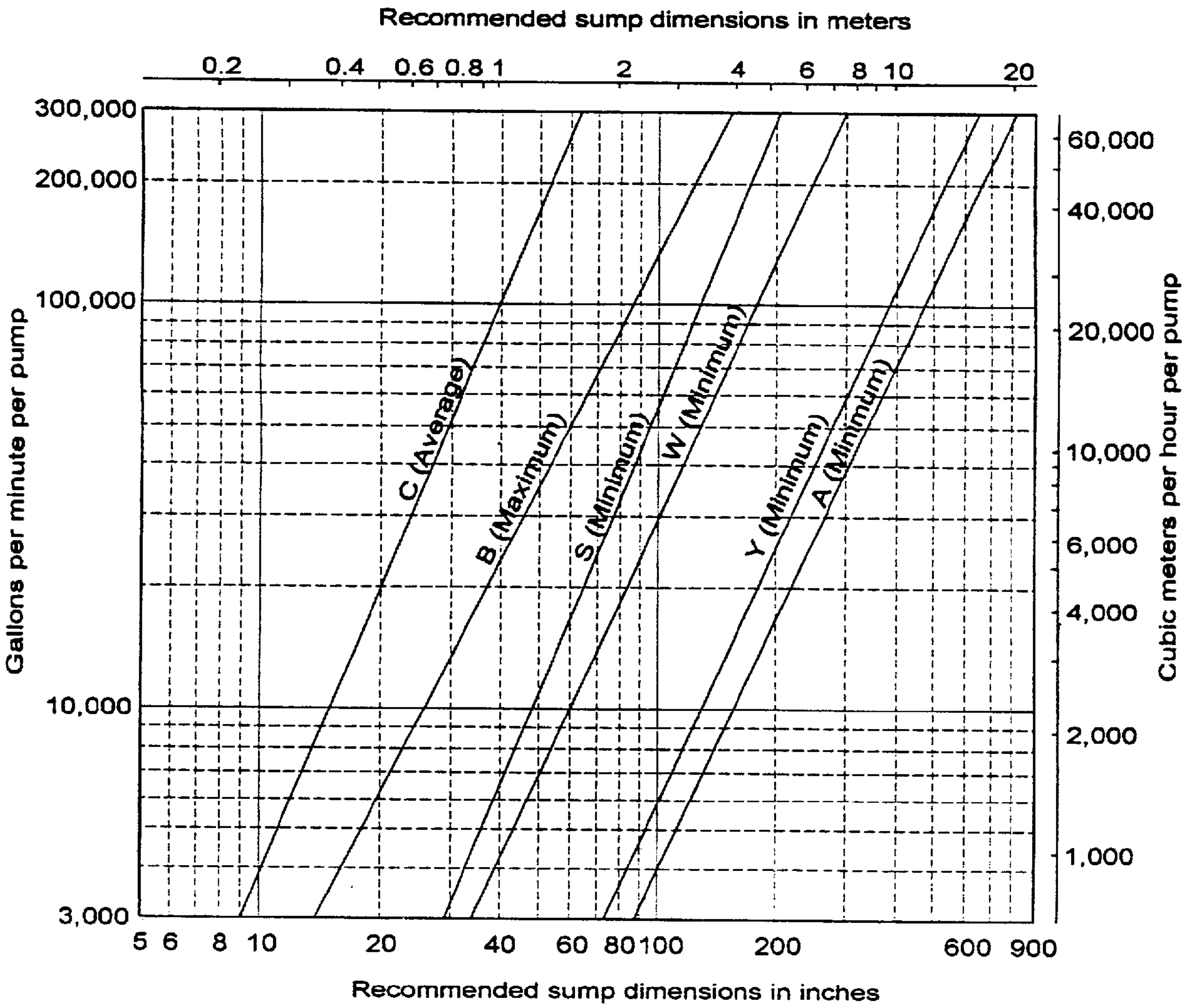


Figure 2.26 — Sump dimensions versus flow

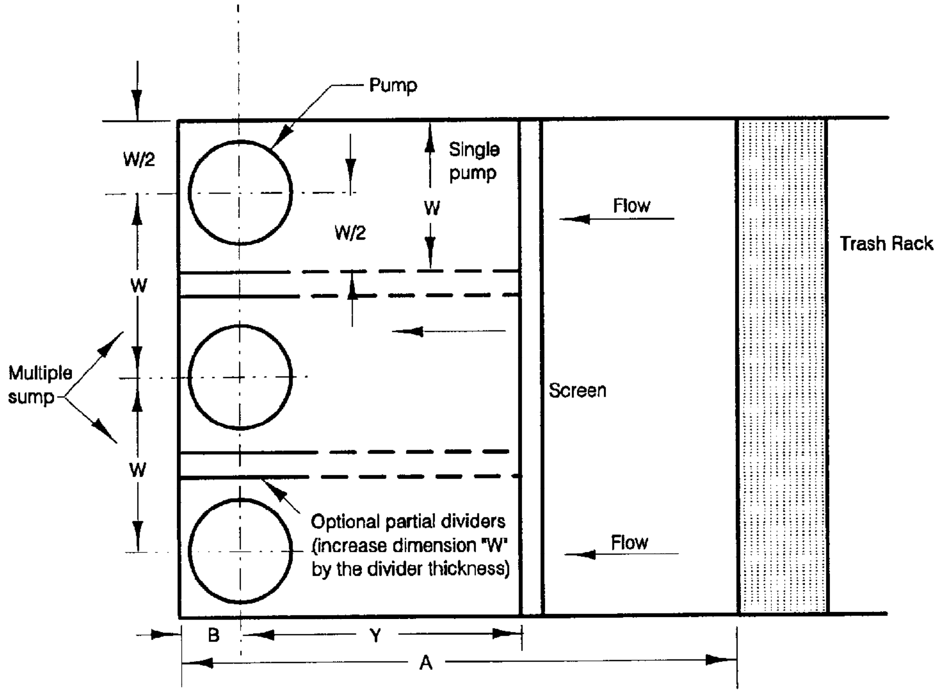


Figure 2.27 — Sump dimensions, plan view, wet pit type pumps

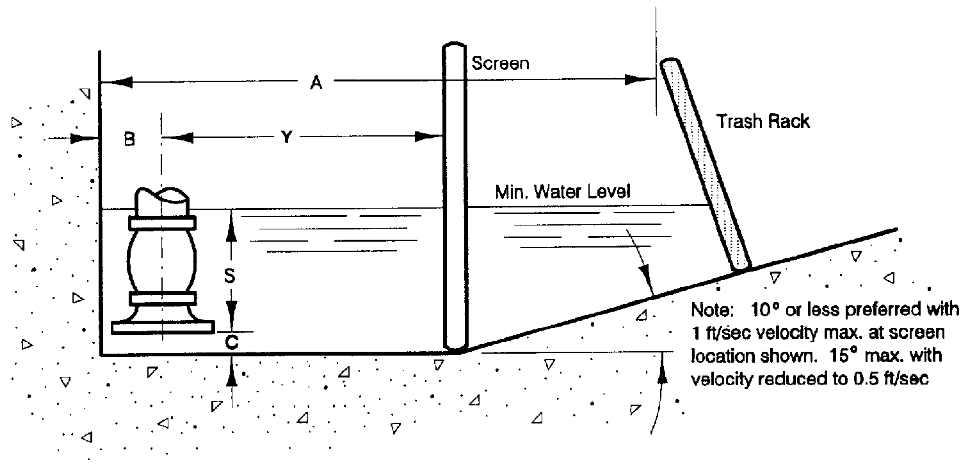
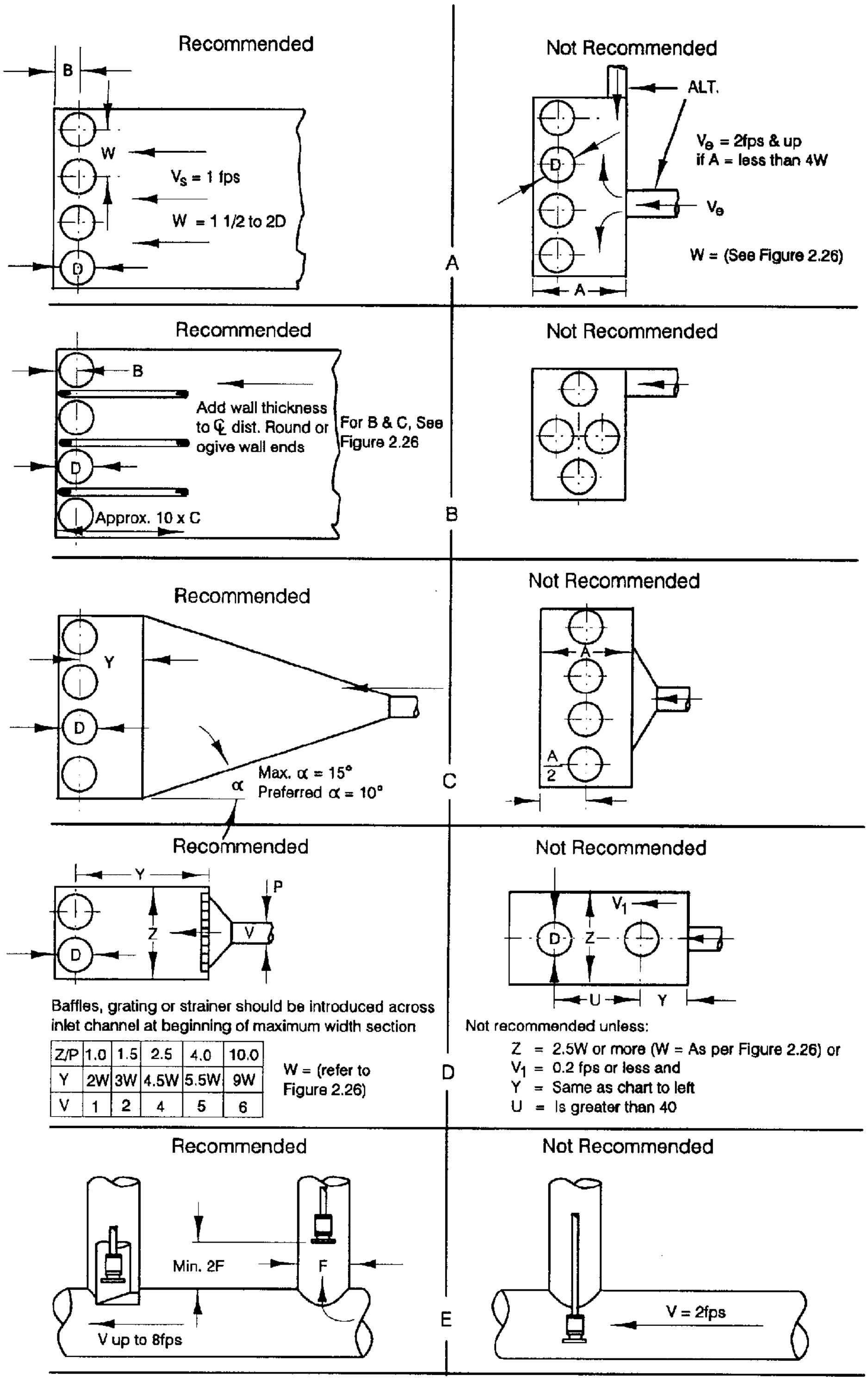


Figure 2.28 — Sump dimensions, elevation view, wet pit type pumps



Z/P	1.0	1.5	2.5	4.0	10.0
Y	2W	3W	4.5W	5.5W	9W
V	1	2	4	5	6

Figure 2.29 — Multiple pump installations

If the sump velocity, as shown in Figure 2.29 (D), can be kept low (approximately one foot per second), an abrupt change from inlet pipe to sump can be accommodated if the sump length equals or exceeds the values shown. As ratio Z/P increases, the inlet velocity at P may be increased up to an allowed maximum of eight feet per second at Z/P= 10. Intakes "in line" are not recommended unless the ratio of sump to intake size is quite large and intakes are separated by a substantial margin longitudinally. A sump can generally be constructed at less cost by using a recommended design.

As shown in Figure 2.29 (E), it is sometimes desirable to install pumps in tunnels or pipe lines. A drop pipe or false well to house the unit with a vaned inlet elbow facing upstream is satisfactory in flows up to eight feet per second. Without inlet elbow, the suction bell should be positioned at least two pipe (vertical) diameters above the top of the tunnel. The unit should not be suspended in the tunnel flow, unless the tunnel velocity is less than two feet per second. There must be no air along the top of the tunnel, and the pump vendor's minimum submergence must be provided.

2.3.5.2 Correction of existing equipment

Vortexing in pump sumps is harmful to pumps and intake structures. While this phenomenon can be avoided in a new design, for existing structures where problems are already apparent, corrective measures are necessary. Possible modifications to correct sump problems are shown in Figure 2.30.

It is suggested that intake model tests be performed to prove the effectiveness of the changes and ensure that unnecessary expenses are not incurred.

Disperse concentrated velocity by changing the direction and/or velocity of inflow by suitable baffling. The baffle may be floor-mounted, extending above the minimum flow level, or may be mounted from above or from side walls, extending close to the floor (see Figure 2.30 (A)).

Change the location of pumps or suction pipes in relation to the inflow. A suitable baffle may be necessary in front of the inlet. See Figure 2.30(B).

A cone may be added to reduce the possibility of submerged vortex formation. See Figure 2.30 (C).

A triangular splitter may be added to sump floor and back wall to minimize vortex formation at these points. See Figure 2.30 (D).

Eliminate sharp corners at gates, screens, etc., by filling in for smooth flow contour (fairing). See Figure 2.30 (E).

Reduce the velocity of flow and eliminate vortexing by adding bell extension suction plate and splitter to pump bell. Splitter must be in line with the flow. See Figure 2.31 (G).

Floating rafts around the pump column or suction pipe can be used as a temporary solution to prevent surface vortices. See Figure 2.31 (H).

Reduce the clearance between the pump or suction pipe and back wall. This will improve velocity pattern and reduce the possibility of vortex formation. See Figure 2.31 (I).

Change inlet flow direction uniformly by means of parallel turning vanes. See Figure 2.31 (J). In general:

- Keep inlet velocity to the sump below two feet per second. Keep velocity in sump below one foot per second. Avoid changing direction of flow from inlet to pump or suction pipe, or change direction gradually and smoothly, guiding flow.
- In addition to the above, other modifications such as horizontal beam with bottom flange submerged to the control water surface, strainer attached to the suction bell (for submerged vortices), and other flow straighteners may be used to correct existing sumps.

The above alterations, singly or in combination, may help create a better flow pattern.

2.3.5.3 Model tests of intake structures

Often an accurate analysis of a proposed intake design can only be made by conducting scale model tests.

Model tests are generally best conducted in the early stages of station design in order to accommodate any recommended changes with a minimum of delays and extra expense. Agreement between the purchaser/user and the model test vendor before the test begins regarding what constitutes unacceptable flow characteristics will reduce later misunderstandings. Figure 2.32 depicts a vortex classification system that may be

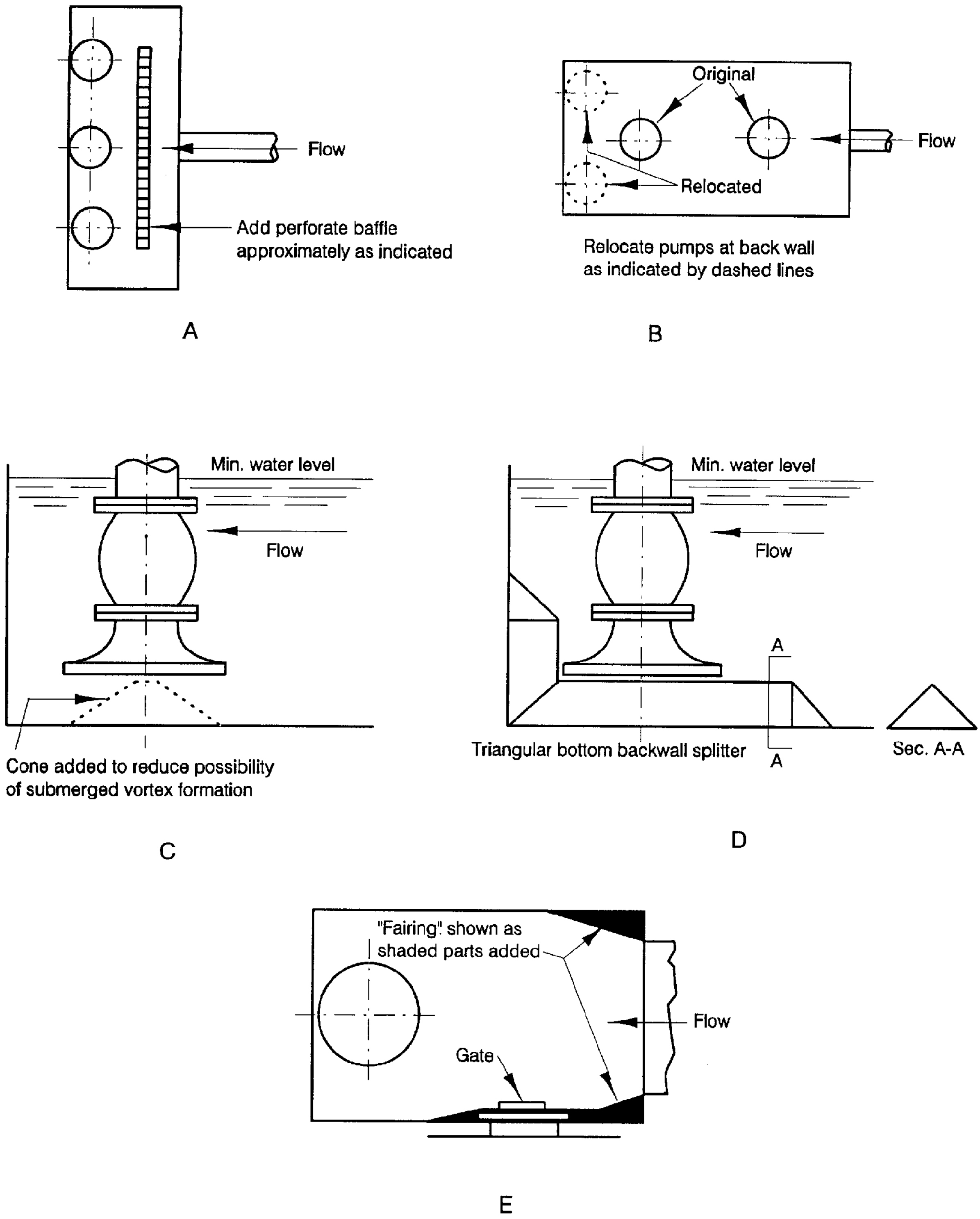


Figure 2.30 — Correction of existing sumps

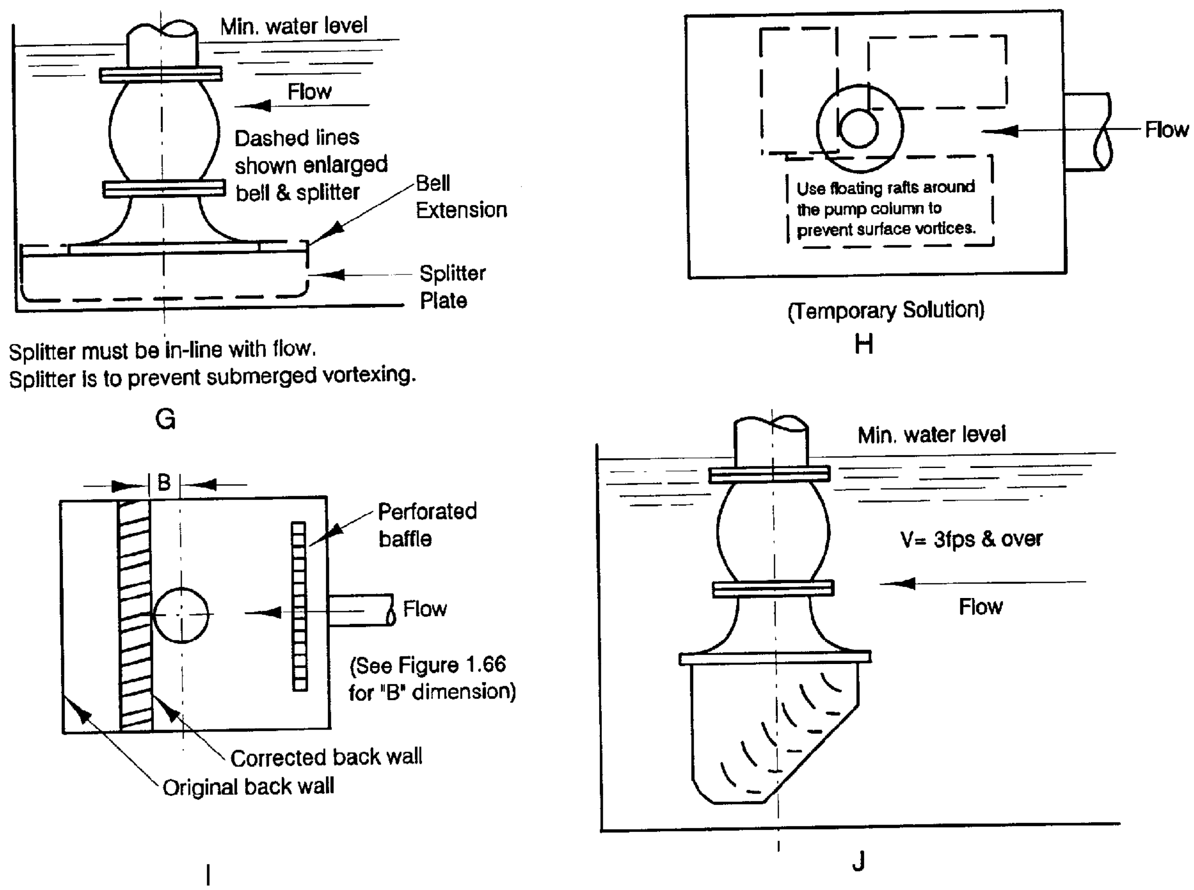


Figure 2.31 — Correction of existing sumps

Vortex class

1) Surface or subsurface swirl

2) Surface dimple, diffuse swirling dye core

3) Well-defined surface dimple, well organized and defined dye core

4) Air or vapor bubbles in core

5) Solid air or vapor core

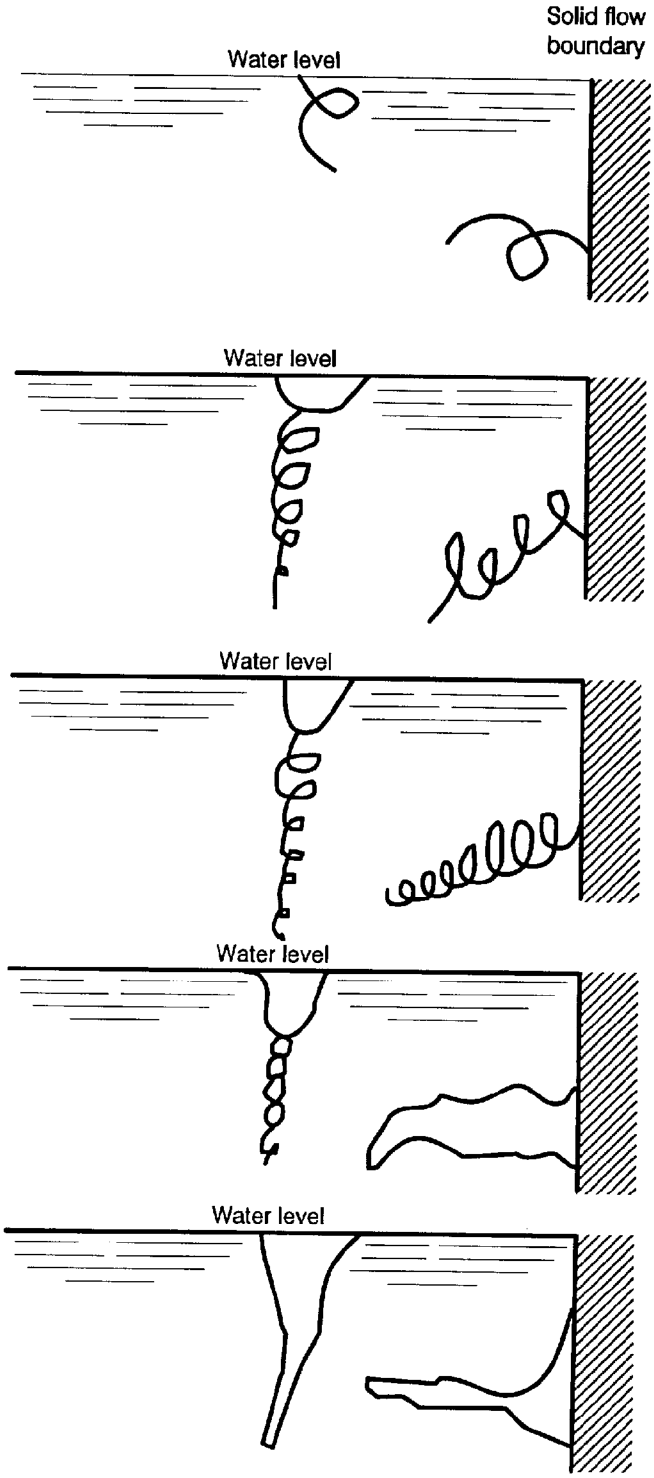


Figure 2.32 — Vortex classification system

used in describing acceptable and unacceptable vortex behavior.

The engineers responsible for the design of the pumping station should consult with the pump manufacturer to establish one or more intake arrangements. A model test can then be conducted by an independent laboratory or by the pump manufacturer. The model test may show modifications of structure or baffling arrangements to be necessary. Model tests may also show how considerable savings can be made in the intake structure.

The model should be extensive enough to include all parts of the channel likely to affect the flow near the pump or suction pipe, including screens and gates. The model should be made so that careful observation of vortices is possible. Plastic or glass walls and adequate lighting are necessities. In addition to surface vortices, submerged vortices may be present which may affect pump noise and performance. These vortices do not pierce the surface but run from the pump to the structure. They can be difficult to observe unless dye or other means are used to accentuate the swirl; therefore, the model should be constructed to accommodate these requirements.

2.3.5.3.1 Dynamic similarity guidelines

Comparable flow in the model is generally considered to be obtained at equal Froude numbers:

$$v_m = v_p R^{1/2}$$

Where:

v_m = Flow velocity of water in the model;

v_p = Flow velocity of water in the prototype;

R = Linear scale ratio of model to prototype;

$$R = \frac{L_m}{L_p}$$

Where:

L_m = Any linear dimension of the model;

L_p = Corresponding prototype dimension.

Some investigators have found better agreement between model and prototype when velocities are equal than when velocities are in accord with the Froude number. In the present state of the art, it is recommended that this entire range of velocities be considered during the model test.

2.3.5.3.2 Intake model test parameters

In order to produce practical solutions, the following parameters should be agreed upon by the purchaser/user and the vendor of the model tests before the tests begin:

1) Model scale;

2) Extent of the model (i.e. upsteam approach geometry will affect the flow into the sump, any surface imperfections of hydraulic boundaries, etc.);

3) Test velocities (Froude, prototype, or other);

4) Flow rates to be simulated;

5) Water levels to be simulated;

6) Pump operation combinations.

2.3.5.3.3 Intake model test evaluation criteria

Ideally, the flow in the approach to the sump and in the sump itself should be:

1) Uniform (the velocity vector is identical in magnitude and direction at every point across the section considered);

2) Irrotational (the fluid within a region considered has no rotation around any axis);

3) Steady (the velocity vector does not change in magnitude or direction with time);

4) Single Phase (there is no entrained air).

The tests should be evaluated with regard to these parameters. In practice, some departures from ideal conditions exist, even in well-designed intakes.

2.3.5.4 Sump design for solids-handling pumps

Sumps for liquids containing solids require particular attention and, when possible, should be based on previously tested and proven designs. The velocities in these sumps must be kept higher to prevent solids separation and settlement. Three feet per second (1 m/s) is frequently used as minimum approach velocity. The sidewalls of the sump should be shaped to avoid settlement of solids in the corners. All other general rules for good intake design should also be followed.

2.3.5.5 Suction tanks

In many process installations, a suction line may be taken off the side or bottom of a process or suction tank. General rules for sound intake design apply to suction tanks as well, and particularly adequate submergence must be provided to avoid vortexing.

Typically, one foot submergence for each foot per second of velocity at the suction pipe inlet is recommended, with a suggested maximum inlet velocity of six feet per second (2 m/s). Bellmouth or rounded inlets should be used to reduce inlet velocities. If the recommended submergence cannot be obtained, the inlet pipe diameter should be increased or vortex breakers should be installed. See Figure 2.33 for recommended breaker designs.

2.3.5.6 Sump or wet pit design (small pumps)

The same general principles as previously outlined apply to the design of sumps for small pumps (less than 3,000 gpm (680 m^3/h) design capacity).

However, since there is a large variety of geometric configurations for these small units, limiting dimensions as shown in Figure 2.26 cannot be sufficiently generalized and so presented.

Where specific wet pit or sump dimensions are required, the pump manufacturer's recommendations should be requested.

In addition to the general design principles outlined for single and multiple pump settings in large sump designs, the following factors are pertinent to the design of small sumps:

2.3.5.6.1 Inlet opening (pit type sumps)

The sump inlet should be below the minimum liquid level and as far away from the pump as the sump geometry will permit. The free discharge of liquid above the surface of the sump at or near the pump or suction pipe can cause entrained air to enter the pump. The influent should not impinge or jet directly into the pump inlet or suction pipe or enter the sump in such a way as to cause rotation of the liquid in the sump. Where required, a distribution nozzle can be used to prevent rotation.

2.3.5.6.2 Wet sump volume

The wetted sump volume should equal or exceed the maximum capacity to be pumped in two minutes. If units operate on float switch control, the sump should be sized to result in no more than three or four starts per hour per sump. These guidelines generally insure sumps of adequate size to dissipate the inflow turbulence and to assure reasonable life of the equipment.

2.3.5.6.3 Minimum liquid level for submersible pumps

In addition to liquid level requirements to satisfy NPSH and vortex avoidance, submersibles may require continuous immersion of some portion of the unit to provide adequate cooling of the motor. The pump manufacturer's recommended dimensions should be used.

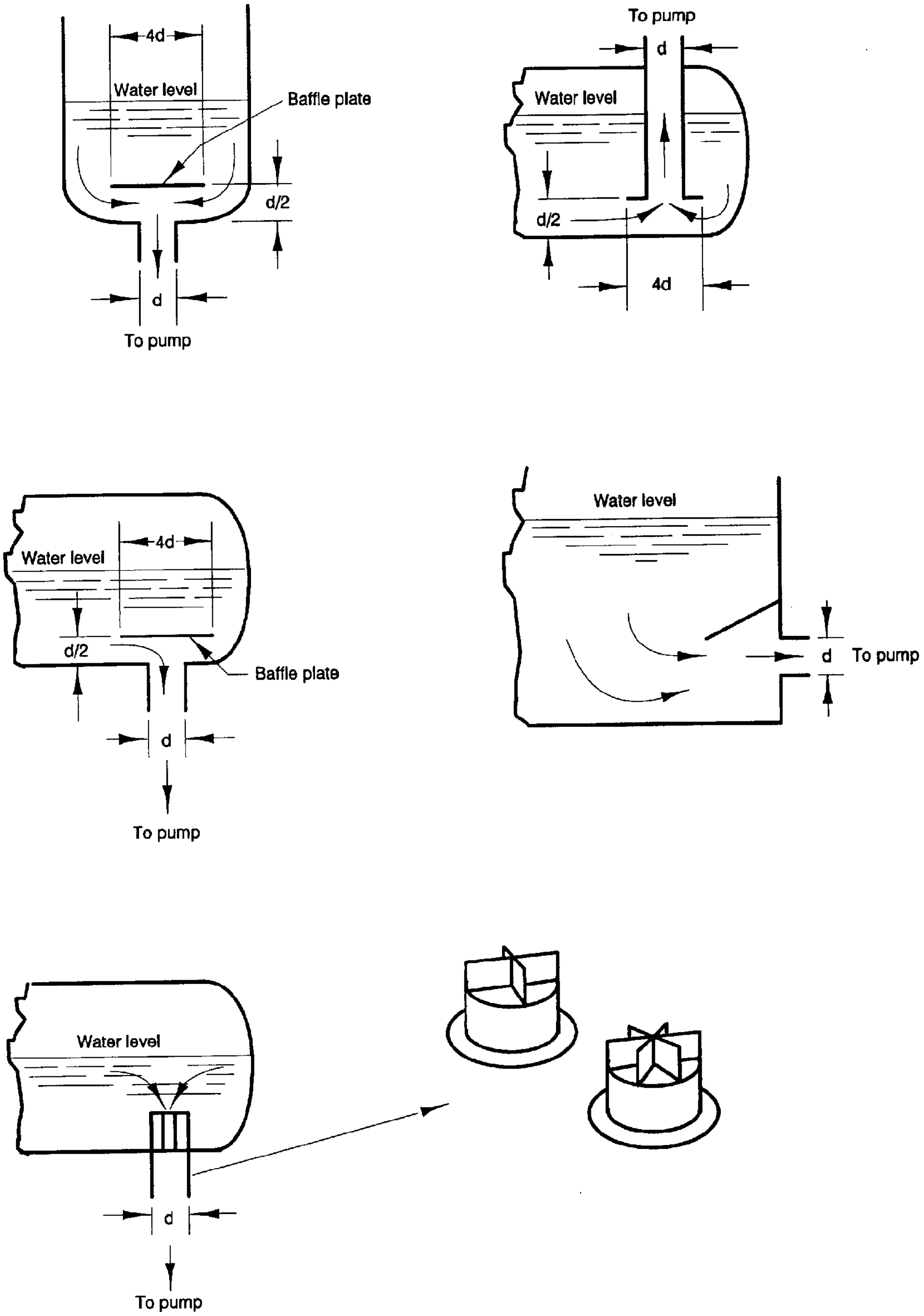


Figure 2.33 — Vortex breakers in typical suction tanks

2.4 Installation, operation and maintenance

Vertical pumps, when properly installed and operated, and when given reasonable care and maintenance, will operate satisfactorily for a long period of time. The following paragraphs outline the general principles that must be considered to insure trouble-free pump operation.

Vertical pumps are built in a wide variety of designs and for many different services. The manufacturer's instruction book should be studied carefully and followed, as there may be specific requirements for a particular machine or application which cannot be covered in a general discussion.

2.4.1 Pre-installation instructions

2.4.1.1 Unloading/inspecting equipment received

Immediately upon receipt of the pump equipment, check carefully to see that all items have been received and are in undamaged condition. Report any shortage or damage to the transport company handling the shipment and the equipment manufacturer, noting the extent of damage or shortage on the freight bill and bill of lading. This should be done at once. Particular care and close adherence to the manufacturer's recommendations are required when unloading long, slender components such as shafting. Improper placements of slings or chains can result in deformation or other serious damage. Do not leave the unit exposed to weather or construction hazards, which may cause damage to the equipment.

2.4.1.2 Storing equipment at site

2.4.1.2.1 Short-term

The pump and equipment, as shipped, have adequate protection for short-term storage in a covered, dry and ventilated location at the job site prior to installation.

2.4.1.2.2 Long-term

If the equipment will be subject to extended storage conditions prior to installation, then the manufacturer must be advised about storage duration, so that special protection can be provided for the equipment. Periodic rotation of the pump and driver shaft is recommended during long-term storage.

2.4.1.3 Handling equipment and tools for installation

For typical installations, the following equipment must be available at the job site when installing or removing the pump: Suitable overhead lifting equipment of adequate capacity to lift the driver, the entire pump (without driver) or the heaviest subassembly of the pump. Adequate headroom must be provided to accommodate the longest section of the pump to be handled, including rigging.

Properly sized slings, chains and shackles for attaching to the equipment lifting lugs. Eye bolts are required for handling pump sections when lifting lugs are not provided.

I-beams for supporting pump subassemblies at the foundation when it is necessary to install the pump in sections. Common millwright's tools are used for this type of work, including a machinist level to insure proper leveling of the foundation plate.

2.4.1.4 Manufacturer's instructions

The Installation and Service Manual and/or special instructions included in the shipment should be read thoroughly before installing or operating the equipment. All instructions should be retained for reference regarding maintenance and operation.

2.4.1.5 Use of manufacturer's service personnel

It is recommended that the services of a manufacturer's erecting engineer be employed for supervising installation and start-up of the pumping equipment, when such equipment is custom-engineered or of a costly, high-precision type. This is to assure that the machinery is properly installed. The purchaser then also has the opportunity to review and see implemented factory-recommended instructions.

2.4.1.6 Site preparation; protection against elements/environment

A clean, drained area must be provided next to the point of installation, of adequate size for placing the pump components and driver in the sequence in which they will be installed. Protective covers should be left on all pump openings until the time of actual installation, to prevent dirt and foreign objects from entering the pump. Protective coatings should likewise be left on machined surfaces to prevent rusting. For outdoor installations, the components should be covered with rainproof tarps during the period of installation for protection against the elements. This is particularly important during freezing conditions, to prevent water from collecting in pump cavities and perhaps causing freezing damage.

2.4.1.7 Foundation requirements (forces and mass requirements)

The mass of the foundation must be sufficient, preferably five times that of the rotating element of the pumping equipment, to form a permanent, rigid support for the base plate. This is equally important whether the pump is installed over a pit or over a well. Base plate and foundation bolt sizing is critical, particularly on high-pressure pumps, to adequately restrain reaction forces such as from directional flow change, system transients and sudden valve closure. Foundation bolts should be embedded in the concrete, located by a drawing or template. A pipe sleeve larger than the bolt should be used to allow movement for final positioning of the bolts. (See Figure 2.34).

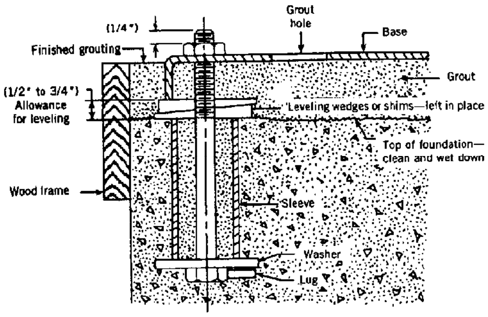


Figure 2.34 — Typical foundation bolt design

2.4.1.8 Access for maintenance and repair

All pumps require regular maintenance. It is therefore important to locate pump discharge piping (and suction piping when applicable), as well as auxiliary equipment, control and starting panels in such a manner that adequate access is provided for maintenance. Adequate floor space and working room must also be provided for repair, including parts placement.

2.4.2 Installation

2.4.2.1 Checking wells

When vertical pumps, either of the lineshaft or submersible type, are to be installed in wells, consideration must be given to the well before application and installation (See Figure 2.35).

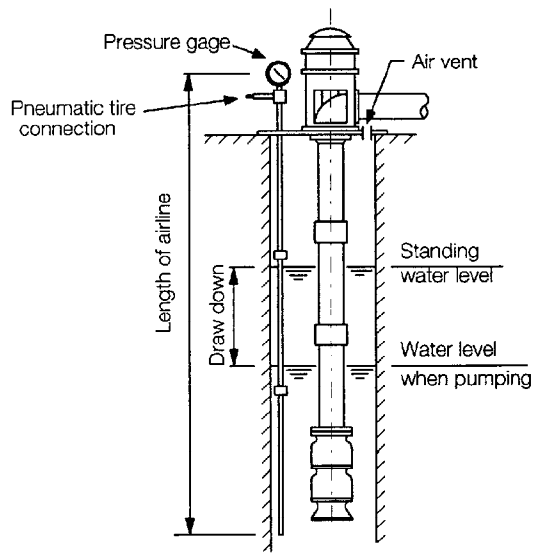


Figure 2.35 — Typical deep well type installation

Installing a unit in a crooked well may bind and distort the pump column or pump-motor assembly with potential resulting malfunction. Well straightness should be within one inch per hundred feet and without double bend. If straightness is in doubt, the well should be "gauged" prior to installation by lowering a dummy assembly, slightly longer and larger on the diameter than the actual pump or pump-motor assembly, on a cable. Gauging is also important when a stepped well casing is used, with the lower part of the well casing having a smaller inside diameter.

Wells that have not been properly constructed or developed, or which produce sand, can be detrimental to a pump. If a well is suspected of producing an excessive amount of sand, a unit other than the production pump should be used to clear the well.

2.4.2.2 Checking wet pits

Before starting the installation of pumps in an open pit, there are several checks that can be made. The configuration of the intake structure is now fixed and should be in conformance with the general guidelines provided in Paragraph 2.3.5 in the Design and Application section. Dimensional checks should be made as follows to preclude installation and servicing problems.

1) Length of pump vs. depth of sump;

2) Correct fit of anchor bolts to sole plate and of the sole plate to the pump mounting base;

3) Satisfactory angular location of anchor bolts or correct lineup of discharge head to discharge piping;

4) Proper conduit location provided for driver;

5) Sufficient head room for handling.

2.4.2.3 Locating pump

The pump should be located so that a short, direct discharge pipe, with the least number of elbows and fittings, may be used to minimize head loss from friction. If practical, it should be placed so that it will be accessible for inspection during operation. The equipment selected should be compatible with the environment. Pumps and drivers, other than submersible types, and controls should be protected against flooding.

2.4.2.4 Entrained air

Entrained air reduces pump performance, with amounts as small as 1% by volume affecting radial flow pumps and 3% to 5% of entrained air affecting axial flow pumps. Cascading water causing air entrainment must therefore be avoided. For well pumps, the perforated casing must be located below the pump suction. Return lines into sumps or tanks should terminate a minimum of two pipe diameters below the low liquid level. Undersized or partially blocked intake screens and trash racks result in similar problems, caused by excess pit velocity. Adequate provisions for cleaning rotating screens and trash racks must be made.

2.4.2.5 Pump leveling/plumbness

Vertical lineshaft and submersible pumps are automatically aligned through registered fits. However, on lineshaft pumps, it is recommended to check the alignment of the head shaft to the driver at the time the latter is mounted.

When the base plate has been correctly leveled, it should be supported on rectangular metal blocks and shims or on metal wedges having a small taper. The support pieces should be placed close to the foundation bolts (See Figure 2.36).

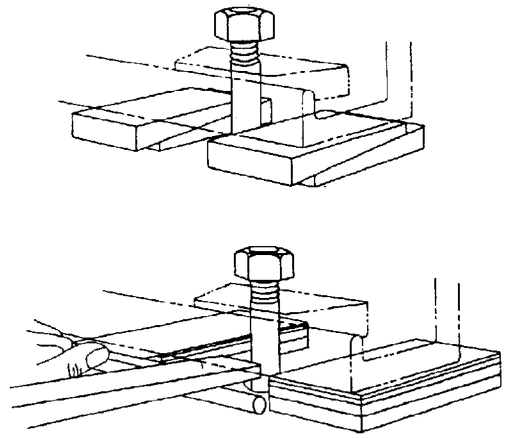


Figure 2.36 — Method of leveling

On large units, small jacks made of cap screws and nuts are often convenient. A gap of about 1 to 2 inches (25 to 50 mm) should be allowed between the base plate and the foundation for grouting.

Submersible pumps do not require special alignment at the foundation. However, solid support must be provided for the surface plate from which the unit is suspended, and a water tight seal may also be required to prevent well contamination.

2.4.2.6 Lining up pump discharge

The leveling or plumbness of a vertical lineshaft pump is established during pump installation. Slightly out of plumb wells or minor out of level foundations require the pumps to be lined up as described in Paragraph 2.4.4.2 of this Section.

The discharge pipe should be brought up to the pump discharge so that a very minimum of pipe strain is transferred to the pump.

2.4.2.7 Grouting

When the alignment is correct, the foundation bolts should be tightened evenly but not too firmly. The unit can then be grouted to the foundation. It is desirable to grout the leveling pieces, shims or wedges in place. Foundation bolts should not be fully tightened until the grout is hardened, usually about 48 hours after pouring.

2.4.3 Suction and discharge piping

2.4.3.1 Pipe supports/anchors/joints

Suction and discharge piping must be anchored, supported and restrained near the pump to avoid application of forces and moments in excess of those permitted by the pump manufacturer.

In calculating forces and moments, the weights of the pipe, contained fluid and insulation, as well as thermal expansion and contraction, must be considered.

If an expansion joint is installed in the piping between the pump and the nearest anchor in the piping, a force equal to the area of the expansion joint times the pressure in the pipe will be transmitted to the pump. Pipe couplings which are not axially rigid have the same effect. This force may be larger than can be safely absorbed by the pump or its support system.

It is therefore recommended that a pipe anchor be installed between an expansion joint and the pump to absorb the axial force.

When proper anchoring cannot be provided, adequate tie rods must be provided and properly adjusted to protect the pump and the expansion joint. Cast iron and nonmetallic pump flanges are usually made with flat faces. To avoid breaking the flange when tightening the bolting, mating pipe flanges should also have flat faces, and a full-face gasket should be used.

2.4.3.2 Suction piping requirements

A vertical pump in a suction barrel performs properly only if it is supplied with a steady flow of liquid arriving at the pump suction flange with sufficient pressure to provide adequate NPSH to the pump and with a uniform velocity profile.

Failure of the suction piping to deliver the liquid to the pump in this condition can lead to noisy operation, swirling of liquid around the suspended pump assembly, premature bearing failure, and cavitation damage to the impeller and inlet portions of the casing.

For pumps operating with suction pressure below atmospheric pressure, or handling fluids near their vapor pressure, the suction line should slope constantly upwards toward the pump to avoid trapping vapor using eccentric reducerswhere necessary (see Figure 2.37).

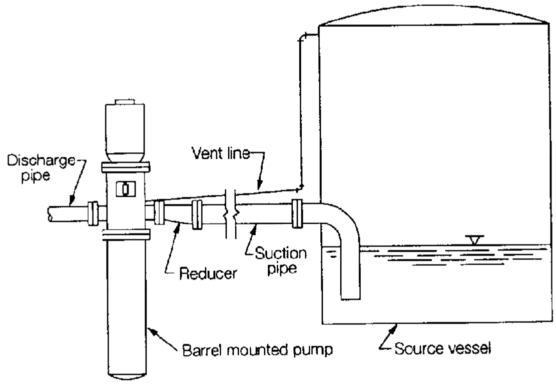


Figure 2.37 — Eccentric reducer

In systems where the suction line is not always kept full of liquid, there is a possibility that a large slug of air or vapor may be swept into the pump during a restart, causing a partial or complete loss of pump prime. Any high point in a suction line will accumulate gas with similar results. Any valves in the suction line should be installed with their shafts horizontal, to avoid air pockets in the valve body.

When it is required to prime the pump before start-up, then the priming connections should be at the high point of the pump suction chamber. A permanent vent line back to the suction source at this point may also be desired for pumps operating in a closed system.

2.4.3.3 Pipe reducers

Reducers are installed just ahead of the pump suction when the pipe is larger than the pump nozzle. Reducers used at the pump suction should be of the conical type and sufficiently long to prevent fluid turbulence. Contour type reducers are not recommended.

With the liquid source below the pump, the reducer must be eccentric and installed with the level side up (see Figure 2.37).

Eccentric or concentric reducers may be used when the liquid source is above the pump and the suction piping is sloping upward towards the source.

2.4.3.4 Suction valves

Check valves used to prevent backflow should not be used in the suction line. They are sometimes used in series-parallel connections to reduce the number of valves which must be operated when changing from one type of operation to the other.

Block valves may be installed in order to be able to isolate the pump for maintenance.

Foot valves are specially designed check valves sometimes used at the inlet to bowl assemblies for well pumps to keep the column water filled and to prevent backspin and well disturbance from rapidly draining water.

2.4.3.5 Strainers

To keep unwanted solids out of the pump, strainers may be installed at the suction bell or in the pump suction piping. The strainer itself usually introduces only a moderate pressure drop, but as debris accumulates, the pressure drop will increase. It is therefore recommended that pipe-mounted strainers be installed with upstream and downstream pressure taps and that the pressure drop be monitored.

Suction bell strainers typically clear themselves by backflow in the pump column when the unit is stopped. For large pumps, trash racks and screens are typically part of the intake structure.

2.4.3.6 Elbow at pump suction

When a straight run of pipe at the pump suction cannot be provided, certain arrangements of fittings must be avoided for vertical pumps installed in suction barrels. When liquid flows through an elbow, or makes a turn through a tee, the exit velocity will be strongly non-uniform. Elbows with a plane perpendicular to the pump can should therefore not be used, since a strong vortex motion can be set up in the fluid in the pump barrel. This could lead to a swirling motion in the suspended pump and result in bearing failure, noisy operation and cavitation damage in the first stage of the pump assembly. Splitters inside the pump barrel can be used to break up the fluid swirl.

2.4.3.7 Suction tanks

In many process applications, a suction line may be taken off the side or bottom of a process or storage vessel. When this is done, it is necessary to ensure that the submergence level over the inlet to the suction pipe is adequate to prevent vortexing. Figure 2.26 in the Design and Application section indicates reasonable minimum values of submergence over the inlet as a function of capacity. If operating levels of liquid in the vessel cannot provide the required submergence at the planned line velocities, then the size of the inlet must be increased as necessary to reduce the velocity to the point where the submergence is adequate.

2.4.3.8 Discharge valves

A check valve and an isolation valve should be installed in the discharge line. The check valve serves to protect the pump from reverse flow and excessive back pressure. The isolation valve is used in priming, starting and when shutting down the pump. Except on axial flow and mixed flow pumps, it is advisable to close the isolation valve before stopping or starting the pump. If increasers are used on the discharge side of the pump to increase the size of piping, they should be placed between the check valve and the isolation valve. If expansion joints are used, they should be placed between the check valve and pump. (See Figure 2.38)

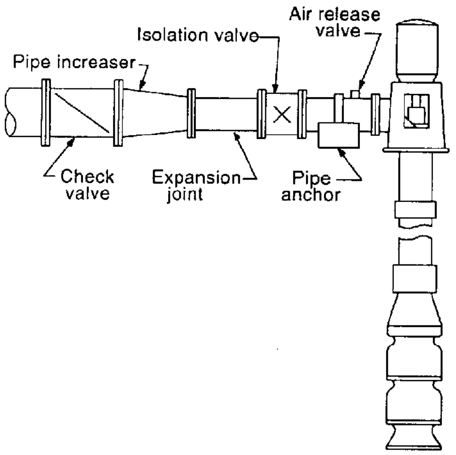


Figure 2.38 — Discharge valve expansion joint

2.4.3.9 Air release valves

For medium and large size vertical wet pit pumps pumping into a pressurized system, an automatic air and vacuum release valve is recommended.

The valve should be located on the pump discharge nozzle or between the pump discharge nozzle and the discharge valve or check valve, whichever is closest.

The release valve prevents a large volume of air from being compressed, and then setting up a severe shock wave when suddenly released, with potential for serious equipment damage. The air release valve also prevents undesirable air from entering the pressurized system.

2.4.3.10 Siphons

When a siphon is used in the pump discharge line, for the purpose of reducing the head requirement for applications such as pumping over a levee, additional equipment requirements are imposed for the system to function satisfactorily.

To clear the siphon of air and make it operational, either a vacuum pump or an air ejector must be provided, or the pump and driver must be suitable for handling the higher head with adequate flow until the siphon is cleared. For high specific speed pumps, this may result in a significant increase in required brake horsepower. Additionally, if the height of the siphon above the discharge water level is substantial, then the flow from the pump at the increased head requirement may not be sufficient to clear the siphon, and a vacuum pump assist is required.

A siphon breaker must be mounted at the high point of the siphon to prevent backflow when the pump is stopped.

2.4.4 Mounting and alignment of drivers

2.4.4.1 Mounting and alignment of vertical solid shaft drivers

Before mounting the driver on the discharge head/driver stand, check the register fit, if furnished, and the mounting face on the driver for acceptable tolerance on runout and squareness respectively, using a dial indicator mounted on the driver shaft. Next, check the squareness of the face of the driver coupling half, mounted on the shaft with a tight fit and seated against a split ring, using a dial indicator on a firm base.

With the driver bolted to the discharge head, mount a dial indicator on the driver shaft above the coupling half and sweep the bore of the stuffing-box. If excess runout exists, some adjustment can be made at the driver mounting fit and the stuffing-box mounting fit. Before installing any additional coupling parts, check the driver for correct rotation, as given in the manufacturer's installation instructions. Next, mount the pump half coupling, shaft adjusting nut and coupling spacer if applicable, and raise the impeller in accordance with the manufacturer's instructions. Then secure the coupling bolts. Make a final check of the shaft runout below the pump half coupling with a dial indicator. If the runout is within acceptable tolerances, check the tightness of the driver hold-down bolts. If dowels are used to secure the driver location, then it should be noted that redoweling is required after disassembly/reassembly, since tolerance buildup in the multiple vertical joints results in alignment variation.

2.4.4.2 Mounting and alignment of vertical hollow shaft drivers

Remove the clutch or coupling from the top of the hollow shaft, and mount the driver on top of the discharge head/driver stand. For designs requiring the pump head shaft to be installed prior to mounting the driver, lower the hollow shaft driver with care over the head shaft to be sure the latter is not damaged. Check the driver for correct rotation, as given in the manufacturer's installation instructions. Install the head shaft, if not already done, and check it for centering in the hollow shaft. If off-center, check for runout in head shaft, misalignment from discharge head to driver, or out of plumbness of the suspended pump. Shims can be placed under the discharge head to center the head shaft, but shims should not be placed between the motor and the discharge head.

Install the driver coupling or clutch, and check the non-reverse ratchet for operability, if furnished. Install the coupling gib key and the adjusting nut, and raise the shaft assembly with the impeller(s) to the correct running position in accordance with the manufacturer's instructions. Secure the adjusting nut to the clutch, and double-check the driver hold-down bolts for tightness.

Most hollow shaft drivers have register fits. Further centering of these drivers is therefore normally not required, nor are dowels recommended.

2.4.5 Preparing the stuffing-box

2.4.5.1 Packed stuffing-box

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 head. Instructions are usually provided with the box of packing. If not, the following 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. When the pump is first started, the gland should be left fairly loose. Once the pump is operating normally, the gland may be tightened while the pump is running, if the leakage is excessive. A slight flow of liquid, about 60 drops per minute, from the stuffing-box is necessary to provide lubrication and cooling.

When the leakage can no longer be controlled by adjusting the gland, all rings of packing should be replaced. The addition of a single ring to restore gland adjustment is not recommended.

If a pump has been left idle for a long period of time, it is recommended that the packing be replaced.

If the liquid to be pumped is dirty, gritty, toxic or corrosive, sealing liquid should be piped to the lantern ring in the stuffing-box from a clean source of supply in order to prevent damage and/or hazardous conditions. Sealing liquid should be at a pressure sufficient to insure a small flow of clean liquid into the pump, but not so high as to require excessive tightening of the packing.

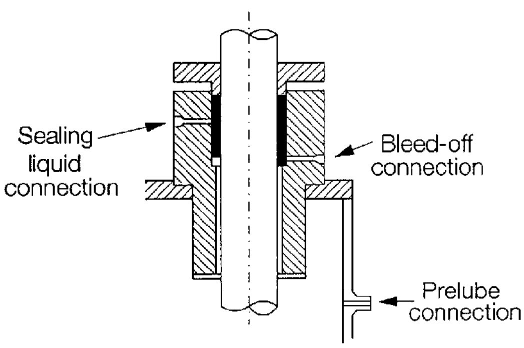


Figure 2.39 — Packed type stuffing box

When a lantern ring is provided, be sure that sufficient packing is placed below the lantern ring, so that the liquid for sealing is brought in at the lantern ring and not at the packing. The pipe supplying the sealing liquid should be fitted tightly so that no air enters. This is particularly important for vertical barrel pumps mounted in a system where a vacuum must be maintained. (See Figure 2.41).

For applications with suspended abrasives, the lantern ring should be placed at the bottom of the stuffing-box below the packing.

2.4.5.2 Mechanical seals

Pumps handling hazardous or expensive liquids, or where normal leakage from the stuffing-box is objectionable, are often furnished with mechanical seals.

Mechanical seals for vertical pumps are of two basic types, depending on whether mounting is to be external or internal. Externally mounted seals are easily adjusted for correct positioning after the impeller(s) is (are) set for correct running clearance. Mechanical seals mounted internally in the stuffing-box, unless of the cartridge type, must be mounted on a shaft sleeve and the sleeve correctly positioned and locked to the shaft after the impeller(s) is (are) lifted for proper running clearance.

Since mechanical seals are made in a wide variety of designs, the instructions for installing and operating a specific seal must be carefully studied and followed. A mechanical seal is a precision device and must be treated accordingly. Particular attention must be paid to any injection or recirculating piping required.

There are two features that can simplify change-out of worn seals. The first is the use of a spacer coupling in the head shaft of the pump. This allows removal of the seal/sleeve assembly without removing the driver. The second is use of an axially split mechanical seal. Change-out of this design does not require any disassembly of the pump.

2.4.6 Pre-lubrication

2.4.6.1 Primary and secondary drivers

Before running the driver, either separately or connected to the pump, check lubrication requirements in the manufacturer's instruction manual. Inspect and make sure that:

- Grease-lubricated bearings have been properly greased with the manufacturer's recommended grade;
- Oil-lubricated bearings on drivers and gears, as well as oil sumps on gears, have been filled to the required level with the recommended oil;
- All automatic oilers are functioning properly.

2.4.6.2 Pumps

Vertical pumps are either furnished with product-lubricated, oil-lubricated, or grease lubricated sleeve bearings. The following inspections and checks should be made for the respective bearings:

1) For product-lubricated bearings, (bearings lubricated by the pumped liquid), pre-lubrication with clean water should be provided for all pump bearings above static water level, when the distance from the mounting floor to the minimum water level exceeds 50 feet (15 meters), or as recommended by the manufacturer. The manufacturer may permit a greater distance without pre-lubricator for bearings made of self-lubricating materials;

2) For pumps with oil-lubricated bearings, it is recommended to pour one or more quarts of oil, depending on pump setting, down the shaft-enclosing tube prior to start-up. Next, make sure that the oil reservoir is filled and that the solenoid valve is functioning properly with the correct amount of oil being gravity-fed into the shaft-enclosing tube;

3) For pumps with grease-lubricated bearings, make sure the correct grade of grease is available. For manual grease injection, make sure the grease nipples are properly connected, clean and accessible. Inject per the manufacturer's instructions.

For motorized grease injection, make sure the grease lines are all securely fastened to the reservoir. Fill the reservoir with grease, energize the grease pump and check the functioning per the manufacturer's instructions. Proceed in accordance with the pump manufacturer's instruction manual.

2.4.6.3 Type of lube filtration

When required to inject water, either for flushing or lubrication of pump components, clean filtered water shall be provided. If such quality water is not available at the site, then process water may be filtered, using either a cyclone separator, a mechanical filter, or a tank with a filter bed. When fluids other than water are handled, such fluids can similarly be filtered and used for injection. The pressure drop across the filter shall be monitored to ensure that the required injection pressure is available, and filter maintenance shall be performed when required. Additional bearing protection can be provided by installing a flow switch in the injection line, set for the minimum flow requirement.

2.4.6.4 Nonreverse ratchets

Non-reverse ratchets are furnished as an integral part of the motor or right angle gear when reverse rotation from backflow in the pump may cause damage. While the motor or gear is still disconnected from the pump, rotate the motor or gear by hand in both directions to check proper functioning of the ratchet. The rotation of the complete drive train should also be checked at this time.

2.4.6.5 Controls and alarms

All control and alarm systems, which may be electrical, hydraulic or pneumatic, must be checked for correct installation and functioning in accordance with the manufacturer's instructions. All alarm point settings must be verified.

2.4.6.6 Final alignment check – Factors causing misalignment

After the grout has set and the foundation bolts have been properly tightened, the unit alignment should be checked. After the suction and discharge piping of the unit have been connected, the alignment should be checked again.

If the unit does not stay in alignment after being properly installed, the following are possible causes:

1) Setting, seasoning or springing of the foundation;

2) Excessive pipe strain distorting or shifting the machine.

2.4.6.7 Special considerations for submersible units

While it is important to comply with the manufacturer's installation instructions for all equipment, this is imperative for submersible pumps to avoid instantaneous start-up failure, since the unit cannot be observed at this stage. Submersible motors vary greatly in basic construction, so only a few general guidelines can be provided.

For storage prior to installation, the manufacturer will specify whether the motor should be kept in a horizontal or vertical position.

For motors filled with either oil or other special fluid, check for leakage at the shaft seal prior to installation. Check the fluid level in the motor and refill with the manufacturer's recommended fluid per the instructions, if required.

If the power cable is to be connected to the motor terminal box in the field, make sure the connection is dry and the gaskets undamaged before bolting up the joint.

Keep the reel with the power cable close to the well head, so that the cable insulation does not become damaged by being dragged over the ground or over the well casing flange when the unit is lowered into the well. Similarly, clamps for securing the cable to the discharge pipe must not have sharp edges.

Before connecting the discharge pipe, it may be desirable to add a check valve above the bowl assembly.

The couplings for the discharge pipe joints must be tightened securely to prevent the motor's induced starting torque from either loosening or further tightening the joints. This would cause the power cable to spiral around the pipe and could cause cable or terminal failure. When the unit has been completely installed, a megger (meg-ohmmeter) reading should be taken on the cable/motor per the manufacturer's instructions to verify complete electrical integrity. If the megger reading is below the manufacturer's recommended minimum, the problem must be identified and corrected before the unit is started.

The necessary electrical controls must be provided in the starting panel. A time delay relay between stops and starts of the unit is recommended by most manufacturers.

2.4.7 Operating vertical pumps

2.4.7.1 System preparation

2.4.7.1.1 Flushing and filling

When the pump is installed in the completed piping system, it is recommended that the system be back-flushed to remove debris such as stubs of welding rod, welding slag and loose scale. The pump and other sensitive equipment should be protected with start-up strainers, which should in turn be removed upon completion of the flushing cycle. For barrel-mounted pumps, it is recommended to remove the pump and let the barrel become the receptacle for the debris for subsequent cleanout.

The pump must not be run unless it is completely filled with liquid or, for vertical lineshaft and submersible units, is provided with the minimum required submergence, as there is danger of damaging some of the pump components. Typically, bowl and impeller rings and internal sleeve bearings depend on liquid for their lubrication and may seize if the pump is run dry.

For pumps mounted in a suction barrel or can, typically for critical NPSH applications, a continuous vent line should be provided from the highest point in the barrel to the vapor phase of the suction source. This prevents inadvertent vapor locking and dry-running of the pump. The vent line must be continuously rising to preclude liquid traps and be fully airtight.

When the required submergence is provided, all submersible units and most vertical turbine pumps can be started without concern for the non-submerged part of the pump. For vertical lineshaft pumps, this, however, depends on the column length and bearing construction, such as metallic and nonmetallic material.

2.4.7.1.2 Priming

Most vertical pumps have the first stage below the liquid level. Therefore, they are automatically primed by proper venting. When required, as for horizontal or barrel pumps, priming may be accomplished by ejector/exhauster or vacuum pump.

2.4.7.1.2.1 Priming by ejector or exhauster

When steam, high-pressure water, or compressed air is available, the pump may be primed by attaching an air ejector to the highest point on the discharge nozzle or discharge pipe, close to the discharge valve. This will remove the air from the pump and suction can for barrel-mounted pumps, provided the discharge valve forms a tight seal. Prime is obtained when a steady stream of fluid flows from the ejector or discharge vent connection. The pump can then be started. A foot valve is unnecessary when this kind of device is used. Note that when the pump discharge nozzle is located above the suction source, and a foot valve is not used, the discharge valve must not be opened until the driver has been started, since this may result in loss of prime.

2.4.7.1.2.2 Priming by vacuum pumps

When neither of the above methods are practicable, the pump may be primed by the use of a vacuum pump to exhaust the air from the pump and suction can if applicable. A wet vacuum pump is preferable, as it will not be damaged if water enters. When a dry vacuum pump is used, the arrangement must preclude liquid from being drawn into the air pump. The manufacturer's instructions should be followed.

NOTE – Careful attention to priming requirements at the time of installation may save later annoyance because of improper equipment or procedure.

2.4.7.2 Start-up

Before starting the pump, check the direction of rotation. The proper direction is usually indicated by a direction arrow on the discharge head or on the driver stand when the discharge is located below the mounting level. When electric motors are used as drivers, the rotation should be checked with the motor disconnected from the pump.

The rotating element in vertical turbine pumps must be raised axially before start-up. An adjustable head shaft nut or pump-to-driver shaft coupling is provided for this purpose, and the pump shaft must be raised per the manufacturer's instructions.

The rotation of submersible units can normally be checked by comparing the pump output against the guaranteed performance curve. Check the manufacturer's start-up instructions.

As a general rule, the differential temperature between the pump and the liquid to be handled should not exceed 100°F (38°C) to avoid thermal shock and potential pump damage.

CAUTION: Before starting the pump, adequate submergence must be provided for vertical turbine and submersible pumps, and the casing and suction line must be filled with liquid for barrel-mounted pumps.

2.4.7.2.1 Speed-torque curves

A plot of speed versus torque requirements during the starting phase of a pump can be checked against the speed versus torque curve of the driving motor. The driver must be capable of supplying more torque at each speed than required by the pump in order to accelerate the pump up to rated speed. This condition is generally easily attainable with standard induction or synchronous motors, but under certain conditions, such as high specific speed pumps or reduced voltage starting, a motor with high pull-in torque may be required. For additional information on speed versus torque requirements, see Paragraph 2.3.2.7 in the Design and Application Section.

2.4.7.2.2 Across-the-line start

When squirrel cage induction motors having line starting controls are used, it is permissible to have the discharge valve open when the pump is being started. However, the length of time of the electrical disturbance, due to starting, may be shortened if the discharge valve remains closed until the pump comes up to full speed.

2.4.7.2.3 Reduced voltage start

Except for axial flow and mixed flow pumps, pumps using squirrel cage induction motors with reduced voltage starting control should always be started with the discharge valve closed or partially opened.

2.4.7.2.4 Soft start/acceleration controls

To achieve a smooth start for the pumping equipment, autotransformers may be connected to the starting panel. These provide a gradual increase in voltage up to rated voltage, insuring even acceleration.

2.4.7.3 Valve setting

2.4.7.3.1 Warning against closed valve operation

Brief shut-off operation of most vertical pumps is often necessary. The necessity may arise from system start-up or shutdown requirements and is normally met by closure of the discharge valve for the minimum possible time. Prolonged operation of the pump under this condition may prove harmful to the structural integrity of the pump mainly because of:

- Increased vibration levels affecting the stuffing-boxes, mechanical seals and areas with close-running fits;
- Increased axial thrust and resultant stresses in the shafts and bearings;
- Heat build up resulting in a dangerous temperature rise of the liquid being handled and pump components in contact with it;
- Damage resulting from internal recirculation and flow separation.

When a pump has been started against a closed discharge valve, it should be opened slowly as soon as pressure develops at the pump side of the valve. Abrupt valve opening can result in surges damaging to the pump and piping.

High specific speed pumps often have high zero flow horsepower. Running such pumps with the discharge valve closed can result in serious mechanical overloads as well as motor overload.

Operation of a pump with the suction valve closed may cause serious damage and should not be attempted. Operation with both valves closed for even brief periods of time is an unacceptable and dangerous practice. It can rapidly lead to a violent pump failure.

2.4.7.3.2 Valve setting at start-up

2.4.7.3.2.1 Position of discharge valve on starting, high or medium head pumps

A high or medium head pump, when primed and operated at full speed with the discharge valve closed, requires less power input than when operated at its rated capacity and head with the discharge valve open. For this reason, it is advantageous to have the discharge valve closed when starting the pump. It is to be noted, however, that, with pumps of 5,000 specific speed and higher, closing of the discharge valve at starting leads to an increased horsepower requirement.

2.4.7.3.2.2 Position of discharge valve on starting, mixed or axial flow pumps

Pumps of the mixed flow type usually require greater input power with the discharge valve closed than open. Axial flow type pumps nearly always require substantially more power at shut-off than at rating and must be started with the discharge valve open or with the opening of the valve sequenced with starting of the pump. Flap valves are commonly used for these purposes. The manufacturer's instructions should be consulted for the characteristics of such pumps.

2.4.7.3.2.3 Reduced flow/minimum flow discharge bypass

When operating at reduced flow, noise levels as well as vibration levels typically increase. This may lead to reduced bearing life and mechanical seal life as well as potential damage to other components.

If it becomes necessary to operate a pump for prolonged periods at flows below the rate specified by the manufacturer as permissible continuous minimum flow, then a bypass line should be installed from the pump discharge to the suction source. The bypass line should be sized so that the system flow plus the bypass flow is equal or larger than the manufacturer's specified minimum.

2.4.7.4 Draw-down in wells

Once pumping starts, the water level in the well will draw down. However, excessive draw-down may cause the unit, either lineshaft or submersible, to break suction, with resulting potential pump damage. Installation of undercurrent relays in the power supply lines will normally provide protection against this occurrence.

2.4.7.5 Checking speed, capacity, pressure, power, vibration and leaks

Once the unit is energized, check operating speed, capacity, suction and discharge pressure, and power input. While it may not be possible to exactly repeat the factory performance, initial field test data becomes a valuable baseline for future checking to determine possible wear and need to perform maintenance. Vibration levels should be checked for the same reason. Auxiliary piping and gasketed joints should be checked for leaks and proper makeup.

2.4.7.6 Water (hydraulic) hammer

Water hammer is an increase in pressure due to rapid changes in the velocity of a liquid flowing through a pipeline.

Water hammer may be controlled by regulating valve closure time, using relief valves or surge chambers and certain other means (see Paragraph 2.3.2.5).

It is recommended that specialized engineering services be engaged for water hammer analysis.

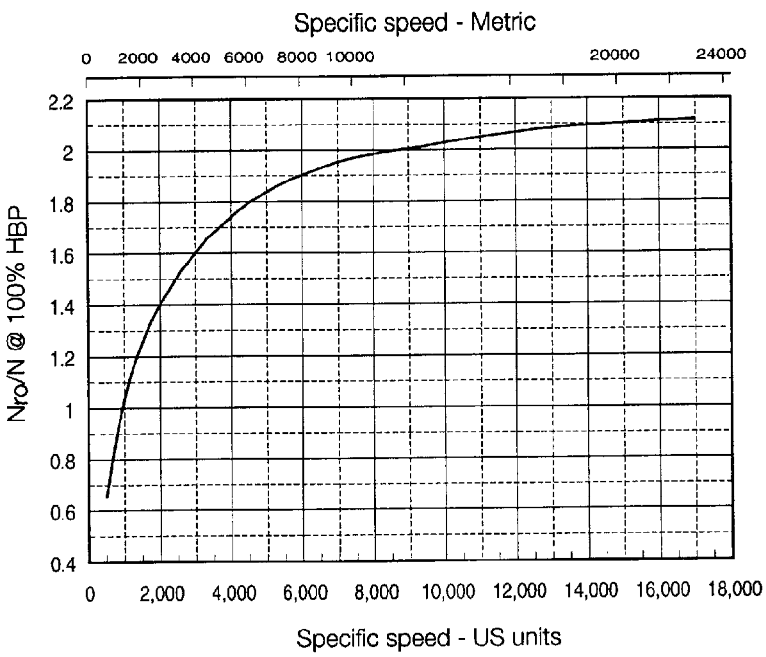


Figure 2.40 — Reverse runaway speed ratio versus specific speed

2.4.7.7 Parallel and series operation

Pumps should not be operated in series or in parallel unless specifically procured for this purpose, since serious equipment damage may occur.

For parallel operation, the pumps must have approximately matching head characteristics. Otherwise, the system operating head may exceed the shut-off head of one or more pumps, resulting in the pump(s) operating with zero output flow. This would have the same effect as operating against a closed discharge valve.

In series operation, the pumps must have approximately the same flow characteristics. Since each pump takes suction from the preceding pumps, the stuffing-boxes must be designed for the corresponding pressure, and the thrust bearing requirements may also change.

2.4.7.8 Stopping unit/reverse runaway speed

A sudden power and/or discharge valve failure during pump operation against a static head will result in a flow reversal, and the pump will operate as a hydraulic turbine in a direction opposite to that of normal pump operation.Vertical pump drivers can be equipped with non-reverse ratchets to prevent reverse rotation. However, their application is not always desirable and a review should always be made with the manufacturer.

If the pump is driven by a prime mover offering little resistance while running backwards, the reverse speed may approach its maximum for the applicable specific speed. This speed is called runaway speed. If the head under which such operation may occur is equal or greater than that developed by the pump at its best efficiency point during normal operation, the runaway speed exceeds that corresponding to normal pump operation. This excess speed may impose high mechanical stresses on the rotating parts both of the pump and prime mover, and knowledge of this speed is therefore essential to safeguard the equipment from possible damage.

It has been found practical to express the runaway speed as a percentage of the speed during normal operation. The head consistent with the runaway speed is assumed to be equal to that developed by the pump at the best efficiency point.

The ratio of runaway speed to normal speed for pumps varies with specific speed. This relationship is shown in Figure 2.40. The data shown should be used as a guide, recognizing that variations may be experienced with individual designs.

It should be pointed out that transient conditions, during which runaway speed may take place, often result in considerable head variations due to surging in the pressure line. Since most pumping units have relatively little inertia, surging can cause rapid speed fluctuations. The runaway speed may in such a case be consistent with the highest head resulting from surging. Therefore, knowledge of the surging characteristics of the pipeline is essential for determining the runaway speed. This is particularly important in case of long lines.

2.4.8 Pump vibration

There are a number of factors which may cause vibration in a pump. Imbalance, misalignment, looseness, bad bearings and fluid turbulence are some of the common sources of vibration. Sometimes, minor vibrations become major problems because of their frequency coinciding with a structural resonance. It might be necessary to either stiffen or weaken the structure and thereby change the resonant frequency. Vibration due to imbalance can, of course, be controlled through more precise balancing of the rotating element(s).

2.4.8.1 Radial vibration

Radial vibration is the most frequent type of pump vibration and is the primary subject of this section. With the radial vibration mode, also called lateral vibration, the displacement is predominantly in a plane perpendicular to the pump axis.

2.4.8.1.1 Vibration limits

Figure 2.41 shows the recommended acceptable vibration limits for vertical pumps. Experience as well as theoretical analysis has shown that there are typically no adverse effects on pump life or reliability due to vibration forces, as long as the vibration amplitudes do not exceed these limits.

The vibration limits are not easily classified for a large range of pumps and applications. Therefore, the recommended vibration levels are to be used as a general acceptance guide with the understanding that vibration levels in excess of the curve values may require investigation and close watch.

Vibrations in excess of the curve values may be acceptable if they show no continued increase over long periods of time and there is no other indication of damage, such as an increase in bearing clearance or noise level.

Machinery vibration severity charts, published by various sources, should not be applied to pumps indiscriminately. They were developed for machines primarily subject to unbalance and misalignment forces without hydraulic damping. Pumps in some applications are also exposed to significant hydraulic exciting forces.

Furthermore, vertical motors or gears are frequently mounted on the pump discharge head, which tends to result in tall and relatively flexible structures. These facts must be and are recognized in this Standard.The displacement values on Figure 2.41 are for average (RMS) peak-to-peak vibration readings. Figure 2.41 assumes:

1) Operation under steady conditions at the rated speed(s) and at the best efficiency point with no cavitation or air entrainment;

2) Discharge piping that is connected and anchored so as to avoid strains on the pump;

3) Drivers that are aligned within the pump manufacturer's recommendations;

4) A foundation of adequate mass and rigidity with proper anchor bolts and grouting.

The location of the vibration sensors for acceptance testing is shown in Figure 2.44.

Conversion formulas for vibration readings are as follows:

$$\delta = 1.919 \times 10^4 \frac{\varphi}{f}$$

$$\varphi = 3.696 \times 10^3 \frac{a}{f}$$

$$a = 2.704 \times 10^{-4} \varphi f$$

Where:

δ = Displacement, peak to peak, in mils (0.001 inches);

φ = velocity, peak, in inches per second;

a = Acceleration, peak in $\frac{inches}{sec^2}$;

f = Frequency, in Hertz.

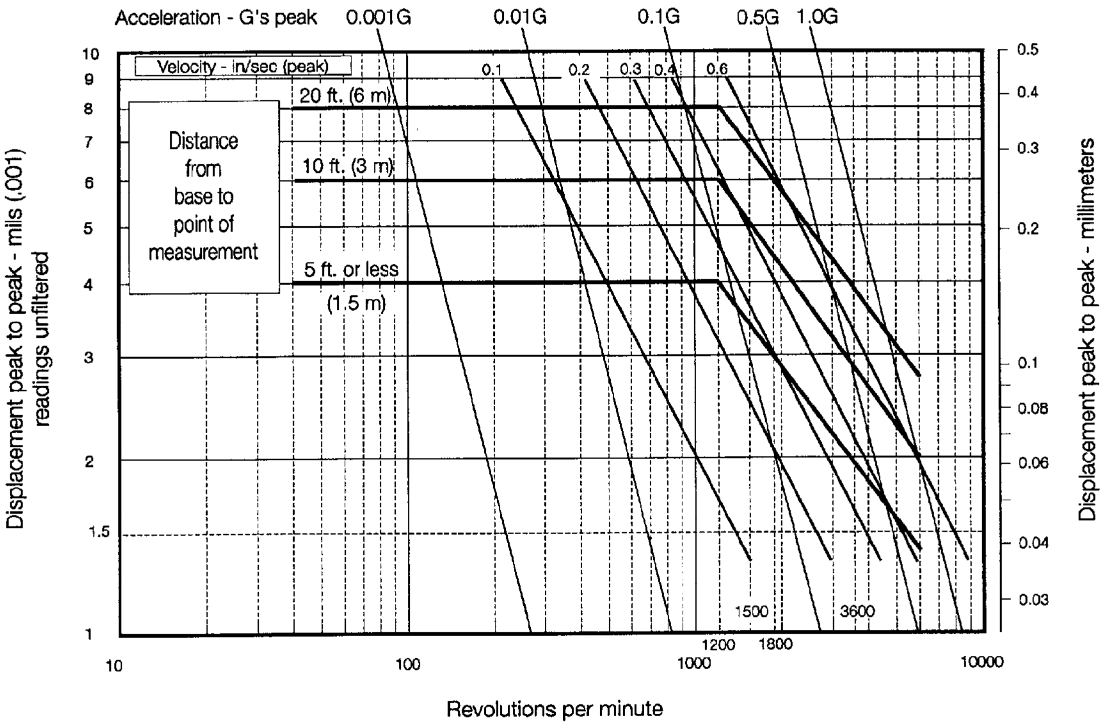


Figure 2.41 — Acceptable field vibration limits for vertical turbine pumps, mixed flow and axial flow pumps at best efficiency at top motor bearing housing

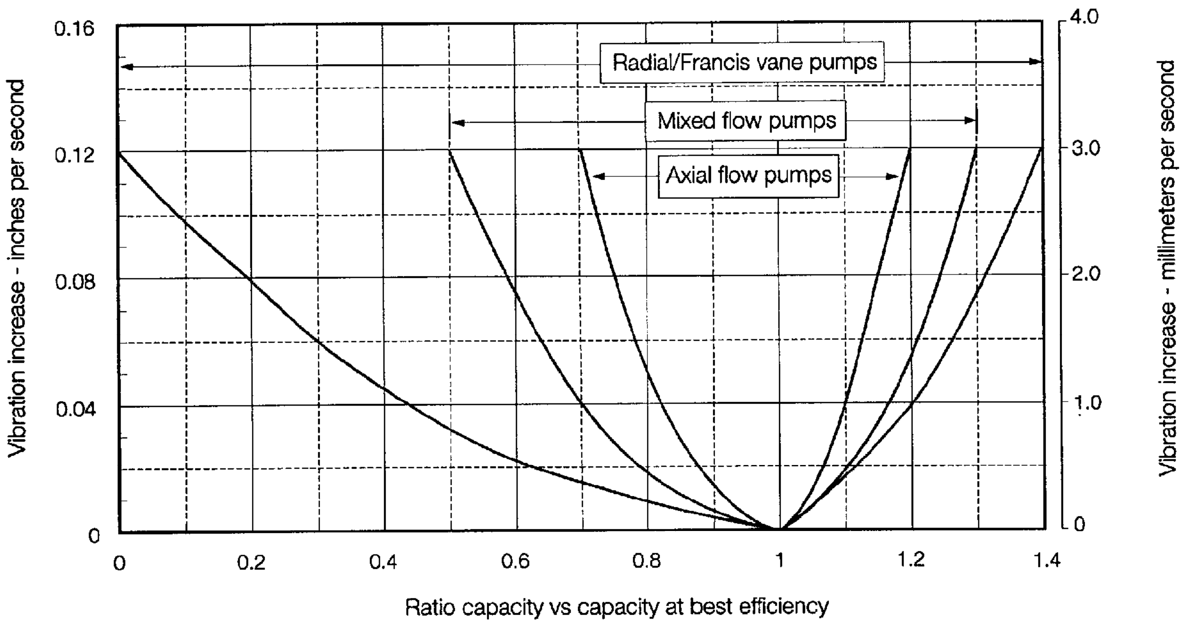


Figure 2.42 — Increase in allowable vibration at capacities other than at best efficiency

2.4.8.1.2 Factors affecting vibration

2.4.8.1.2.1 Operation at other than best efficiency point

When vibration measurements are made at capacities other than at best efficiency, the pump will tend to vibrate more. The allowable increase in vibration can be determined from Figure 2.42.

2.4.8.1.2.2 Unbalance of rotating parts and rotor balancing

Unbalance of the pump rotor can generate high unbalance forces, result in excessive bearing and shaft loading and induce high vibration levels. The balancing method and the residual unbalance limits are as follows:

Component balance shall be single-plane spin balance to ISO 1940-1986 E balance quality grade G6.3 (See Figure 2.43) When the ratio of the largest outside diameter of the component divided by the distance between the correction planes is less than six, a two-plane balance may be required. Other grades may be used if agreed upon by the user and manufacturer.

NOTE — In the specific case of impellers, the width is measured at the periphery, including the thickness of any shrouds but not the backvane.

Balance machine sensitivity shall be adequate for the part to be balanced. This means that the machine is capable of measuring unbalanced levels to one-tenth of the maximum residual unbalance allowed by the balance quality grade selected for the component being balanced.

Balance machines shall be calibrated at least annually. When specified, calibration shall be done just prior to balancing.

Pertinent aspects:

- The balance grade specified in the above standard yields a level or residual unbalance in rotating components consistent with clearance fits between the impeller and shaft. Rotating assembly balance is recommended if tighter quality grades, e.g., G2.5 or G1.0, are desired. For those instances where non-clearance fits are applicable, agreement between the manufacturer and customer should be reached if a different component balance quality grade is desired;
- Balance machine sensitivity is a function of the ratio of the weight of the part to the weight rating of the machine. (Above 100%, one must check with the manufacturer of the balance machine.) As an example, a 100 lb.-rated machine may provide adequate sensitivity and accuracy for a 10-lb. part, but a 20-lb. rated machine would be much more suited for the task, and a 3-lb. part may not balance at all on the 100-lb. machine, to the quality grade required.

Following are guidelines for the quality of balance procedure, equipment, tooling and also for rotor geometry, so that users and manufacturers alike can have a common ground for discussing these issues which have been learned through experience.

1) Inherent balance and/or runout in balancer drive or balancing arbor.

The balancer drive may be checked by periodically rotating the drive splines 180 degrees after a part has been balanced and checking the residual unbalance. It should be within 10% of the original unbalance. Runout in the balancing arbor should be checked when assembled in the balancer. It should be no more than .001 total indicator movement;

2) Keys/keyway geometry errors.

Special care must be taken to ensure that keys and keyways in balancing arbors are dimensionally identical to those in the assembled rotor. Like the arbor, they should be of hardened tool steel, to resist error introduced through wear;

3) Excessive looseness between impeller hub and balancing arbor.

The following guidelines for maximum looseness between balancing arbor and impeller, for general cases, is suggested. At no time should this looseness be greater than that found on the assembled rotor;

Diameter rotor journal impeller hub bore – inches	Maximum looseness (diametral)	
	(≤ 1800 rpm)	(1800 ≤ 3600 rpm)
0 — 1.499 in. 0 — (38 mm)	.0015 in. (38 mm)	.0015 in. (.038 mm)
1.5 — (1.999 in.) (38 mm) (50.8 mm)	.0020 in. (.051 mm)	.0015 in. (.038 mm)
2.0 — above (50.9 mm) — above	.0025 in. (.051 mm)	.0015 in. (.038 mm)

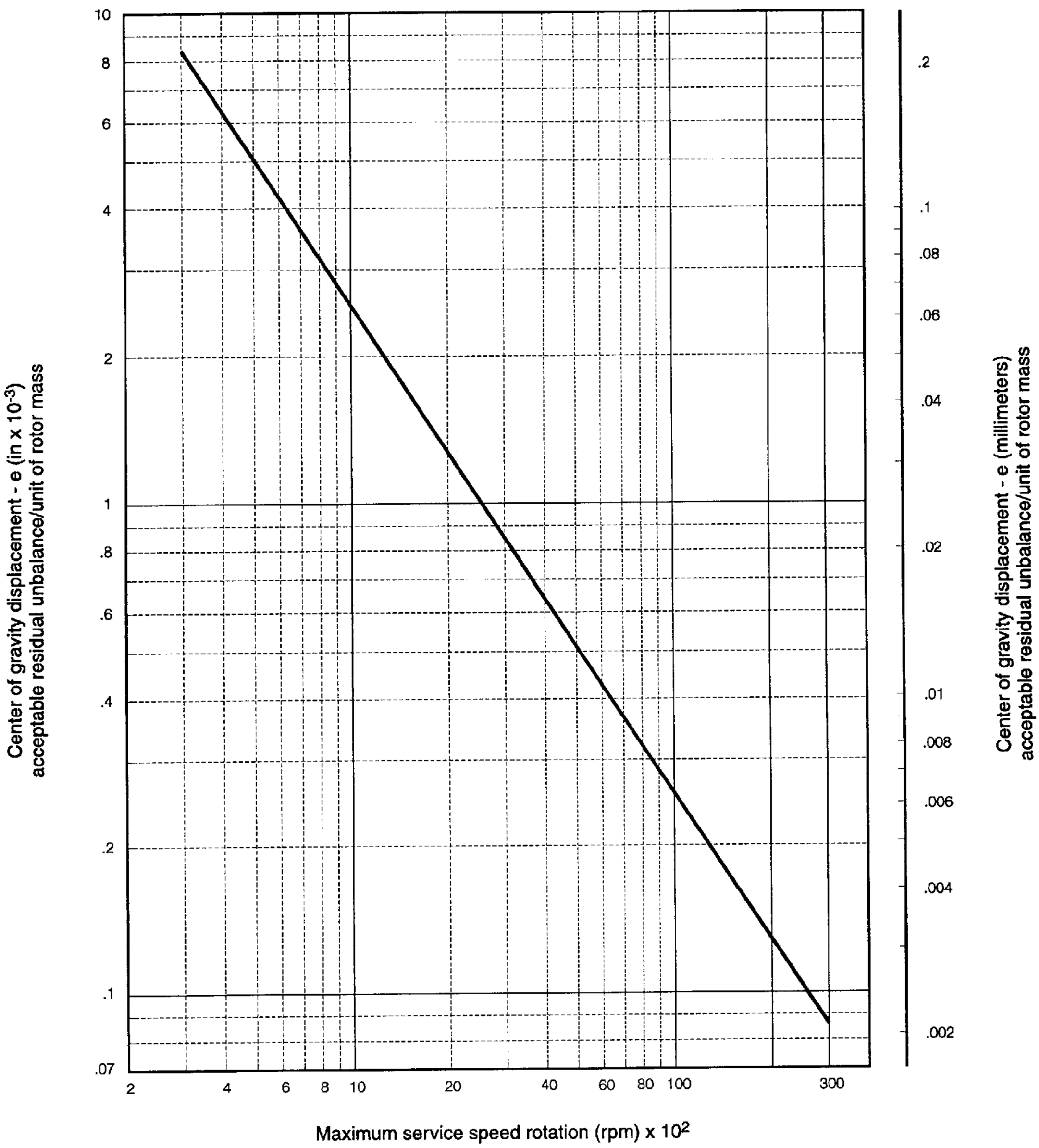


Figure 2.43 — Allowable residual unbalance in pump impellers – grade G6.3

NOTE — For those impellers to be run at 3600 rpm and having a straight radial clearance with the shaft, the maximum looseness specified, while a practical limit, may not assure a G6.3 residual unbalance when removed from the balance arbor and mounted on the pump shaft. G10 specification on impeller balance or a rotating assembly balance is recommended.

4) Removal or addition of material.

Material removal: This should be done in a way to spread the balance correction as evenly as possible over the surface. If a shroud is used, the thickness removed should be no more than 1/3 of the original, and the subsequent finish should be equal to the remainder of the shroud. If the impeller vane is used for balance correction, no more than 1/4 of the vane thickness should be removed, always from the low-pressure side. Removal by drilling and/or end milling should follow the same thickness guidelines, with appropriate consideration to minimizing flow.

Material addition: Sometimes for very large rotors with large amounts of unbalance it becomes desirable to add material, so that the shroud/vane thickness guidelines are not violated. This is permissible as long as impeller finish and discontinuities to flow are not radical and the method of material addition is consistent with requirements for mechanical integrity and material properties of the impeller/component for the intended/specified service.

The removal or addition of material to the impeller to effect the balance tolerance should be performed so as to preserve the impeller's geometry and minimize any flow discontinuities.

Unbalance of the driver, especially when it is mounted on the pump, and unbalance of the coupling will have the same effect as pump rotor unbalance and require the same treatment.

2.4.8.1.2.3 Field balancing

It is sometimes impractical to balance a pump driver assembly in the shop because of size, speed, etc. Furthermore, it is usually found that a shop-balanced assembly produces different vibration levels in the field because of the dissimilarities between shop and field foundations, job or test driver, piping, etc. Field balancing is an acceptable means for meeting guaranteed field vibration limits.

2.4.8.1.2.4 Natural frequency and resonance

Operation of a pump at a rotational speed near one of the lower natural frequencies (reed frequency) of the structure can result in a resonance. Theoretically, the vibration levels could become infinite, but the presence of damping generally limits the levels to several times that shown in Figure 2.41.

Pump manufacturers can calculate or determine by test the natural frequency of the pump assembly. However, in a field installation, the vibrating structure comprises, in addition to the pump, the foundation, the mounting, the piping and its supports, and the driver. The natural frequency of the vibrating structure is determined by the stiffness of the total structure and by its equivalent mass. It may therefore differ significantly from the natural frequency of the pump alone.

In the absence of specific information, the pump manufacturer will assume that the piping is installed rigidly and anchored close to the pump connections and that the hold-down bolts are securely embedded in a concrete foundation of infinite mass and rigidity.

The natural frequency of the system must not fall within the pump operating speed range. One must also be aware of the much lower stiffness of fabricated system foundation structures as compared to concrete and of problems associated with calculating accurately the stiffness of unconventional and composite structures. This is especially true in the case of pumps with variable speed drives.

2.4.8.1.2.5 Hydraulic disturbances

Vibration is always caused by a driving force. Hydraulic disturbances in the pump may generate this force. Following is a list of some typical hydraulic disturbances.

1) Recirculation and radial forces at low flows;

2) Fluid separation at high flow;

3) Cavitation due to NPSH problems;

4) Flow disturbances in the pump intake due to improper intake design;

5) Air entrainment or aeration of the liquid;

6) Hydraulic resonance in the piping.

The pump manufacturer must establish application limits for low flow. The system designer is responsible for giving due consideration to the remaining items.

2.4.8.1.2.6 Other mechanical problems

Misalignment of the shafting, damaged bearings, bent shafts, inadequate piping supports, and expansion joints without tie rods can also cause vibration.

2.4.8.1.2.7 Effect of rigidity

The amplitude of the vibration resulting from a given driving force is related to the rigidity of the vibrating structure.

For example, a short vertical pump would be more rigid than a tall one, and lower vibration amplitudes would be expected. The higher flexibility of the taller structure, however, allows higher vibration amplitudes for the same effect on life of the components.

The vibration limits for vertical pumps (Figure 2.41) recognize this fact, allowing higher levels for tall pumps.

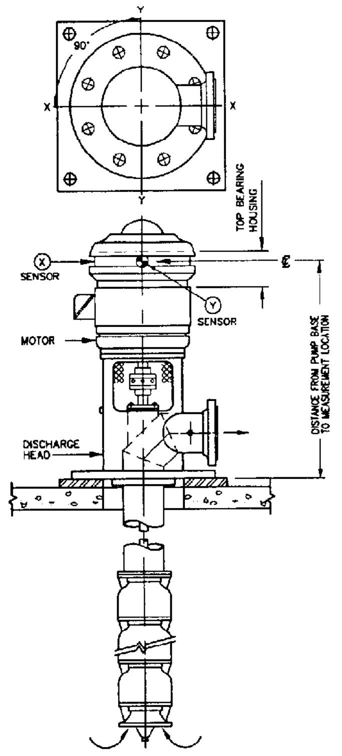


Figure 2.44 — Location of vibration sensors

2.4.8.2 Torsional vibration

Rotor torsional criticals can theoretically be present any time there are two rotating masses connected by shafting that is not infinitely stiff. This implies that any pump rotor coupled to a driver has a torsional natural frequency.

With vertical pumps, the primary hydraulic exciting force is generated by the impeller vanes passing the bowl vanes. The energy level of this force is low, with resulting negligible vibration amplitudes. Torsional vibration problems are therefore rarely seen on pumps with direct electric motor drives.

On the other hand, when a pump is driven through a gear, the inaccuracies in the gear can provide the exciting force and cause high torsional criticals at tooth-meshing frequencies. Similarly, engines can also cause high torsional vibrations at higher frequencies. It is therefore recommended that gear- or engine-driven pumps be analyzed for torsional criticals.

Computer programs are available for predicting the torsional critical speeds and expected amplitudes. The mathematical models on which these calculations are based are quite accurate and yield realistic results. The analysis should be made during the design stage of the installation. Shifting of the natural frequency can be relatively easily accomplished by selecting a flexible coupling with proper torsional stiffness.

The existence of torsional vibration rarely shows itself in the pump column or motor housing vibration. The gear bearings or housings may or may not show a problem. Thus pump shafts, couplings or gears can fail without the usual vibration-monitoring equipment indicating any danger.

Rather complicated equipment is required to measure torsional vibrations. It may consist of strain gauges mounted on the rotating shafting with proper signal transmission devices, or torsiongraphs based on pulse gear train or seismometer principles. Due to the sophistication of this equipment and problems installing it on the pump, torsional tests are rarely run in the field.

2.4.8.3 Vibration measurement

The vibration sensors should be placed on the upper motor bearing housing in both x and y axis locations.

Figure 2.44 illustrates the sensor location. The sensors must not be placed on a flexible panel or cylinder wall, such as on motor end covers. Such covers should be removed to allow measurements on a rigid part of the machine.

Due to technical difficulties with submerged sensors and accessibility problems in actual field installations, sensors are normally not located on the pump bowl or pump column.

2.4.9 Noise in pumping machinery

Sound is energy and may be produced by movement within machinery. This is also true for pumps. Sound is produced by liquid flowing within the pump, the bearings within the pumping unit, the coupling, and the unit driver. Not all sound is objectionable. Sound may be transmitted in three ways:

1) Airborne within the machinery room;

2) Liquid-borne by the liquid being pumped;

3) Structure-borne through the attached piping and support system.

Two of the most important factors in minimizing sound in pump installations are the correct selection of the pump type for the operating conditions and the equipment installation. To insure minimum sound, the pump should be chosen for operation near the point of best efficiency and proper suction conditions should be provided.

The prevention of noise is greatly dependent upon the pump installation. Proper alignment of the pump and the driver is essential, as well as the support of the suction and discharge piping. The manner in which the pump is installed and in which the piping is supported may contribute to objectionable noise and vibration. A greater degree of noise prevention may be obtained when the pumping unit is supported free of building structures by the use of vibration isolators and flexible piping and conduit connectors. Noise emanating from the motion of high-velocity liquids within the piping system, particularly from partly opened valves, should not mistakenly be attributed to the pumping unit. Further discussion of noise and sound is contained in *HI 9.1–9.6 Pumps – General Guidelines.*

2.4.9.1 Hydraulic resonance in piping

Severe vibration problems are often caused by a resonant condition within the pump/piping system which amplifies normal pump-induced pulsations. Such a condition is referred to as a hydraulic resonance.

Hydraulic resonance is defined as a condition of pulse reinforcement in which pulses reflected by the piping system are repeatedly added in phase to the source pulse, producing large pulsation amplitudes. Hydraulic resonance in piping may result in unacceptable noise or vibration, or if uncorrected it can ultimately result in mechanical fatigue failures in either the piping or pump components.

In cases where the existence of a hydraulic resonance is known to be a problem, experience has shown that the following solutions, aimed at alleviating the resonant condition, may prove effective:

1) Alter the resonant piping;

2) Change the pump speed;

3) Change the internal design characteristics of the pump;

4) Insert a pulsation damper on the pump/piping system.

Modifications to the pump or piping, including the supporting structures, which do not change the pulsation response of the pump/piping system, will not affect the resonant condition and therefore will not be effective.

2.4.10 Seismic analysis

For certain critical installations, such as nuclear power plants, the pumps, supports and accessories must be earthquake-resistant. The design specifications to achieve earthquake resistance vary, depending upon geographical area, class of the equipment (defining how critical the survival of the equipment is), and the characteristics (acceleration response) of the structure or foundation supporting the pump.

Complete specifications for earthquake-resistance requirements must be supplied by the customer. This includes:

- The seismic criteria, such as acceleration, magnitudes, frequency spectrum, location and direction relative to pump;
- The qualification procedure required, i.e., analysis, testing, or a combination of these and requirements for operability during and/or after test.

2.4.11 Maintenance of vertical pumps

When handling water, care should be taken to prevent the pump from freezing during cold weather when the pump is not in operation. It may be necessary to drain the pump casing on dry pit applications during shutdown periods by removing the bottom drain plug. In some pumps, draining of the suction line is sufficient. For vertical wet pit pumps, removal of the unit is required.

2.4.11.1 Wear/parts replacements

Wear rings are commonly fitted in the bowls (bowl rings) and if specified, on the impeller (impeller rings). These wear rings provide a close-running, renewable clearance, to reduce the quantity of liquid leaking from the high-pressure side to the suction side. These rings depend on the liquid in the pump for lubrication. They will eventually wear so that the clearance becomes greater and more liquid passes into the suction. This rate of wear depends on the character of the liquid pumped. Badly worn wear rings will result in severe degradation of pump performance, particularly on small pumps. Examination of wear patterns can provide valuable information in diagnosing pump problems and determining their origin.

It is not possible to recommend minimum spares to cover all conditions. However, the following may be taken as a guide:

a) For intermittent service:

- stuffing-box packing or mechanical seal;
- gaskets and "0" rings (complete set);
- packing gland and studs or gland bolts.

b) For continuous service (in addition to above);

- stuffing-box bearing;
- headshaft (if used);
- lineshaft (1 set);
- lineshaft coupling (1 set);
- sleeve bearings, both lineshaft and bowlshaft;
- pump shaft;
- impeller lock collets (1 set);
- bowl and/or Impeller wear rings (1 set).

2.4.11.2 Troubleshooting

When investigating pump trouble at the job site, every effort must first be made to eliminate all outside influences. If the performance is suspect, the correct use and accuracy of instruments should first be checked. In addition, note that pump performance is substantially affected by such fluid characteristics as temperature, specific gravity and viscosity.

2.4.11.2.1 No discharge

Lack of discharge from a pump may be caused by any of the following conditions:

1) Pump not primed;

2) Speed too low;

NOTE – When direct-connected to electric motors, determine whether the motor is across the line and receives full voltage. When direct-connected to steam turbines, make sure the turbine receives full steam pressure.

3) System head too high;

4) Suction lift higher than that for which pump is designed;

5) Impeller completely plugged;

6) Impeller installed backwards;

7) Wrong direction of rotation;

8) Air leak in the suction line;

9) Air leak through stuffing-box;

10) Well draw-down below minimum submergence;

11) Pump damaged during installation (wells);

12) Broken lineshaft or coupling;

13) Impeller(s) loose on shaft;

14) Closed suction valve.

2.4.11.2.2 Insufficient discharge

Insufficient discharge from a pump may be caused by any of the following conditions:

1) Air leaks in suction line or stuffing-boxes;

2) Speed too low;

NOTE – When direct-connected to electric motors, determine whether or not the motor is across the line and receives full voltage. When direct-connected to steam turbines, make sure the turbine receives full steam pressure.

3) System head higher than anticipated;

4) Insufficient NPSHA: Suction lift too high;

5) Clogged suction line or screen;

6) Not enough suction head for hot or volatile liquids;

7) Foot valve too small;

8) Impeller partially plugged;

9) Rings worn;

10) Impeller damaged;

11) Impeller(s) loose on shaft;

12) Excessive lift on rotor element;

13) Suction valve partially closed;

14) Leaking joints (well application);

15) Foot valve of suction opening not submerged enough;

16) Impeller installed backwards;

17) Wrong direction of rotation.

2.4.11.2.3 Insufficient pressure

Insufficient pressure from a pump may be caused by any of the following conditions:

1) Speed too low;

NOTE – When direct-connected to electric motors, determine whether the motor is across the line and receives full voltage. When direct-connected to steam turbines, make sure the turbine receives full steam pressure.

2) System head less than anticipated;

3) Air or gas in liquid;

4) Rings worn;

5) Impeller damaged;

6) Impeller diameter too small;

7) Impeller for wrong direction of rotation;

8) Wrong direction of rotation;

9) Excessive lift;

10) Leaking joints (well application).

2.4.11.2.4 Loss of suction following period of satisfactory operation

Loss of suction under these conditions may be caused by any of the following conditions:

1) Suction line drawing air;

2) Water seal plugged;

3) Suction lift too high or insufficient NPSHA;

4) Air or gas in liquid;

5) Casing gasket defective;

6) Clogging of strainer;

7) Excessive well draw-down.

2.4.11.2.5 Excessive power consumption

Excessive power consumption may be caused by any of the following conditions:

1) Speed too high;

NOTE – When direct-connected to electric motors, determine whether the motor is across the line and receives full voltage. When direct-connected to steam turbines, make sure the turbine receives full steam pressure.

2) System head lower than rating, pumps too much liquid (radial and mixed flow pumps);

3) System head higher than rating, pumps too little liquid (axial and mixed flow pumps);

4) Specific gravity or viscosity of liquid pumped is too high;

5) Shaft bent;

6) Rotating element binds;

7) Stuffing-boxes too tight;

8) Rings worn;

9) Electrical or mechanical defect in submerged motor;

10) Undersized submersible cable;

11) Incorrect lubrication of driver;

12) Lubricant in shaft-enclosing tube too heavy.

2.5 Reference and source material

Hydraulic Institute, *Engineering Data Book*

ANSI/HI 9.1-9.6, *Pumps – General Guidelines*

Hydraulic Institute, 9 Sylvan Way, Parsippany, NJ
07054-3802

American National Standard for

Vertical Pump Tests

Hydraulic
INSTITUTE

9 Sylvan Way
Parsippany, New Jersey
07054-3802

ANSI/HI 2.6-1994

ANSI/HI 2.6-1994

American National Standard for

Vertical Pump Tests

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Approved August 23, 1994

American National Standards Institute, Inc.

American National Standard

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Contents

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

a. To develop and publish standards for pumps;

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

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

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

e. To support educational and research activities;

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

Purpose of Standards

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

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

Definition of a Standard of the Hydraulic Institute

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

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

Comments from users

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

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

Revisions

The Standards of the Hydraulic Institute are subject to review, and revisions are undertaken whenever it is found necessary because of new developments and progress in the art.

Scope

This Standard is for vertical diffuser type centrifugal pumps. It includes detailed procedures on the setup and conduct of hydrostatic and performance tests of such pumps.

Several methodologies to test centrifugal and vertical pump equipment are available to pump manufacturers, users and other interested parties. The United States has two sets of pump test Standards which represent two approaches to conducting and evaluating pump performance. One, promulgated by the American Society of Mechanical Engineers (ASME) and designated PTC 8.2, Centrifugal Pumps, provides for two levels of tests and is based on a detailed procedure that produces results of a low level of uncertainty. The other, promulgated by the Hydraulic Institute (HI), designated HI 1.6, Centrifugal Pump Tests and HI 2.6, Vertical Pump Tests, also provides for two levels of test in which the test procedures are less restrictive. The ASME Code relies on the parties to the test to agree beforehand on the Scope and Conduct of the test and does not specify how the test results shall be used to compare with guarantee. The ASME is especially suited to highly detailed pump testing, whereas HI Standards detail test scope, conduct and acceptance criteria, and are thus suited to commercial test practices. ASME Codes do not permit the use of acceptability tolerances in reporting results, while the HI Standards do. It is recommended that the specifier of the test standard become familiar with both the ASME Code and the HI Standards before selecting the one best suited for the equipment to be tested, since there are a number of other differences between the two which may affect accuracy or cost of the tests.

Both the ASME and HI Standards can be used for testing in either field or factory installations. The detailed requirements of the ASME test Code are intended to reduce the effect of various installation arrangements on performance results and are applied more to field testing. The HI Standard specifies test piping and more controllable conditions, which is more suitable to factory testing. The HI Standards do not address field testing. Surveys have shown that both ASME and HI Standards have been applied successfully to applications from small chemical pumps (1 hp) to large utility pumps (over 5000 hp).

Units of Measurement

US Customary units of measurement are predominantly used, and, where appropriate, Metric unit equivalents appear in brackets following the US units. Sample calculations are shown with US units only.

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

The following organizations, recognized as having an interest in the standardization of vertical 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.

Agrico Chemical Corporation
American Petroleum Institute
Amer. Society of Heating, Refrigerating & Air-Conditioning Engineers
Amer. Society of Mechanical Engineers
Amoco Oil Company
Aurora Industries
Bechtel Corporation
Black & Veatch
BP America
Brown & Caldwell
Camp Dresser & McKee, Inc.
CH2M Hill
Chiyoda International Corporation
Commonwealth Edison
DeWanti & Stowell
Dexter Corporation
DuPont Engineering
Durametallic Corporation
Edison Electrical Institute
Electric Power Research Institute
Florida Power Corporation
Florida Power & Light
Fluor Daniel
Freese and Nichols, Inc.
G.E. Motors
HDR Engineering
Holabird & Root
Hydraulic Institute
Institute of Paper Science & Tech.
J. Brunner - Consultant
John Carollo Engineers
John Crane, Inc.
Malcolm Pirnie, Inc.
Marine Spill Response Corporation
Min Proc Eng., Inc.
Mobil Research & Development Corp.
Monsanto Chemical Company
Montana State University
Montgomery Watson
M. W. Kellogg Company
Naval Sea Systems
Naval Surface Warfare Center
Newport News Shipbuilding
Pacific Gas & Electric
Raytheon Engineers & Constructors
Riverwood International Georgia, Inc.
San Francisco Bureau of Engineering
Siemens Energy & Automation
Simons-Eastern Consultants
Sordoni-Skanska Construction Co.
Star Enterprises
State Farm Mutual Automobile Ins. Co.
State of California Dept. of Water Res.
Stone & Webster
Summers Engineering, Inc.
T. Hopkins - Consultant
Tennessee Eastman
Union Carbide Chemicals & Plastics Co.
US Bureau of Reclamation

2.6 Test

2.6.1 Scope

This standard applies to tests of the pump only, unless stated otherwise.

The type of test(s) performed, and the auxiliary equipment to be used, should be agreed upon by the purchaser and manufacturer prior to the test.

It is not the intent of this standard to limit or restrict tests to only those described herein. Variations in test procedures may exist without violating the intent of this standard. *Exceptions may be taken if agreed upon by the parties involved without sacrificing the validity of the applicable parts of this standard.*

This standard is limited to the testing of vertical diffuser type centrifugal pumps with clear water. The tests conducted under these standards shall be made and reported by qualified personnel.

2.6.1.1 Objective

To provide uniform procedures for hydrostatic, hydraulic, and mechanical performance testing of vertical pumps and recording of the test results. This standard is intended to define test procedures which may be invoked by contractual agreement between a purchaser and manufacturer. It is not intended to define a manufacturer's standard practice.

2.6.2 Types of tests

This standard describes the following tests:

a) Performance test to demonstrate hydraulic and mechanical integrity;

and the following optional tests when specified:

b) Hydrostatic test of pressure-containing components;

c) Net positive suction head required test (NPSHR test);

d) Priming time test.

For vibration testing, see *HI 2.1–2.5 Vertical Pump Standards* and for airborne sound testing see *HI 9.1–9.5, Pumps – General Guidelines.*

2.6.2.1 Test conditions

Unless otherwise specified, the capacity, head, efficiency, NPSHR and priming time are based on shop tests using water corrected to 68°F (20°C). If the facility cannot test at rated speed because of limitations in power, electrical frequency, or available speed changers, the pump may be tested at between 50% to 200% of rated speed.

2.6.3 Terminology

The following terms are used to designate test parameters or are used in connection with pump testings.

2.6.3.1 Symbols

See Table 2.11.

2.6.3.2 Subscripts

See Table 2.12.

2.6.3.3 Specified condition point

Specified condition point is synonymous with rated condition point.

2.6.3.4 Rated condition point

Rated condition point applies to the capacity, head, speed, NPSH and power of the pump as specified by the purchase order.

2.6.3.5 Normal condition point

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.

2.6.3.6 Best efficiency point (b.e.p.)

The capacity and head at which the pump efficiency (η_p) is a maximum.

2.6.3.7 Shut off (so)

The condition of zero flow where no liquid is flowing through the pump.

2.6.3.8 Volume

The unit of volume shall be one of the following:

- US units: US gallon;
- US units: cubic foot;
- Metric: cubic meter.

The specific weight of water at a temperature of 68°F (20°C) shall be taken as 62.3 lb. per cu. ft. (9.89 kN/m^3). For other temperatures, proper specific weight corrections shall be made using values from the ASME steam tables.

Table 2.11 – Symbols

Symbol	Term	US Customary Unit	Abbreviation	Metric unit	Abbreviation	Conversion factor [1)]
A	Area	square inches	in^2	square millimeter	mm^2	645.2
β (beta)	Meter or orifice ratio	dimensionless	—	dimensionless	—	1
D	Diameter	inches	in	millimeter	mm	25.4
Δ (delta)	Difference	dimensionless	—	dimensionless	—	1
η (eta)	Efficiency	percent	%	percent	%	1
g	Gravitational acceleration	feet/second/second	ft/sec^2	meter/second/second	m/s^2	0.3048
γ (gamma)	Specific weight	pounds/cubic foot	lb/ft^3	kiloNewton/cubic meter	kN/m^3	0.1571
h	Head	feet	ft	meter	m	0.3048
H	Total head	feet	ft	meter	m	0.3048
n	Speed	revolutions/minute	rpm	revolutions/minute	rpm	1
NPSHA	Net positive suction head available	feet	ft	meter	m	0.3048
NPSHR	Net positive suction head required	feet	ft	meter	m	0.3048
Ns	Specific speed $N_S = nQ^{1/2}/H^{3/4}$	dimensionless	—	dimensionless	—	1.162
ν (nu)	Kinematic viscosity	feet squared/second	ft^2/sec	millimeter squared/sec	mm^2/s	92900
π	pi = 3.1416	dimensionless	—	dimensionless	—	1
p	Pressure	pounds/square inch	psi	kilopascal	kPa	6.895
P	Power	horsepower	hp	kilowatt	kW	0.7457
q	Capacity	cubic feet/second	ft^3/sec	cubic meter/hour	m^3/h	101.94
Q	Capacity	US gallons/minute	gpm	cubic meter/hour	m^3/h	0.2271
ρ (rho)	Density	pound mass/cubic foot	lbm/ft^3	kilogram/cubic meter	kg/m^3	16.02
s	Specific gravity	dimensionless	—	dimensionless	—	1
t	Temperature	degrees Fahrenheit	°F	degrees Celcius	°C	(°F–32) x 5/9
τ (tau)	Torque	pound-feet	lb-ft	Newton – meter	N·m	1.356
v	Velocity	feet/second	ft/sec	meter/second	m/s	0.3048
x	Exponent	none	none	none	none	1
Z	Elevation gauge distance above or below datum	feet	ft	meter	m	0.3048

1) Conversion factor x US units = metric units.

Table 2.12 – Subscripts

Subscript	Term	Subscript	Term
1	Test condition or model	min	Minimum
2	Specific condition or prototype	mot	Motor
a	Absolute	ot	Operating temperature
atm	Atmospheric	OA	Overall unit
b	Barometric	p	Pump
ba	Bowl assembly	s	Suction
d	Discharge	t	Test temperature
dvr	Driver input	t	Theoretical
g	Gauge	v	Velocity
im	Intermediate mechanism	vp	Vapor pressure
max	Maximum	w	Water

2.6.3.9 Capacity (Q)

The capacity 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.

2.6.3.10 Speed (n)

The number of revolutions of the shaft in a given unit of time. Speed is expressed as revolutions per minute.

2.6.3.11 Datum

The reference line or eye of the first stage impeller from which all elevations are measured (see Figure 2.51).

Optional tests can be performed with the pump mounted in a suction can. Irrespective of pump mounting, the pump's datum is maintained at the eye of the first stage impeller (see Figure 2.52).

The elevation head (Z) to the datum is positive when the gauge is above datum and negative when the gauge is below datum.

2.6.3.12 Head (h)

Head is the expression of the energy content of the liquid referred to a datum. It is expressed in units of energy per unit weight of liquid. The measuring unit for head is foot (meter) of liquid.

2.6.3.12.1 Gauge head (h_g)

The pressure energy of the liquid determined by a pressure gauge or other pressure measuring device:

$$\text{(US units) } h_g = \frac{(2.31)(62.3)(p_g)}{\gamma} = \frac{2.31(p_g)}{s}$$

$$\text{(Metric) } h_g = \frac{p_g}{9.8\ s}$$

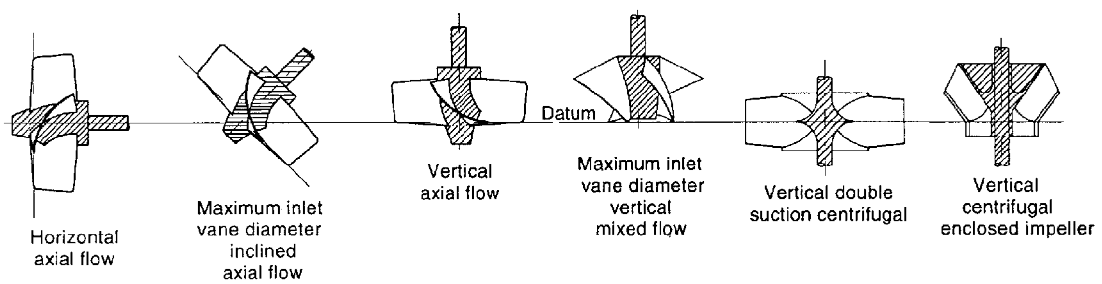


Figure 2.51 — Datum elevations for various pump designs

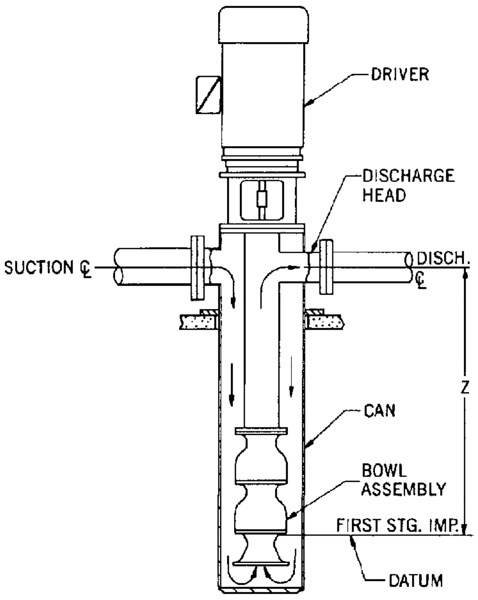


Figure 2.52 — First stage impeller datum closed suction – can pump

2.6.3.12.2 Velocity head (h_V)

The kinetic energy of the liquid at a given section. Velocity head is expressed by the following equation:

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

2.6.3.12.3 Elevation head (Z)

The potential energy of the liquid due to this elevation relative to a datum level measured to the center of the pressure gauge.

2.6.3.12.4 Total suction head (h_S) – open suction test

For open suction (wet pit) tests, the first stage impeller of the bowl assembly is submerged in a pit (see Figure 2.53).

The total suction head (h_S) at datum (see Figure 2.51) is the submergence in feet of water (Z_W). The average velocity head of the flow in the pit is small enough to be neglected:

$$h_s = Z_w$$

Where Z_w = vertical distance in feet from free water surface to datum.

NOTE – When absolute suction head is required for NPSH considerations, refer to Paragraph 2.6.6.4 for definition.

2.6.3.12.5 Total suction head (h_S) – closed suction test

For closed suction tests, the pump suction nozzle may be located either above or below grade level (see Figure 2.54).

The total suction head (h_S), referred to the eye of the first stage impeller, is the algebraic sum of the suction gauge pressure in feet (h_{gs}) plus the velocity head (h_{vs}) at point of gauge attachment plus the elevation (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 an amount exceeding the sum of the elevation head and the velocity head.

NOTE – When absolute suction head is required for NPSH considerations, see Paragraph 2.6.6.4 for definition.

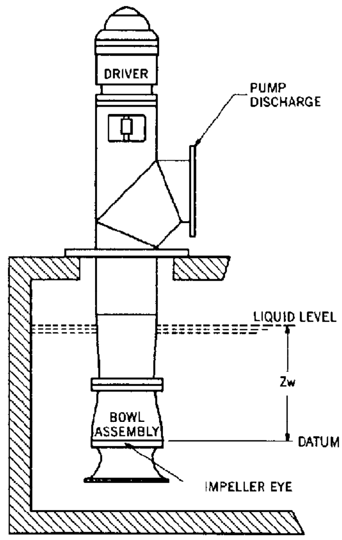


Figure 2.53 — Total suction head – open suction

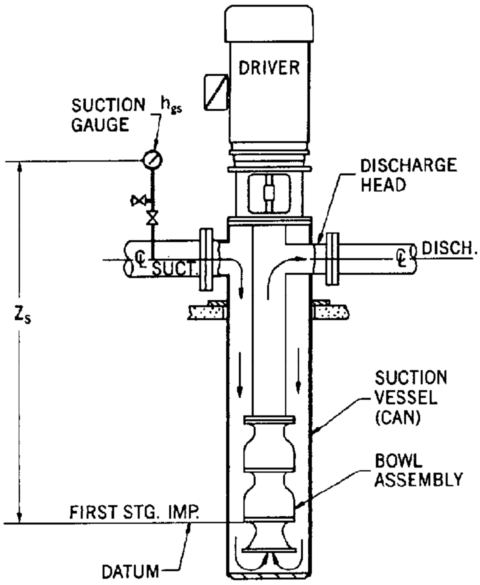


Figure 2.54 — Total suction head – closed suction

2.6.3.12.6 Total discharge head (h_d)

The total discharge head (h_d) is the sum of the discharge gauge head (h_{gd}), discharge velocity head (h_{vd}) and the elevation head (Z_d) from the discharge gauge centerline to the pump datum (see Figure 2.55).

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

For location of instruments for head measurements, see Paragraph 2.6.8.2.

2.6.3.12.7 Total head (H)

This is the measure of work increase per unit mass of the liquid, imparted to the liquid by the pump, and is the algebraic difference between the total discharge head and the total suction head.

This is the head normally specified for pumping applications. Since the complete characteristics of a system determine the total head required, this value must be specified by the user.

2.6.3.12.7.1 Open suction tests

For open suction tests, the total head (H) is the sum of the pressure head (h_{gd}) measured on the discharge pipe downstream from the discharge head, plus the velocity head (h_v) at point of gauge attachment, plus the vertical distance (Z_d) from datum to the pressure gauge centerline, minus the submergence (Z_w) (see Figure 2.55).

$$H = h_d - h_s = (h_{gd} + h_{vd} + Z_d) - Z_w$$

Hydraulic losses between the bowl assembly and the discharge nozzle are charged to the pump.

2.6.3.12.7.2 Bowl assembly total head (H_{ba}) (established on open suction test).

This is the developed head at the discharge of the bowl assembly and is a multiple of the head per stage as typically shown on the pump manufacturer's rating curves.

The bowl assembly total head (H_{ba}) is the gauge head (h_{gd}) measured at a gauge connection located on the column pipe downstream from the bowl assembly, plus the velocity head (h_v) at point of gauge connection, plus the elevation head (Z_d) from datum to the pressure gauge centerline, minus the submergence Z_w which is the vertical distance from datum to the liquid level.

$$H_{ba} = h_{gd} + h_{vd} + Z_d - Z_w$$

Friction losses in suction pipe and strainer, if used in the test setup, must be added to the measured head. The friction loss in the column between the bowl assembly outlet and the gauge connection must also be added if significant.

2.6.3.12.7.3 Closed suction tests

For closed suction tests (can pumps), and with the total discharge head (h_d) and the total suction head (h_s) referenced to datum (Figure 2.56), the total head is:

$$H = h_d - h_s$$

$$= (h_{gd} + h_{vd} + Z_d) - (h_{gs} + h_{vs} + Z_s)$$

When the suction gauge head (h_{gs}) is negative (below atmospheric) and the gauge connecting line free of liquid, then Z_s becomes the elevation distance from the pump suction centerline to datum.

All hydraulic losses between the pump suction and discharge nozzles are charged to the pump.

2.6.3.12.8 Atmospheric head (h_{atm})

Local atmospheric pressure expressed in feet (meters).

2.6.3.12.9 Net positive suction head available (NPSHA)

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

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

Where:

h_{sa} = Total suction head absolute = $h_{atm} + h_s$;

or

$$NPSHA = h_{atm} + h_s - h_{vp}$$

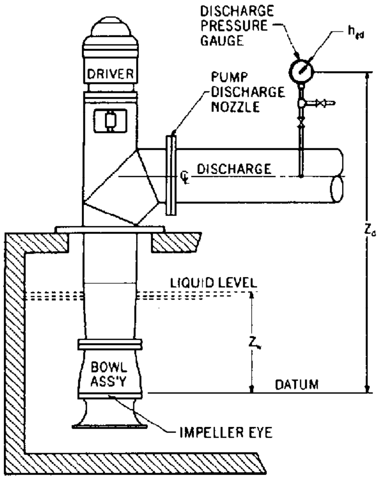


Figure 2.55 — Total head – open suction

or

$$\text{(US units) } NPSHA = \frac{2.31}{s}(p_{atm} - p_{vp}) + h_s$$

$$\text{(Metric) NPSHA} = \frac{(p_{atm} - p_{vp})}{9.8s} + h_s$$

2.6.3.12.10 Net positive suction head required (NPSHR)

Net positive suction head required (NPSHR) is the total suction head of liquid absolute determined at the first stage impeller datum less the absolute vapor pressure of the liquid in head of liquid pumped, required to prevent more than 3% loss in total head from the first stage of the pump at a specific capacity.

2.6.3.13 Power (P)

2.6.3.13.1 Pump input power (P_p)

The pump input power is the power needed to drive the complete pump assembly including bowl assembly input power, line shaft power loss, stuffing box loss and thrust bearing loss. With pumps having a built-in thrust bearing, the power delivered to the pump shaft coupling is equal to the pump input power. With pumps that rely on the driver thrust bearing, the thrust bearing loss shall be added to the power delivered to the pump shaft. It is also called brake horsepower.

2.6.3.13.2 Electric driver input power (P_{mot})

The electrical input to the driver expressed in horsepower (kilowatts).

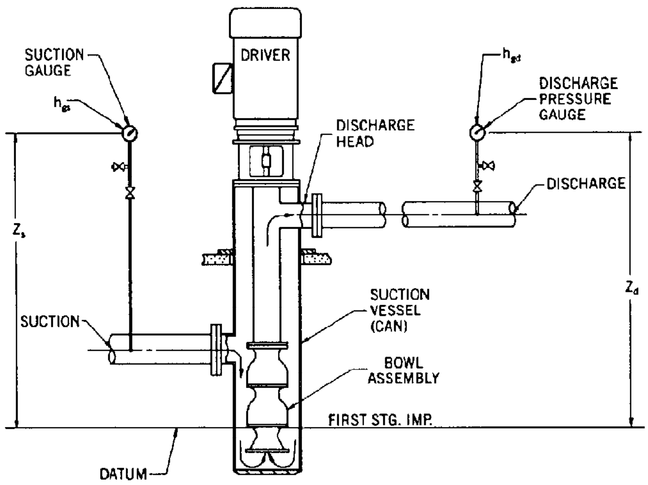


Figure 2.56 — Total head – closed suction

2.6.3.13.3 Bowl assembly input power (P_{ba})

The horsepower delivered to the bowl assembly shaft.

2.6.3.13.4 Pump output power (P_w)

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

$$\text{(US units) } P_w = \frac{Q \times H \times s}{3960}$$

$$\text{(Metric) } P_w = \frac{Q \times H \times s}{366}$$

2.6.3.13.5 Bowl assembly output power (P_{wba})

The power imparted to the liquid by the bowl assembly. It is also referred to as the bowl assembly water horsepower.

2.6.3.14 Efficiency (η)

2.6.3.14.1 Pump efficiency (η_p)

The ratio of the pump output power (P_w) to the pump input power (P_p); that is, the ratio of the water horsepower to the brake horsepower expressed as a percent:

$$\eta_p = \frac{P_w}{P_p} \times 100$$

2.6.3.14.2 Bowl assembly efficiency (η_{ba})

This is the efficiency obtained from the bowl assembly, excluding all losses within other pump components. This is the efficiency usually shown on published performance curves.

To obtain bowl efficiency, a complete pump must be tested. Losses, both hydraulic and mechanical, attributed to test components other than the bowl assembly must be considered. Thus, we have the following considerations.

Bowl assembly head (H_{ba}) is measured as stated in Paragraph 2.6.7.12.6. Friction losses in suction pipe and strainer, if used in the test setup, must be added to the measured head. The friction loss in the column between the bowl assembly outlet and the gauge connection must also be added if significant.

Bowl assembly input power (P_{ba}) is the pump input power (P_p) minus the sum of the driveshaft bearing losses and other losses such as shaft sealing losses and thrust bearing losses, if the latter is not included in driver losses.

Therefore, bowl assembly efficiency:

$$\eta_{ba} = \frac{P_{wba}}{P_{ba}} \times 100$$

2.6.3.14.3 Overall efficiency (η_{OA})

The ratio of the energy imparted to the liquid by the pump (P_w) to the energy supplied to the driver (P_{mot}) expressed as a percent. This efficiency takes into account losses in both the pump and the driver:

$$\eta_{OA} = \frac{P_w}{P_{mot}} \times 100$$

2.6.4 Hydrostatic test

2.6.4.1 Objective

To demonstrate that the pump when subjected to hydrostatic pressure(s) will not leak or fail structurally. For purposes of this requirement, the containment of fluid means only prevention of its escape through the external surfaces of the pump, normally to the atmosphere.

2.6.4.2 Test parameters

Each part of the pump which contains liquid under pressure shall be capable of withstanding a hydrostatic test at not less than the greater of the following:

– 150% of the pressure which would occur in that part when the pump is operating at rated condition for the given application of the pump;

– 125% of the pressure which would occur in that part when the pump is operating at rated speed for a given application, but with the pump discharge valve closed.

In both instances, suction pressure must be taken into account.

– *Components or assembled pumps:* The test shall be conducted on either the liquid-containing components or the assembled pump;

– *Components:* The test shall be conducted on the liquid-containing components such as the bowls and discharge heads. Care must be taken not to impose pressure in excess of 150% of design on areas designed for lower pressure operation. Test flanges or cylinders can be used for isolating differential pressure;

– *Assembled pump:* The test shall be conducted on the entire liquid containing area of the pump but care must be taken not to impose pressure in excess of 150% of design on areas such as suction head areas;

– *Test duration:* Test pressure shall be maintained for a sufficient period of time to permit complete examination of the parts under pressure. The hydrostatic test shall be considered satisfactory when no leaks or structural failure are observed for a minimum of 5 minutes;

– *Test liquid:* Test liquid shall be water or oil having a maximum viscosity of 150 SSU (32 Cst) at test temperature;

– *Temperature:* If the part tested is to operate at a temperature at which the strength of material is below the strength of the material at room temperature, the hydrostatic test pressure shall be multiplied by a factor obtained by dividing the allowable working stress for the material at room temperature by that at operating temperature. This pressure thus obtained shall then be the minimum pressure at which hydrostatic pressure shall be performed. The data sheet shall list the actual hydrostatic test pressure.

2.6.4.3 Test procedure

Items to be tested shall have all the openings adequately sealed to allow a maximum of ten drops per minute leakage through the openings. Provisions shall be made to vent all the air at the high points on the item. The item shall be filled with the test liquid, pressured, and the test pressure shall be maintained for the duration of the test. No leakage, through the item tested shall be visible; however, leakage up to ten drops per minute through the stuffing box packing shall be permitted.

2.6.4.4 Records

Complete written or computer records shall be kept of all pertinent information and kept on file, available to the purchaser by the test facility, for two years. This information shall include:

a) Identification by model, size, serial number;

b) Test liquid;

c) Maximum allowable working pressures and temperature;

d) Hydrostatic test pressure and test duration;

e) Date of test;

f) Identity of personnel in charge.

2.6.5 Performance test

2.6.5.1 Acceptance criteria

Acceptance test tolerances apply to a specified condition point only, not to the entire performance curve, unless previously agreed to between the purchaser and the manufacturer. Testing at other than rated speed must also be mutually agreed upon, when special circumstances require such testings (see Paragraph 2.6.5.10).

Pumps must be checked for satisfactory mechanical operation during performance testing; the degree and extent of such checking is dependent upon the pump type and the contractual requirements.

2.6.5.2 Witnessing of tests

The purchaser or purchaser's designated representative may witness the test when requested by the purchaser in the purchase order.

2.6.5.3 Acceptance test tolerances

In making tests under this standard no minus tolerance or margin shall be allowed with respect to capacity, total head, or efficiency at rated or specified conditions.

Acceptance of the pump test results will be judged at rated capacity and rpm with applicable total head and efficiency as follows:

Total head	Tolerance
Under 200 ft and 2999 gpm	+ 8%, – 0
Under 200 ft and 3000 gpm and over	+ 5%, – 0
From 201 ft to 500 ft, any gpm	+ 5%, – 0
501 ft and over, any gpm	+ 3%, – 0

NOTE – Minimum efficiency at rated rpm and capacity shall be contract pump efficiency η_p.

Alternately, the pump test results may be judged at rated total head and rpm versus capacity as follows:

– Capacity tolerance at rated head, + 10, – 0%;

– Minimum efficiency at rated rpm and head shall be contract pump efficiency η_p.

It is only required to comply with either the capacity or the head tolerance. It should be noted that there will be an increase in horsepower at the rated condition when complying with plus tolerances for head or capacity at the quoted efficiency.

A minimum number of 7 test points are required.

When the test head exceeds the maximum head allowed by the acceptance criteria, but is within 8% of the rated head, then the impeller diameter may be reduced and test and calculated results submitted as a final acceptance test without further testing.

2.6.5.4 Instrumentation

Test instrumentation shall be selected so that it can provide measurements with accuracy shown in Paragraph 2.6.5.4 at BEP. Instruments need not be calibrated specifically for each test, but are to be periodically calibrated with certified records kept by the manufacturer. Description and suggested maximum period between calibration are contained in Table 2.13.

2.6.5.4.1 Fluctuation and accuracy of instruments

High-accuracy instrumentation is recommended when efficiency accuracy is of primary importance. This is usually more important on high-power consumption pumps.

It is common practice to use the actual recorded test readings from calibrated instruments for computation of efficiency (for fulfillment of the manufacturer's guarantee) and to disregard the effect of instrument accuracy.

	Acceptable fluctuation of test reading in ± %	**Required accuracy of the instrument in ± % of the specified values being observed**
Capacity	2	1.0
Differential pressure or head	2	1.0
Discharge head	2	0.5
Suction head	2	0.5
Pump power input	2	0.75
Pump speed	0.3	0.3

2.6.5.5 Test setup

This section contains general guidelines for testing to ensure accurate and repeatable test results.

The test setup may utilize, but is not limited to, the following:

1) Standard laboratory pump test mounting. This should be rigid enough to restrain the pump against reaction forces developed by flow and pressure;

2) Facility or purchaser-furnished driver. Depending on the method used to measure pump input power, driver efficiency data may be required;

3) Facility or purchaser-furnished speed-increaser/reducing unit. To accurately establish pump input power, equipment efficiency data may be required, depending on method used to measure power input;

4) Pump test configuration:

a) Open suction bowl assembly test, (see Figure 2.57). Vertical pumps are manufactured in such diverse physical configurations that, unless otherwise agreed to between purchaser and manufacturer, it is the industry practice to permit testing of the bowl assembly only for hydraulic performance. Test laboratory column, shaft, discharge head or elbow, and laboratory drivers may be used.

Such items as test pit depth limitations, discharge head, elbow physical constraints, or pump lubricants such as oil or grease contaminating laboratory water and instruments may make the test of complete units impractical.

The hydraulic and mechanical losses occurring in the pump components not tested

Table 2.13 — Recommended instrument calibration interval [3]

Capacity		**Power (continued)**	
Quantity meter		Torque bar	1 yr
Weighing tanks	1 yr	Calibrated motor	Not req'd [2]
Volumetric tank	10 yr	KW transducer	3 yr
Rate meters		Watt-amp-volt, portable	1 yr
Venturi	[1]	Watt-amp-volt, permanent	1 yr
Nozzle	[1]	Strain gauges	6 mo
Orifice plate	[1]	Transmission gears to 500 HP	10 yr
Weir	[1]	Transmission gears above 500 HP	20 yr
Turbine	1 yr	**Speed**	
Magnetic flow	1 yr	Tachometers	3 yr
Rotometer	5 yr	Eddy current drag	10 yr
Propeller	1 yr	Electronic	Not req'd [2]
Ultra-sonic	5 yr	Frequency responsive devices	
Pressure		Vibrating reed	10 yr
Bourdon tube (pressure gauge)	4 mo	Electronic	10 yr
Manometers	Not req'd	Photocell	10 yr
Dead weight tester	1 yr	Stroboscopes	5 yr
Transducers	4 mo	Torque meter (speed)	1 yr
Digital indicator	1 yr	**Temperature**	
Power		Electric	2 yr
Dynamometer w/scale	6 yr	Mercury	5 yr
Dynamometer w/load scale	6 mo		

[1] Calibration is not required unless it is suspected there are critical dimensional changes.

[2] Unless electrical or mechanical failure.

[3] Use instrument manufacturer's recommendation if shorter than listed above.

must then be added to arrive at the complete pump performance. When test facility limitations do not permit full stage testing, it is permissible to perform reduced stage tests when previously agreed to between purchaser and manufacturer. No adjustment of test results per stage for reduced stage tests shall be made;

b) Closed suction and closed loop, pump or bowl assembly performance test (see Figures 2.58 and 2.59). These types of pump test set ups are used when both NPSH and performance testing is required. The loop is typically arranged so that either vacuum or pressure can be controlled on the suction side. This test configuration is also often used when a model rather than a prototype test is performed;

c) Pump performance test, general. When a customer specifies it and it is reasonable considering test facility limitations, a complete pump performance test will be run. This is desired both for mechanical integrity checks and to accurately establish hydraulic performance. Special pump and test facility modifications may be required to test the complete pump and its driver;

5) A pit configuration that will ensure that the flow into the pump is free from swirl induced by

the installation and has a normal, symmetrical velocity distribution;

6) A suction pressure gauge, manometer, compound gauge, or pressure transducer suitable for measuring the complete range of pressures, whether positive or negative;

7) A discharge pipe with a valve or other pressure breakdown (throttling) device;

8) A discharge pressure gauge or transducer suitable for the full operating range;

9) Damping devices such as needle valves or capillary tubes to minimize pressure pulsations at the gauges;

10) A means for measuring input power to the pump or driver suitable for the power range;

11) A means for measuring pump speed, such as a revolution counter or timer, tachometer, frequency responsive device or stroboscope;

12) A means for measuring capacity, such as by weight, by volume or by rate meters;

13) Test setups for NPSH testing shall be provided with a means of lowering the suction pressure to the pump, such as a closed tank with a vacuum source or a suction throttle valve with screen (optional) and straightening vanes. In an open system (wet pit), the suction pressure may be reduced by lowering the liquid level;

14) A means for measuring the temperature of the test liquid;

15) The actual inside dimensions of the suction and discharge pipe where pressure readings are to be taken shall be determined, so that velocity head calculations can be made.

2.6.5.6 Pretest data requirements

When applicable, the following data shall be obtained prior to the test run and written for the record to be retained for two years (see sample data sheet on page 12):

1) Record of pump type, size and serial number;

2) To verify liquid properties such as viscosity and specific gravity, temperature of the liquid shall be taken before and after testing or more often when testing for NPSH or for high horsepower pumps;

3) Ambient conditions such as air temperature and barometric pressure;

4) Record of critical installation dimensions, such as pressure gauge elevation above datum, pipe internal dimensions and lengths, and liquid levels (submergence) relative to datum;

5) Record of driver data such as type, serial number, horsepower speed range, amperage, voltage and efficiency;

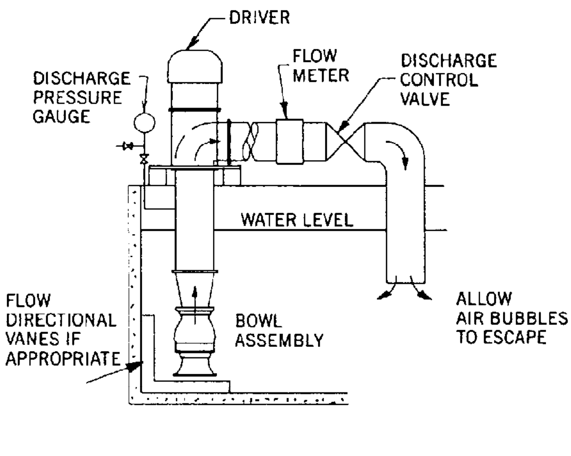


Figure 2.57 — Bowl assembly performance test – open sump

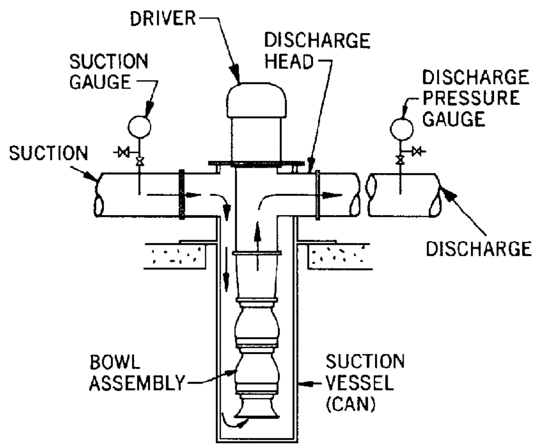


Figure 2.58 — Pump performance test – closed suction

Summary of necessary data on pumps to be tested

The following information should be furnished on pumps to be tested:

General:

1. Owner's name ____
2. Plant location ____
3. Elevation above sea level ____
4. Type of service ____

Pump:

1. Manufactured by ____
2. Manufacturer's designation ____
3. Manufacturer's serial number ____
4. Arrangement: open sump ___ can pump ___
5. Inlet: single ____ double ____
6. Number of stages ____
7. Size suction: nominal ____ in.
   (can pump) actual ____ in.
8. Size discharge: nominal ____ in.
   actual ____ in.

Intermediate transmission:

1. Manufactured by ____
2. Type ____
3. Serial number ____
4. Speed ratio ____
5. Efficiency ____

Driver:

1. Manufactured by ____
2. Serial number ____
3. Type: motor ____ turbine ____ other ____
4. Rated horsepower ____
5. Rated speed ____
6. Characteristics (voltage, frequency, etc.) ____
7. Calibration data ____

Specifying rated conditions

The following information is necessary in specifying rated conditions:

1. Liquid pumped (water, oil, etc.) ____
2. Specific weight ____
3. Viscosity at pumping temperature ____
4. Temperature ____
5. Vapor pressure ____
6. Capacity ____
7. Total suction lift (h_s) ____
8. Total suction head (h_s) ____
9. Net positive suction head required (NPSHR) ____
10. Total discharge head (h_d) ____
11. Total head (H) ____
12. Output power (P_w) ____
13. Efficiency (η_p) ____
14. Input power (P_p) ____
15. Speed ____

Test information

Test information should be listed substantially as follows:

General:

1. Where tested ____
2. Date ____
3. Tested by ____
4. Test witnessed by ____

Capacity:

1. Method of measurement ____
2. Meter—make and serial number ____
3. Calibration data ____

Head:

1. Suction gauge—make and serial number ____
2. Calibration data ____
3. Discharge gauge—make and serial number ____
4. Calibration data ____

Power:

1. Method of measurement ____
2. Make and serial number of instrument ____
3. Calibration data ____

Speed:

1. Method of measurement ____
2. Make and serial number of instrument ____
3. Calibration data ____

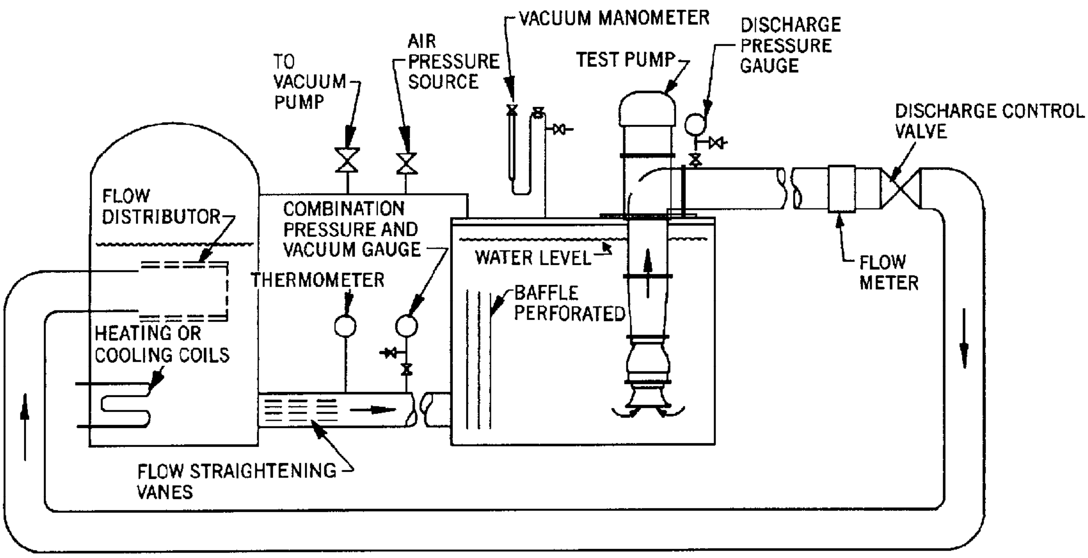


Figure 2.59 — Pump performance test – closed loop

6) Record of auxiliary equipment such as vibration monitors, temperature sensors, low- or high-pressure monitors, leakage detectors, alarms;

7) Instrument calibration records and correction factors in accordance with the calibration section of this standard;

8) Identity of principal test personnel;

9) Dimension of areas where pressure readings are to be taken for accurate determinations of the velocity head.

2.6.5.7 Records

Complete written or computer records shall be kept of all information relevant to a test and retained on file, available to the purchaser by the test facility for two years.

2.6.5.7.1 Introduction

The manufacturer's serial number or other appropriate means of identification of each pump tested shall be recorded, along with impeller information such as diameter and vane filing.

While these records apply to the complete unit including the driver, this standard applies only to the test of the pump.

2.6.5.8 Calculations

2.6.5.8.1 Calculations of total suction head (h_s)

For a closed system (can pump):

$$h_s = h_{gs} + Z_s + \frac{v_s^2}{2g} \text{ (see Figure 2.54)}$$

For an open system (pump in open pit):

$$h_s = Z_w \text{ (see Figure 2.53)}$$

In a pit application, the entrance losses to the pump are charged to the pump. Also, the average velocity head of the pit flow is typically small enough to be neglected.

2.6.5.8.2 Calculation of total discharge head (h_d)

For closed suction (can pump):

$$h_d = h_{gd} + Z_d + \frac{v_d^2}{2g} \text{ (see Figure 2.56)}$$

For open suction; wet pit pump:

$$h_d = h_{gd} + Z_d + \frac{V_d^2}{2g} \text{ (see Figure 2.55)}$$

The discharge pressure gauge is located downstream of the pump's discharge head, and all internal pump hydraulic losses are therefore included.

2.6.5.8.3 Calculation of bowl assembly total head (H_{ba})

$$H_{ba} = h_{gd} + (Z_d - Z_w) + \frac{V_d^2}{2g} \text{ (see Figure 2.73)}$$

The discharge gauge pressure tap is located a minimum of 2 diameters above the bowl assembly, thereby excluding column and discharge head losses from the readings.

2.6.5.8.4 Calculation of total head (H)

$$H = h_d - {}^*h_s$$

Total head is the algebraic difference between total discharge head and total suction head.

$^*h_s = Z_w$ on open suction test.

2.6.5.8.5 Calculation of pump input power (P_p)

The pump input power (P_p), when measured by transmission dynamometer, is calculated from the following formula:

$$\text{(US units) } P_p = \frac{2\pi LWn}{33{,}000} = \frac{n\tau}{5250}$$

$$\text{(Metric units) } P_p = \frac{n\tau}{60{,}000}$$

Where:

L = Length of lever arm in ft. (m);

W = Net force in lbs. (N);

n = Speed in rpm;

τ = Torque in pound feet (N•m).

The input power to an electric motor in horsepower is given by:

$$P_{mot} = \frac{kW}{0.746}$$

Where kW = Electrical input power in kilowatts.

The input power to a pump driven by an electric motor is:

$$P_p = P_{mot} \times \eta_{mot}$$

Where η_{mot} = calibrated efficiency of motor.

The input power (P_{ba}) to a pump bowl assembly is:

$$P_{ba} = P_{mot} \times \frac{\eta_{mot}}{100} \times \frac{\eta_{im}}{100} - P_l$$

Where:

P_l = the driveshaft bearing and thrust bearing losses;

η_{im} = intermediate mechanism efficiency, which includes gear and variable speed drives.

2.6.5.8.6 Calculations of pump output power (P_w)

The pump output power (P_w) is computed from the following formula:

a) US units:

$$P_w = \frac{\text{pounds of liquid pumped per minute} \times \text{total head in feet of liquid}}{33{,}000}$$

For water at 68°F, the specific weight is 62.3 pounds per cubic foot, then:

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

For liquids with different specific weight, or water at specific weight other than 62.3 lbs/ft^3, the above formula must be corrected using the applicable specific gravity(s) as follows:

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

b) Metric units:

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

2.6.5.8.7 Calculation of efficiency (η)

Testing can be performed to establish bowl assembly efficiency, pump efficiency or overall pump/driver efficiency.

2.6.5.8.7.1 Bowl assembly efficiency (η_{ba})

This efficiency value excludes all losses outside the bowl assembly proper:

(US units) $\eta_{ba} = \dfrac{Q \times H_{ba} \times s}{3960 \times P_{ba}} \times 100$

(Metric units) $\eta_{ba} = \dfrac{Q \times H_{ba} \times s}{366 \times P_{ba}} \times 100$

NOTE – Refer to the *Hydraulic Institute Engineering Data Book* for column pipe friction losses and driveshaft bearing losses.

2.6.5.8.7.2 Pump efficiency (η_p)

This efficiency value excludes losses in the primary and secondary driver but includes hydraulic losses through suction piping, strainer, bowl assembly column pipe, and surface discharge head or discharge elbow, as well as mechanical losses in driveshaft bearings and the shaft seal:

$$\eta_p = \frac{P_w}{P_p} \times 100$$

2.6.5.8.7.3 Overall efficiency (η_{OA})

This is pump efficiency reduced by losses such as, but not limited to, driver losses including thrust bearing losses and gear losses where applicable.

$\eta_{OA} = \eta_p \times$ driver efficiency x gear efficiency less efficiency loss from thrust bearing (if applicable).

For calculation purposes, all efficiency values must be in decimal form.

Vertical motor efficiencies generally do not include thrust bearing losses due to thrust load.

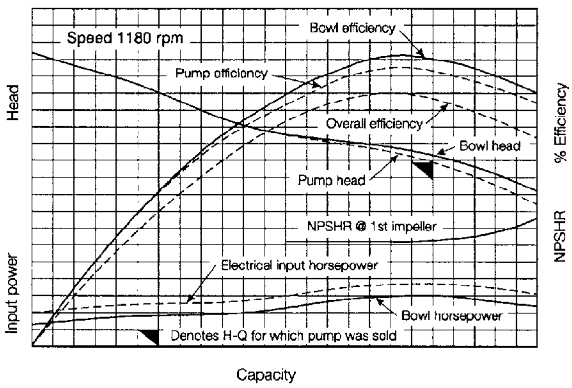


Figure 2.60 — Pump performance curves

The overall efficiency of a motor-driven unit is calculated by:

$$\eta_{OA} = \frac{P_w}{P_{mot}} \times 100 = \eta_p \times \eta_{mot}$$

2.6.5.9 Plotting results

The head, efficiency and horsepower are plotted as ordinates on the same sheet with capacity as the abscissa (see Figure 2.60). The bowl assembly values are commonly plotted and correspond with the manufacturer's published performance curves. The curves must be clearly labeled as to whether they apply to the bowl assembly, the complete pump, or the complete unit (pump and driver).

2.6.5.10 Test at non-rated speed

2.6.5.10.1 Test of full-sized pumps at reduced speed

For reduced-speed tests, the relative power loss in bearings and stuffing box friction may be greater, and the hydraulic friction losses may also be relatively larger due to reduction in the Reynolds number. This effect may be significant in small pumps. These factors must, therefore, be considered in determining an acceptable speed, which should be mutually agreed upon prior to testing.

In order to establish test conditions, the following relationships shall be used for determining head and capacity from the rated (specified) point:

$$\frac{n_1}{n_2} = \frac{Q_1}{Q_2} = \left(\frac{H_1}{H_2}\right)^{1/2} = \left(\frac{P_1}{P_2}\right)^{1/3} = \left(\frac{NPSHR_1}{NPSHR_2}\right)^{1/2}$$

Where:

n_1 = Test speed in rpm;

n_2 = Rated speed in rpm;

Q_1 = Test capacity;

Q_2 = Rated capacity;

H_1 = Test head;

H_2 = Rated head;

P_1 = Power on test;

P_2 = Power on installation;

$NPSHR_1$ = NPSHR on test;

$NPSHR_2$ = NPSHR for installation.

EXAMPLE: A four-stage bowl assembly is rated at 400 gpm against a bowl head of 240 feet, NPSHR of 14 feet and running at 2950 rpm (50 Hz frequency). If the factory only has 60 Hz power available, the test will be run at a reduced speed of 1770 rpm. What head, capacity and NPSHR should the factory test pump produce at reduced speed to meet the rated conditions?

Applying the relationships given above, the equivalent head for the factory test is:

$$H_1 = H_2\left(\frac{n_1}{n_2}\right)^{1/2} = 240\left(\frac{1770}{2950}\right)^2 = 86.4 \text{ feet}$$

The equivalent capacity for the factory test is:

$$Q_1 = Q_2\frac{n_1}{n_2} = 400\,\frac{1770}{2950} = 240 \text{ gpm}$$

The NPSH required for the factory test is:

$$NPSHR_1 = NPSHR_2\left(\frac{n_1}{n_2}\right)^2 = 14\left(\frac{1770}{2950}\right)^2 = 5 \text{ feet}$$

Note that specific speed is a pump characteristic unaffected by operating speed.

2.6.5.10.2 Test of full-sized pumps at increased speed

Under unusual circumstances, it may be desirable to carry out tests at higher speeds than specified for the installation. This may be due, for example, to the limitations of available prime movers or correct electrical frequency. In this case, if such tests do not exceed safe operating limits of the pump, all of the above considerations apply.

Cases may arise in which the limitations of the factory test facilities may preclude establishing the required suction lift to comply with the installation NPSH. In such cases, the desired NPSHR can be obtained by increasing the speed and the pumping head instead of by a reduction in suction head or an increase in suction lift.

2.6.5.10.3 Correcting for test speed variations

The pump test speed will vary with operating conditions.

For purposes of plotting the test results, capacity, head and power shall be corrected from the values at test speed to the value of rated speed for the pump. The corrections are made using the same relationships as shown in Paragraph 2.6.5.10.1. However, when the pump is tested with the purchaser's motor, the performance shall be plotted at actual test speed.

2.6.5.10.4 Temperature variations

Variations in temperature of the liquid pumped cause changes in specific weight and viscosity, with resultant changes in pump performance.

A reduction in specific weight, as caused by an increase in temperature, results in a directly proportional reduction in output power (see Paragraph 2.6.3.13 Power) and in input power; therefore, the efficiency is not changed.

Reduced viscosity of water due to a temperature increase will impact efficiency. For pumps in the lower range of specific speed, typically below 1500, reduced viscosity will:

- Increase internal leakage losses;
- Reduce disc friction losses;
- Reduce hydraulic skin friction losses.

The net effect of a reduction in viscosity due to higher temperature will depend on specific speed and on the design details of the pump. Where substantiating data is available, consideration may be given to adjusting the performance data from a cold water test to hot water operating conditions on the basis of the following formula:

$$\eta_{ot} = 1 - (1 - \eta_t)\left(\frac{\nu_{ot}}{\nu_t}\right)^x$$

Where:

η_{ot} = Efficiency at operating temperature, decimal value;

η_t = Efficiency at test temperature, decimal value;

ν_{ot} = Kinematic viscosity at operating temperature;

ν_t = Kinematic viscosity at test temperature;

x = Exponent to be established by manufacturer's data based on the pump type in question (approx. range: .05 to .1) .1 selected for example below.

EXAMPLE: A test on water at 100°F resulted in an efficiency of 80 percent. What will be the projected efficiency at 350°F?

$$\eta_{ot} = 1 - (1 - \eta_t)\left(\frac{\nu_{ot}}{\nu_t}\right)^x$$

$$\eta_{ot} = 1 - (1 - .80)\left(\frac{.00000185}{.0000076}\right)^{0.1}$$

$$\eta_{ot} = .826 = 82.6\%$$

2.6.5.10.5 Specific weight variations

If the test is run with a liquid of different specific weight from that of the field installation, there will be a revision in required input power, which will be determined as follows:

$$(P_p)_2 = (P_p)_1 \times \frac{\gamma_2}{\gamma_1}$$

There is no change in efficiency.

2.6.5.10.6 Viscosity variations

Viscosity has a significant effect on pump performance with respect to head, capacity, efficiency and brake horsepower. Pumps for viscous service, which are tested on water, will require corrections to approximate the viscous performance. See the Design and Application Section of the *Hydraulic Institute Vertical Pump Standards HI 2.1 – 2.5.*

2.6.5.10.7 Solids in suspension

Solids in suspension affect the operating conditions of the pump, depending on the percentage and nature of the solids. Corrections for solids handling are not part of this test standard.

2.6.5.11 Report of test

Parties to the test shall be furnished a copy of the performance curve at constant speed, as drawn in accordance with Paragraph 2.6.5.9. When specifically requested by the purchaser, additional test documentation shall be made available.

2.6.6 Net positive suction head required test

2.6.6.1 Objective

To determine the NPSH required (NPSHR) by the pump.

2.6.6.2 Test arrangement

Four typical test setups are shown for determining the NPSHR characteristics of pumps.

In the first arrangement, shown in Figure 2.61, the pump is supplied from a sump through a throttle valve, which is followed by a section of pipe

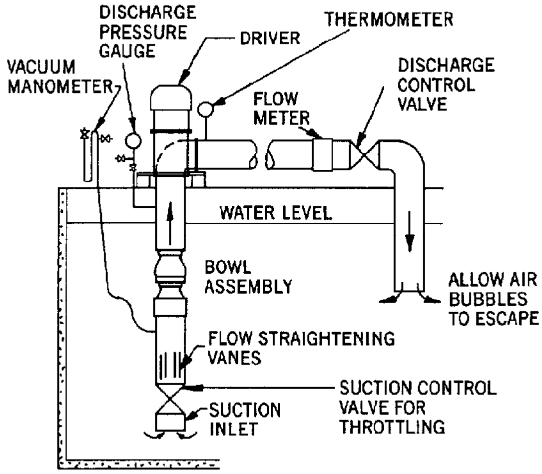


Figure 2.61 — Suction throttling NPSH test constant sump level

containing a screen and straightening vanes. This minimizes the turbulence produced by the throttle valve and makes possible an acceptable reading of suction head at the pump inlet.

This arrangement usually is satisfactory for NPSHR greater than 10 feet, although the turbulence at the throttle valve tends to accelerate the release of dissolved air or gas from the liquid at reduced pressure. As a result, this arrangement typically will indicate a higher NPSHR than other test methods.

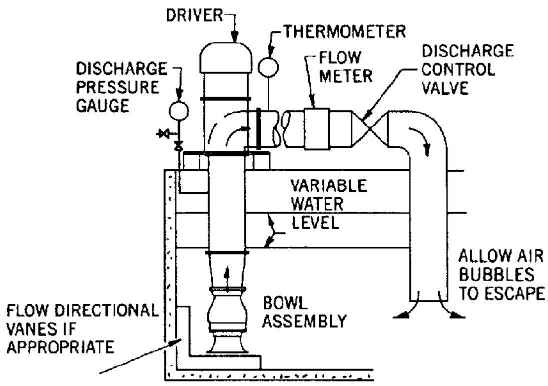


Figure 2.62 — Level control NPSH test

In the second arrangement, Figure 2.62, the pump is supplied from a sump in which the liquid level can be varied to establish the desired suction head. This arrangement more accurately reflects typical operating conditions. This arrangement is suitable for testing with suction head in

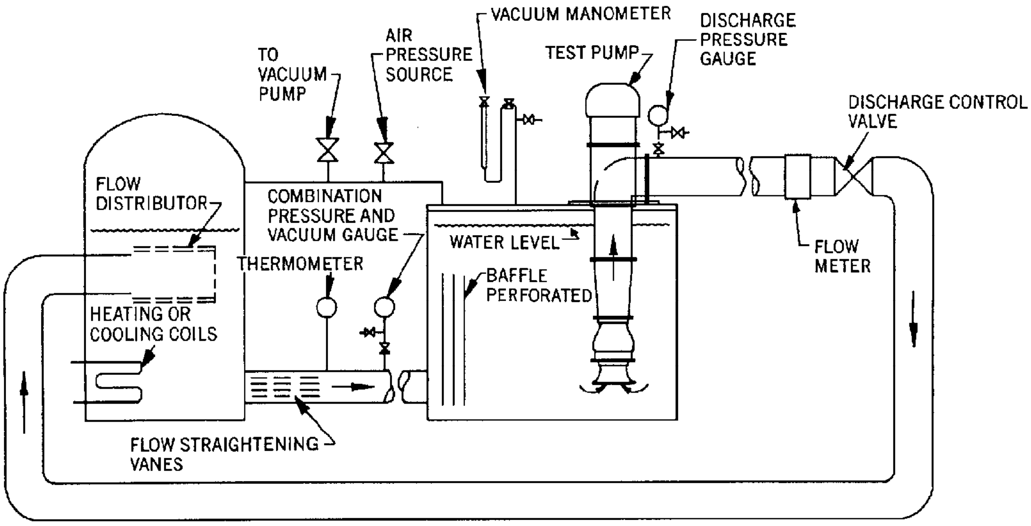


Figure 2.63 — Closed loop NPSH test

excess of atmospheric pressure. Care must be taken to prevent vortexing when the liquid level is varied.

In the third arrangement, Figure 2.63, the pump is supplied from a closed tank in which the level is held constant and the suction lift or suction head is adjusted by varying the air or gas pressure over the liquid, the temperature of the liquid, or both.

This arrangement tends to strip the liquid of dissolved air or gas. It gives a more accurate measurement of the pump performance and is not influenced by the release of air or gas at pressures below the vapor pressure of the liquid. This arrangement typically duplicates service conditions where a pump takes its supply from a closed vessel with the liquid at or near its vapor pressure.

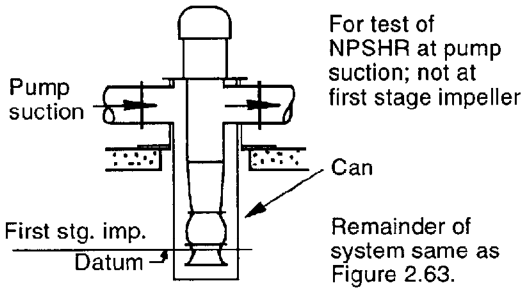


Figure 2.64 — Closed loop NPSH test - alternate arrangement for can pump

This arrangement is more effective for high specific speed mix flow and propeller pumps.

The fourth arrangement, Figure 2.64, shows a typical NPSHR test for a can pump. This arrangement is used when the suction condition approaches zero ft. NPSHA at the suction centerline (datum) elevation. The first stage of the bowl assembly is located in a can or tank, in which the pressure can be regulated and reduced to the desired level to meet the test criteria. The distance from the suction centerline elevation to the first stage impeller centerline is adjusted by the column length to provide sufficient head (NPSHA) to operate the pump. The test results must, when applicable, reference the difference between the pump's datum elevation and the elevation at which the NPSHA is specified in the application.

Other precautions to be taken in test arrangements are:

- *Liquid*: Water shall be used as the test liquid;
- *Aeration*: Fluid aeration shall be minimized by taking the following precautions:
  - Intake structure designed to avoid vortexing. (See *Hydraulic Institute Vertical Pump Standards HI 2.1-2.5, Design and Application Section)*;
  - Submerged lines when pressure is below atmospheric, if practical;
  - Reservoir sized for long retention to

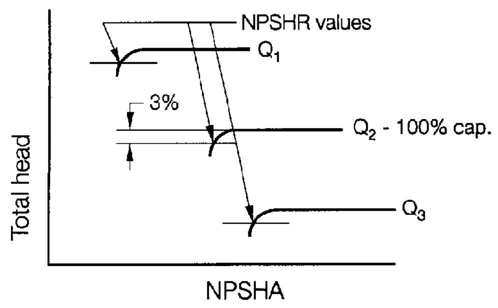
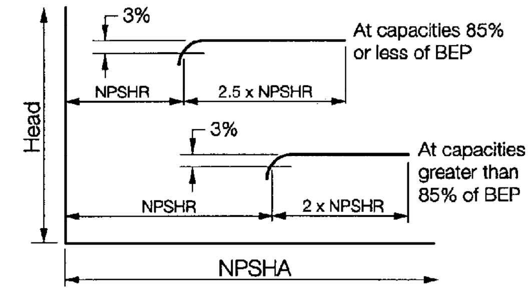


Figure 2.65 — NPSH at constant capacity



NPSH test at constant capacity
Recommended NPSHA range for NPSHR test when no previous data on pumps non-cavitating performance is available.

Figure 2.66 — NPSH at constant capacity

allow air to escape. Inlet to sump located to prevent vortexing;

– Reservoir baffles to isolate outlet from inlet line;

– Tight pipe joints and quenched stuffing boxes to prevent air leakage into the system.

For large pumps, cavitation testing may, for practical reasons, be performed on models. Reference is made to the section on model testing in Paragraph 2.6.12.

2.6.6.3 Test procedure

Unless otherwise agreed between the purchaser and the manufacturer, the test shall be run for the range of ± 20% of rated capacity with 3 test capacities to determine the NPSH required.

The NPSHR of a pump can be determined by one of the following procedures:

The preferred method is to run the pump at constant capacity and speed with the suction head varied. As NPSHA is reduced, and the corresponding pump head plotted for each NPSH value, a point is reached where the head curve breaks away for the straight line trend (see Figure 2.65) indicating a deterioration in pump performance. The 3% head drop is the standard to determine NPSHR (NPSH required). For multistage pumps, the 3% applies to the first stage only. The test is repeated at various flow rates and the total head plotted against NPSHA. Figure 2.65 shows the results typical of tests at capacities both above and below pump design flow.

A second method for determining the cavitation characteristics is to hold the speed and suction head (h_s) constant and vary the capacity. The test is repeated for various suction head values and the total head plotted against capacity. Such tests will result in a family of curves, as shown in Figure 2.67. Where the pump head for any suction head (h_s) breaks away from the normal head-capacity curve by 3%, NPSHR (NPSH required) is established.

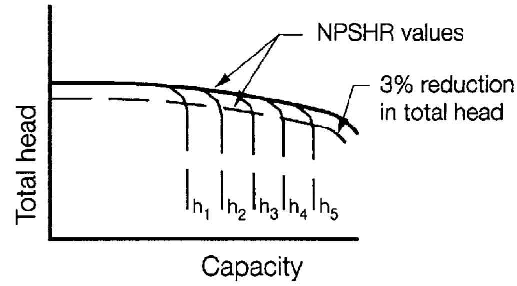


Figure 2.67 — NPSH at varying capacity

Accurate determination of the start of cavitation, and the cavitation point, requires careful control of all factors which influence the operation of the pump. A minimum of five test points bracketing the point of change must be taken to determine when the performance starts to deviate from that with excess NPSHA. Any change in performance—either a drop in head or power at a given capacity or a change in sound or vibration—may indicate the presence of cavitation. With the difficulty in determining just when the change starts, a drop in head of 3 % at a given capacity or NPSH is generally accepted as evidence that cavitation is present. The NPSH at this point is defined as the NPSH Required (NPSHR). Note that for multistage pumps, the 3% drop is applied to the first stage head.

The NPSHA value required to properly establish the non-cavitating performance of a pump must be determined from prior full-scale or model tests of the specific pump in question. If such prior tests are not available, then an NPSHA value of at least twice the predicted NPSHR for capacities greater than 85% of BEP, and at least two and one half times the predicted NPSHR for capacities below 85% of BEP, is recommended for assurance that non-cavitating conditions exist (see Figure 2.66).

Tests performed to establish NPSHR for a specific pump must begin with a non-cavitating NPSHA value in line with the recommendations above.

When testing with water, an accurate temperature measurement usually is sufficient to establish the vapor pressure. However, the degree of aeration of the water may have a considerable influence on performance. Consistent results are more readily obtained when water is deaerated.

If the pump is of multistage design, it is preferable to test the first stage separately, so that the drop in head can be measured more accurately.

Correction to rated speed for net positive suction head (NPSH):

$$NPSHR_2 = \left(\frac{n_2}{n_1}\right)^2 \text{ x } NPSHR_1$$

$$\text{and} \quad Q_2 = \frac{n_2}{n_1} \text{ x } Q_1$$

Where:

$NPSHR_1$ = Net positive suction head at test speed;

$NPSHR_2$ = Net positive suction head at rated speed;

n_1 = Test speed in rpm;

n_2 = Rated speed in rpm;

Q_1 = Test capacity;

Q_2 = Capacity at rated speed.

NPSHR: Experimental deviation from the square law.

The affinity relationships define the manner in which head, capacity, horsepower and NPSHR vary in vertical pumps with respect to speed changes. If a pump operates at or near its cavitation limit, other factors also have an effect, and NPSHR value may not vary exactly as the square of the speed. Some of these factors are: thermodynamic effect of the vapor pressure of the fluid, change in surface tension, and test differences due to the relative air content of the liquid.

If the manufacturer can demonstrate from tests that, with a given pump under particular conditions, an exponent different than the square of the speed exists, then such exponent may be recognized and used accordingly.

2.6.6.4 Suction conditions

The suction lift or suction head is to be measured as specified in Paragraph 2.6.3.12.

For factory performance testing the exact value of the NPSH available is unimportant, as long as it has been established that the NPSHA is well in excess of the NPSH required by the pump throughout the test range.

The net positive suction head available (NPSHA) is the total suction head in feet of liquid absolute, determined at the first stage impeller eye (datum), less the absolute vapor pressure in feet of the liquid pumped:

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

Where:

h_{sa} = Total suction head in feet absolute = $h_{atm} + h_s$;

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

$$\text{or} \quad NPSHA = \frac{144}{\gamma}(p_{atm} - p_{vp}) + h$$

For pumps mounted in a suction barrel (can), the hydraulic losses from the suction nozzle to the impeller inlet must be taken into account by the manufacturer in establishing the NPSHA at the first stage impeller eye.

2.6.6.5 Records

Complete written or computer records shall be kept by the test facility of all data relevant to the NPSH test for a minimum of two years. (See sample data sheet on page 12).

These records must include:

1) Specified NPSHR/NPSHA;

2) Water levels above first stage impeller datum;

3) Distance from first stage impeller datum to suction gauge centerline;

4) Inside diameter of pipe at location of suction pressure tap;

5) Observed data (each run): water temperature, suction pressure, shaft speed, discharge pressure, capacity;

6) Type of test setup;

7) Type of flow meter and calibration;

8) Type, number and calibration of pressure gauges;

9) Any abnormal observation (noise, vibration, etc.);

10) Type and serial number of pump and driver;

11) Date of test and person in charge.

2.6.6.6 Report of test

All parties to the test shall be furnished a copy of the NPSHR curve or curves, as described in Paragraph 2.6.6.3.

2.6.7 Measurement of capacity

2.6.7.1 Introduction

Any flow measuring system may be used for measuring pump capacity. However, it must be installed so that the entire flow passing through the pump also passes through the instrument section, and the instrument can measure capacity with an accuracy of 1.5% at BEP.

Capacity instruments are classified into two functional groups. One group primarily measures batch quantity, and the other primarily measures rate of flow.

2.6.7.2 Capacity measurement by weight

Measurement of capacity by weight depends upon the accuracy of the scales used and the accuracy of the measurement of time. A certification of scales shall become part of the test record, or in the absence of certification, the scales shall be calibrated with standard weights before or after test. Time interval for the collection period shall be measured to an accuracy of *one-quarter of one* percent.

2.6.7.3 Capacity measurement by volume

This is done by measuring the change in volume of a tank or reservoir during a measured period of time. The tank or reservoir can be located on the inlet or discharge side of the pump, and all flow into or out of the tank or reservoir must pass through the pump.

In establishing reservoir volume by linear measurements, considerations shall be given to the geometric regularity (flatness, parallelism, roundness, etc.) of the reservoir surfaces, to dimensional changes due to thermal expansion or contraction, or to deflection resulting from hydrostatic pressure of the liquid.

Liquid levels shall be measured by means such as hook gauges, floats and vertical or inclined gauge glasses.

In some locations and under some circumstances, evaporation and loss of liquid by spray may be significant and may be greater than the effects of thermal expansion or contraction. Allowance must be made for such loss or the loss prevented.

2.6.7.4 Capacity measurement by head type rate meters

This is done by introducing a reduced area in the flow stream, which results in a reduction in gauge head as the velocity is increased. The gauge head differential is measured and used to determine the capacity. The meters discussed in Paragraphs 2.6.7.4.1, 2.6.7.4.2 and 2.6.7.4.3 use this principle.

Meters falling within this classification, and acceptable for capacity determination under this standard, when used as prescribed herein, are venturis, nozzles and orifice plates.

For any such meter, compliance with this standard requires that a certified curve showing the calibration of the meter shall be obtained from the calibrating agency. This certification must state the method used in calibration and whether the meter itself was calibrated, or whether calibration was obtained from an exact duplicate.

When a flow meter is used on the discharge, it is preferable to install it in the high-pressure section between the pump and the pressure breakdown valve. If the working pressure of the meter is lower than the pump discharge pressure at shutoff, it may be installed downstream of the pressure breakdown valve, with a back pressure valve located downstream of the flow meter to insure that the pressure will stay above vapor pressure during operation and be free of cavitation in the high-velocity section of the meter.

These precautions are stipulated to assure uniform flow velocity within ± 20% at the meter inlet and stable flow at the downstream pressure taps. If there is a question as to whether or not

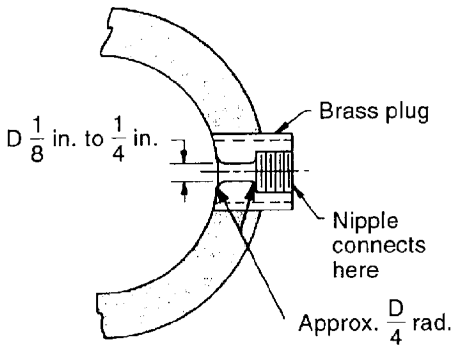


Figure 2.68 — Pressure tap opening

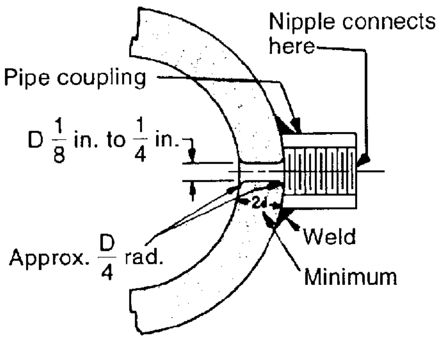


Figure 2.69 — Welded-on pressure tap opening

uniform flow has been obtained, it shall be checked by a velocity head traverse of the pipe immediately preceding the meter to assure symmetrical velocity distribution within the pipe.

The pipe for one diameter preceding the upstream pressure taps shall be free from tubercles or other surface imperfections which would establish a local disturbance in line with these openings. Pressure tap opening shall be flush with the interior of the pipe or meter element as appropriate and shall be free of burrs (see Figures 2.68 and 2.69).

Table 2.14 — Straight pipe required following any fitting before venturi meter in diameters of pipe

Meter ratio β (throat to inlet diameter)	0.4	0.5	0.6	0.7	0.8
One standard short radius elbow	1	2	3	4	6
Two elbows in same plane	2	3	4	6	8
Two elbows in planes at 90 degrees and with straightening vanes	2	3	4	5	7
Standard C.I. flanged reducer	2	5	7.5	10	13
Standard C.I. flanged increaser	1	2	3	4.5	6
Globe valve — with straightening vanes	2	4	6	9	12
Gate valve — 0.2 open	2	4	6	9	12
Gate valve — 0.5 open	2	3	4	6	8
Gate valve — full open	0	0.5	1	2	3

2.6.7.4.1 Capacity measurement by venturi meter

To insure accurate results in the measurement of capacities with venturi meters, certain minimum lengths of straight pipe are required upstream of the meter. Table 2.14 shows these minimum lengths, expressed in terms of pipe diameters.

2.6.7.4.2 Capacity measurement by nozzles

To insure accurate results in the measurement of capacities with nozzle type meters, a length of straight pipe is required preceding and following the nozzle. Tables 2.15 and 2.16 show the length of straight pipe required.

NOTE – A centrifugal pump pumping directly into a venturi meter should have at least 10 diameters of straight pipe between it and the meter.

2.6.7.4.3 Capacity measurement by thin, square-edged orifice plate

Whenever possible, the orifice plate should be calibrated in place in the piping system by weight or volume. When this is not possible, a certified curve showing the calibration of the orifice plate shall be obtained. This certification shall conform to requirements given in Paragraph 2.6.7.4, and shall, in addition, indicate the exact location and

Table 2.15 — Straight pipe required following any fitting before nozzle or orifice plate meter in diameters of pipe

Meter ratio β (throat to inlet diameter)	0.2	0.3	0.4	0.5	0.6	0.7	0.8
Tee or wye within line flow	6	6	6.5	7	8.5	10.5	14
One elbow, branch flow through tee or wye, or flow from drum or separator	6	6	6.5	7	9	13	20.5
Globe valve — wide open	9	9	9.5	10.5	13	15	21
Gate valve — wide open	6	6	6	6	7.5	9.5	13.5
Two or more short radius elbows or bends in the same plane	7.5	7.5	8.5	10.5	13.5	18	25
Two or more long radius elbows or bends in the same plane	6	6	6.5	8	11	16	23
Two short radius elbows or bends in different planes	14.5	16	17.5	20.5	24.5	30	40
Two long radius elbows or bends in different planes	7	8	10	12	16	22	33

NOTE – A centrifugal pump pumping directly into a nozzle or orifice should have at least 10 diameters of straight pipe between it and the meter.

size of pressure taps, which are then to be duplicated in the test installation.

To insure accurate results in the measurement of capacities with orifice type meters, a length of straight pipe is required preceding and following the orifice plate. Tables 2.15 and 2.16 show the length of straight pipe required, expressed in terms of equivalent pipe diameters.

2.6.7.5 Capacity measurement by weirs

This is done in open channel flow by allowing the liquid to cascade over a dam through a sharp-crested contraction in the dam, which results in an increase in velocity at the contraction. The drop in liquid level at the contraction is measured and used to determine capacity.

The rectangular sharp-crested weir with smooth vertical crest wall, complete crest contraction, free over-fall and with end contraction suppressed is acceptable for capacity determination under this standard. It may be used for either factory or field testing.

For a detailed discussion of weirs, their construction, installation and operation, the user is referred to *Fluid Meters, Their Theory and Application*, a report of the ASME Research committee on fluid meters.

Table 2.16 — Straight pipe required following downstream pressure tap of a nozzle or orifice plate meter before any fitting in diameters of pipe

Meter ratio β (throat to inlet diameter)	0.2	0.3	0.4	0.5	0.6	0.7	0.8
Gate valve — wide open	0	0	0	0	0	0	0
Wye	0	0	0	0	0	0	4
Tee	0	0	0	0	0	3.5	4
Expansion joint	0	0	0	0	0	3.5	4
45-degree elbow	0	0	0	0	3.5	3.5	4
Long radius elbow or bend	2	2.5	2.5	3	3.5	3.5	4
Regulators, control valves, and partly throttled gate valves	6	6	6	6	6	6	6

2.6.7.6 Capacity measurement by pitot tubes

A pitot tube is a double tube, one within the other. Capacity is measured by inserting the tube so that it points into the flow stream. The inner tube measures the velocity head and gauge head of the liquid, and the outer tube with holes in the outer wall measures gauge head only. The head differential is measured and used to determine the the velocity head which in turn determines capacity.

Where it is impossible to employ one of the methods described above, the pitot tube is often used. When the flow conditions are steady during the time required to make a traverse, that is, with variations less than ± 0.5%, the flow may be determined with a fair degree of accuracy.

The procedure set forth in the *ANSI/ASME PTC 18.1—1978 Pumping Mode of Pump/Turbines* is recommended.

2.6.7.7 Other methods of capacity measurement

When the methods of capacity measurement described above are not applicable, there are other methods not included in this standard which may be utilized, provided the accuracy of the instrument as described in Paragraph 2.6.5.4.1 can be demonstrated.

2.6.8 Head — measurement

The units of head and the definition of total head and its component parts are covered in Paragraph 2.6.3.12.

It is important that steady flow conditions exist at the point of instrument connection. Pressure and head measurements should therefore be taken on a section of pipe without directional changes and with constant cross section. A minimum length of straight pipe equaling five (5) diameters upstream and two (2) diameters downstream from the instrument connections must be provided, following any curved member, valve or other obstruction. For bowl assembly tests, the pressure tap in the column pipe shall be located a minimum of two pipe diameters downstream from the bowl or concentric increaser (see Figure 2.57).

If the pipe friction loss between the pump discharge flange or the bowl assembly outlet and the point of instrument connection exceeds 0.1% of the pump head, it shall be added to the measured total discharge head (h_d). Similarly, if the friction loss between the suction nozzle and the point of instrument connection exceeds 0.1% of the pump head, then this loss shall be subtracted from the measured total suction head (h_s). The friction factor used for the calculation should be based on the appropriate roughness ratios for the actual pipe section. (See Paragraph 2.6.3.12 for definition of the total discharge head and total suction head.)

For such meters, compliance with this standard requires that a certified curve showing the calibration of the meter must be obtained. This certification must state the method used in calibration and whether the meter itself was calibrated, or whether calibration was obtained from an exact duplicate.

The inside wall of the water passage shall be smooth and of unvarying cross section. For a distance of at least 12 inches preceding the opening, all protrusions and roughness shall be removed with a file or emery cloth.

The opening shall be of a diameter from 1/8 inch to 1/4 inch (3 to 6 mm) and of a length equal to twice the diameter.

The edges of the opening shall be provided with a suitable radius tangential to the wall of the water passage and shall be free from burrs or irregularities. Figures 2.68 and 2.69 show suggested arrangements of taps in conformity with the above.

A single tap connection (Figure 2.70) is used under normal conditions when the test arrangement is in compliance with this standard.

Multiple tap connections (Figure 2.71) are used where abnormal velocity profiles are suspected or conditions preclude compliance with test arrangements in this standard.

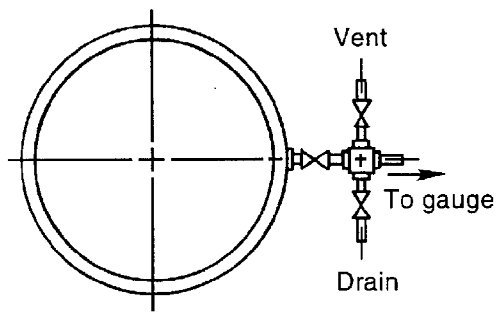


Figure 2.70 — Single tap connection

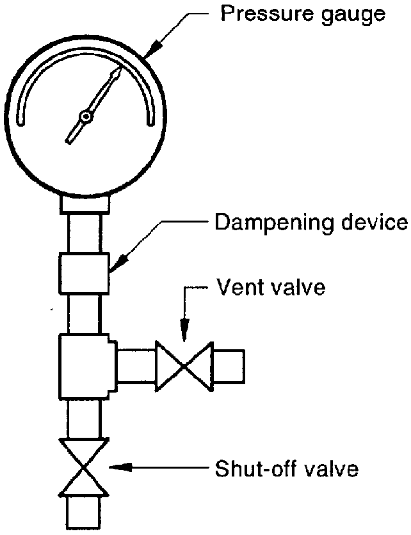


Figure 2.71 — Gauge valve arrangement

Where multiple taps are used, separate connections, properly valved, shall be made. As an alternative, separate instruments can be provided.

Multiple taps shall not be connected to a common header for the head measuring instrument if the pressure difference between any two taps is more than 1%. In this case, each tap shall be measured separately and averaged.

All connections and lines from the orifice tap shall be free of fluid leakage and as short and direct as possible. For the dry-tube type of lines, suitable

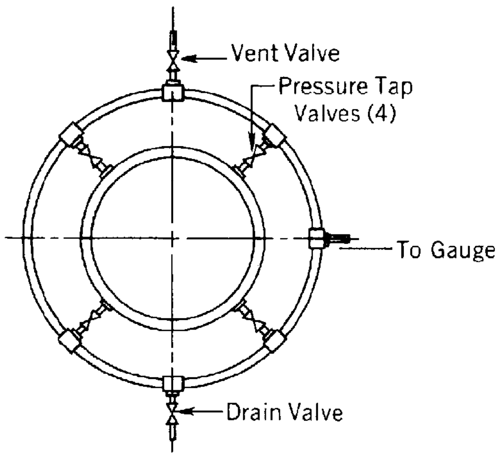


Figure 2.72 — Loop manifold connecting pressure taps

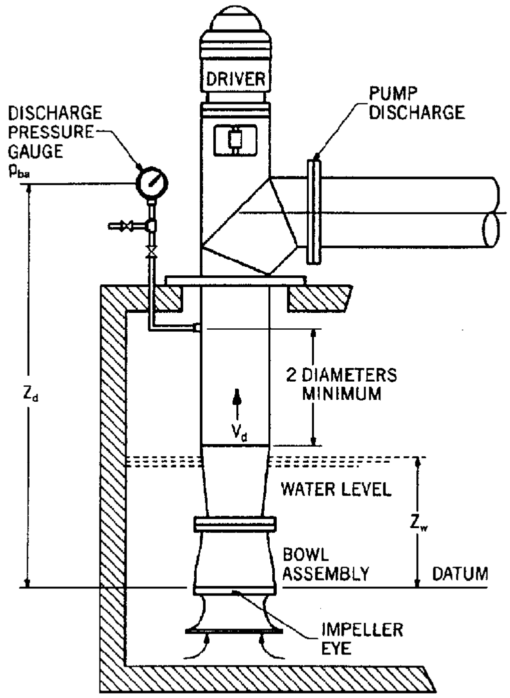


Figure 2.73 — Bowl assembly head measurement

drain pots shall be provided, including a loop of sufficient height to keep the pumped liquid from entering the lines. For the wet-tube type of lines, cocks for venting shall be provided at high point or loop crest.

Figure 2.72 shows a typical gauge/valve arrangement. A suitable damping device or a finely adjusted needle valve will reduce gauge fluctuations. A vent valve is required to bleed off air in the lines. A gauge shut-off valve is required to protect the gauge against pressure surges during startup and shut-down of the pump.

2.6.8.1 Complete measurement of head by pressure gauges

The definitions in Paragraph 2.6.3 apply to nomenclature in this section.

2.6.8.1.1 Bowl assembly total head measurement (see Figure 2.73)

When the connecting tube to the pressure gauge is filled with water, then:

$$H_{ba} = \frac{2.31}{s} p_{gba} + Z_d - Z_w + \frac{v_d^{\,2}}{2g}$$

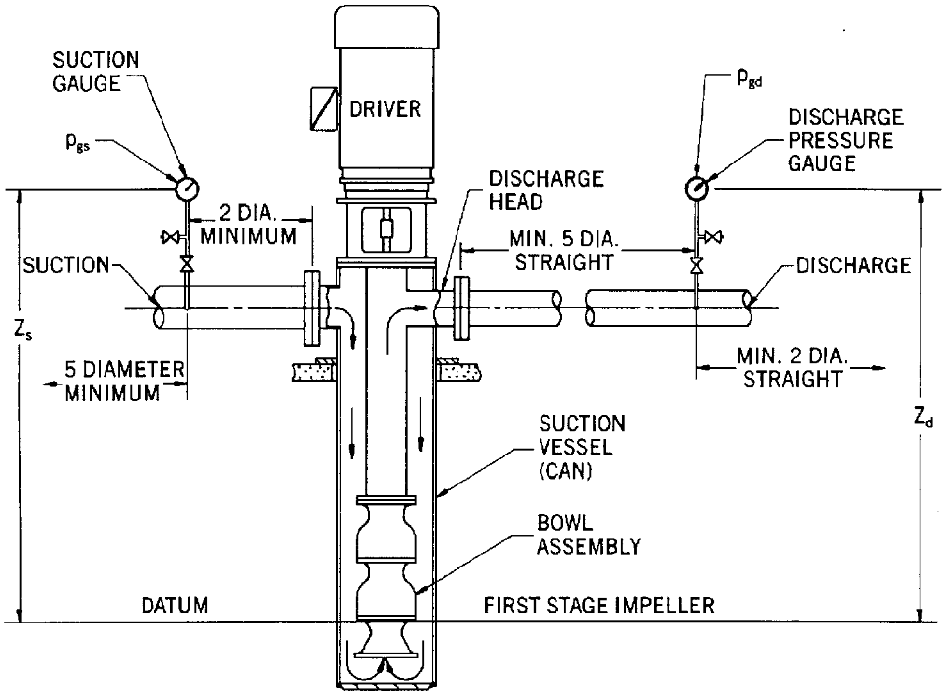


Figure 2.74 — Total head measurement – can pump

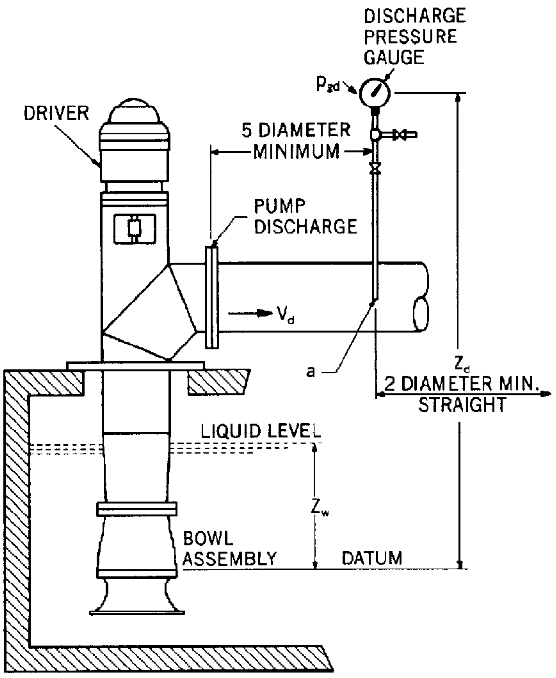


Figure 2.75 — Total head measurement – wet pit

2.6.8.1.2 Total head measurement, closed suction above atmospheric pressure (see Figure 2.74)

When the gauge pressures are above atmospheric pressure and the connecting tubes are filled with water, then:

$$H = \left(\frac{2.31}{s} p_{gd} + Z_d + \frac{{v_d}^2}{2g}\right) - \left(\frac{2.31}{s} p_{gs} + Z_s + \frac{{v_s}^2}{2g}\right)$$

The elevation (Z) is measured to the centerline of the gauge and is negative if the centerline of the gauge lies below the datum line.

2.6.8.1.3 Total head measurement – open suction above atmospheric pressure (see Figure 2.75)

When the gauge pressure is above atmospheric pressure and the connecting tube is filled with water, then:

$$H = \frac{2.31}{S} p_{gd} + Z_d - Z_w + \frac{{v_d}^2}{2g}$$

2.6.8.1.4 Measurement of head with bourdon gauge below atmospheric pressure (see Figure 2.76)

If the pressure at the suction gauge connection (a) is below atmospheric pressure and the connecting tube is completely air-filled, then:

$$h_s = \frac{2.31}{s} p_{gs} + \frac{V_s^2}{2g} + Z_s$$

NOTE – There is no elevation correction to datum since the gauge line is filled with air.

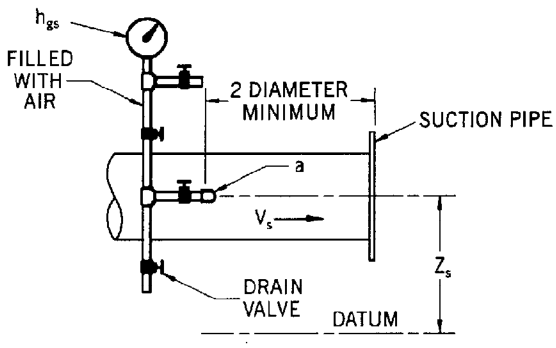


Figure 2.76 — Gauge below atmospheric pressure

2.6.8.2 Measurement of head with fluid gauge below atmospheric pressure (see Figure 2.77)

When the gauge pressure at connection is below atmospheric pressure and the connecting tube has a rising loop and is completely filled with air to prevent water from passing to the fluid measuring column, then (h_s) is calculated as follows:

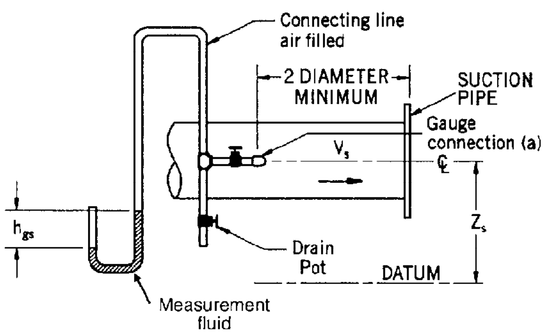


Figure 2.77 — Fluid gauge with air leg below atmospheric pressure

$$h_s = h_{gs} \frac{\gamma \text{ measurement fluid}}{\gamma \text{ water}} + \frac{V_s^2}{2g} + Z_s$$

2.6.9 Power measurement

Pump input power may be determined by transmission dynamometers, torsion dynamometers, strain gauge type torque measuring devices, or other sufficiently accurate measuring devices that result in measurement accuracy of ± 1.5% at the specified condition.

Readings of power shall be taken at the same time that capacity is measured.

When pump input power is determined by transmission dynamometers, the unload dynamometer shall be statically checked prior to the test by measuring the load reading deflection for a given torque and by taking the tare reading on the dynamometer scale at rated speed with the pump disconnected. After the test, the dynamometers shall be rechecked to assure that no change has taken place. In the event of a change of 0.5% of the power at BEP, the test shall be rerun. An accurate measurement of speed within ± 0.3% is essential.

The use of calibrated dynamometers or motors is an acceptable method for measurement of pump input power.

Calibration of the dynamometer shall be conducted with the torsion-indicating means in place. The indicator shall be observed with a series of increasing loadings and then with a series of decreasing loadings. During the taking of readings with increasing loadings, the loading is at no time to be decreased; similarly, during the decreasing loadings, the loading shall at no time be increased. The calculation of output shall be based on the average of the increasing and decreasing loadings as determined by the calibration. If the difference in readings between increasing and decreasing loadings exceeds 1%, the torsion dynamometer shall be deemed unsatisfactory.

Dynamometers shall not be employed for testing pumps with a maximum torque below one-quarter of the rated dynamometer torque.

When strain gauge type torque measuring devices are used to measure pump input power, they shall be calibrated, with their accompanying instrumentation, at regular intervals. After the test, the readout instrumentation balance shall be rechecked to assure that no appreciable change has taken place. In the event of a change of 0.5% of the power at BEP, the test shall be rerun.

Calibrated electric motors, along with calibrated transformers and laboratory type electric motors, are commonly used to measure power input to pumps. The electric input to the motor shall be measured at the motor terminals by acceptable methods such as single and polyphase wattmeters or voltmeter-ammeter, with the proper power factor. Electric power readings at the motor terminals are required to exclude line losses between the controls and the terminals.

For the proper application of the electrical power measuring equipment, refer to *IEEE Standard 552.*

Calibrated electric motors shall have efficiency determined by the methods outlined in the latest revision of the following publications:

- *Standard Test Procedures for Polyphase Induction Motors and Generators, ANSI/IEEE 112 — 1992;*
- *Standard Test Procedures for Direct Current Machines, ANSI/IEEE 113 — 1985;*
- *Standard Test Procedures for Synchronous Machines, ANSI/IEEE 115 — 1983 (R-1991).*

Certified calibration shall be conducted on the specific motor in question and not on a similar machine. Motor thrust bearing losses under actual load are typically not included in the calibration. These losses must, therefore, be deducted from the motor power input to arrive at the true pump's efficiency when performance testing.

For non-calibrated motors, the efficiency may be determined by the segregated losses methods (see *Standards ANSI/IEEE 112, 113, and 115*), if mutually agreed upon prior to the test.

When submersible pump power measurements are taken, the power loss through the electrical cable must be deducted to obtain the actual pump power consumption.

When laboratory testing pumps designed to pump liquids lighter than the laboratory liquid, typically water, with 1.0 specific gravity, consideration must be given to the additional power requirement as well as the increased pressure.

The use of transmission dynamometers and motors that have been calibrated by acceptable methods previously covered shall be taken as giving the actual input power to the pump.

2.6.10 Methods of rotary speed measurement

Test speeds for centrifugal pumps may be in the range of a few hundred to thousands of revolutions per minute. Since the pump test data will be taken under steady state conditions, the maximum permissible short-term speed fluctuation shall be no more than 0.3%. The instruments shall also be capable of measuring speed with an accuracy of ±.3%. The speed measuring methods described, therefore, are those which, at moderate speeds, will give a measure of the average speed over an interval of from less than one second up to two minutes, depending on the type of instrumentation.

NOTE – The various methods and instrumentation are discussed in detail in *Instruments and Apparatus Part 13, Measurement of Rotary Speed 1961, PTC 19.13.*

The revolution counter and timer method, as its name implies, involves the counting of the number of revolutions over an interval of time. A major source of error is inexact synchronization of counter and timer. In cases where this is automatic (e.g., digital tachometers), accuracy is achieved over a time interval of a few seconds. In the case where a hand held counter and stopwatch are used, the timing interval should be about two minutes. During this time, the speed must be constant, and slippage of the counter on the shaft must be avoided. The stopwatch shall be periodically checked against a standard timer.

Tachometers provide a direct reading of speed averaged over a fixed time interval. Some types automatically repeat the reading process; hand-held units must be reset manually. The above comments regarding uniform speed and slippage pertain here also. A tachometer shall be checked periodically against a counter and stopwatch.

Frequency-responsive devices have the advantage of not requiring direct contact with the motor or pump shaft and hence impose no additional load on the motor. The vibrating reed type is of use only when the shaft is completely inaccessible. Electronic units may be converted to read rpm directly using a shaft mounted gear and a non-contacting magnetic pickup. Since normally the line frequency (which determines the timing interval) is 60 Hz ± 0.1 percent, the method is accurate to the nearest rpm, as read on a digital readout. The timing interval may be set as short as 0.1 second, thus making any speed fluctuations readily discernible.

Most stroboscopes are limited in accuracy due to uncertainty in the precision of the strobe frequency. The only approach suitable for pump test purposes is to use the strobe to determine motor slip under load relative to synchronous speed, using a stopwatch to time the slippage while driving the strobe at line frequency (which is known to the accuracy given above and can be determined with even greater precision for the time and location of the test).

2.6.11 Temperature measurement and instruments

Temperature shall be measured as close to the pump inlet as possible. The temperature measuring device shall have no effect on the measurements of pressure and flow rate.

All temperature-sensing instruments shall be properly supported and installed directly into the liquid stream. When this is not feasible, wells filled with suitable intermediate conducting materials may be used.

Temperature may be measured by etched stem, liquid-in-glass thermometers, thermocouples or resistance thermometers. Thermocouples and resistance thermometers, when employed, require potentiometric instruments.

2.6.12 Model tests

2.6.12.1 Procedure

In many installations involving units of large size, model tests are of great utility. Even when it might be feasible to test the large unit in the factory, a model may often be tested with greater accuracy and thoroughness. By adopting a standard size of model for various pumps, comparable performances can be obtained. The model impeller should be not less than 12 inches (300 mm) outside diameter. The exact model-to-prototype ratio shall be selected by the builder. Comparisons between model tests are valid only when all dimensions of the model hydraulic passages to prototype are in accordance with the model-to-prototype ratio.

Testing models in advance of final design and installation of a large unit not only provides advance assurance of performance but makes alterations possible in time for incorporation into the prototype unit.

Not all installations lend themselves to a practical model investigation. The pumping of water carrying considerable quantities of sand or other foreign material is not readily reproduced in model operation. This standard, therefore, is limited to the pumping of clear water, free from abnormal quantities of air or solids, both in field installation and factory tests. The effects of wear and deterioration, the effects of free surface disturbances in open channel sumps, interference between neighboring units, and peculiar problems caused by abnormal settings are covered by model sump tests.

The model hydraulic passages should have complete geometric similarity with the prototype, not only in the pump proper, but also in the intake and discharge conduits, as specified above for tests on full-sized pumps. If cavitation tests are not available, the NPSHA should be such as to give the same suction specific speed as the prototype. As previously explained, if the prototype NPSHR is known to be safely below the NPSHA, then a higher NPSHA can be used for the model tests, although it is preferable to maintain the same value.

There is danger of air separation destroying similarity relationships if the absolute pressure is reduced too low. Consequently, condensate pumps should not be modeled.

If corresponding diameters of model and prototype are D_1 and D_2 respectively, then the model speed n_1 and model capacity Q_1, under the test head H_1, must agree with the relationships:

$$\frac{n_1}{n_2} = \left[\frac{D_2}{D_1}\right]\left[\frac{H_1}{H_2}\right]^{0.5}$$

and

$$\frac{Q_1}{Q_2} = \left[\frac{D_1}{D_2}\right]^2\left[\frac{H_1}{H_2}\right]^{0.5}$$

If a model wet pit pump is tested in its corresponding model intake structure, it should be noted that the conditions to satisfy the pump model relationship and the Froude sump model relationship cannot be obtained simultaneously. Combining these tests is therefore not recommended.

The efficiency of the model will not, in general, be exactly equal to that of the prototype. In testing a model of reduced size, the above conditions being observed, complete hydraulic similarity may not be attained because of certain influences. For example, complete geometric similarity will not be obtained unless the relative roughness of the impeller and pump casing surfaces are the same. With the same surface texture in both model and prototype, the model efficiency will be lower than that of the larger unit. Further, it is generally not practical to model running clearances or bearing sizes. When such is the case, the model efficiency will be reduced.

When a high degree of understanding exists between manufacturer and user relative to the comparison limitations encountered going from model to prototype, thought may be given to the practicability of increasing the prototype efficiency on the basis of model test results. However, this should be done only by mutual agreement before the job is let, on the basis of all the available test data of a similar nature.

Numerous comparisons of prototype and model efficiencies, with consistent surface finish of models and prototypes, are necessary for a given factory to establish a basis for calculating model performance to field performance. This calculation can be applied conveniently according to the formula in use for turbines; namely

$$\frac{1-\eta_1}{1-\eta_2} = \left[\frac{D_2}{D_1}\right]^x$$

The exponent (x) is to be determined from actual data as described above.

The values for the exponent (x) have been found to vary between zero and 0.26, depending on relative surface roughness of model and prototype and other factors.

Example: A single stage pump designed to deliver 90,000 gpm against a head of 400 feet at 450 rpm and have an impeller diameter of 6.8 feet. This pump is too large for a factory test and, in place of such test on the actual pump, a model is to be tested at a reduced head of 320 feet. The model impeller is to be 18 inches (0.457 m) in diameter. Determine speed and capacity for the above model test.

Apply the above relationships:

$$\frac{n_1}{n_2} = \left[\frac{D_2}{D_1}\right]\left[\frac{H_1}{H_2}\right]^{0.5}$$

or

$$n_1 = n_2\left[\frac{D_2}{D_1}\right]\left[\frac{H_1}{H_2}\right]^{0.5}$$

$$= 450\left[\frac{6.8}{1.5}\right]\left[\frac{320}{400}\right]^{0.5} = 1825 \text{ rpm}$$

$$\frac{Q_1}{Q_2} = \left[\frac{D_1}{D_2}\right]^2\left[\frac{H_1}{H_2}\right]^{0.5}$$

or

$$Q_1 = Q_2\left[\frac{D_1}{D_2}\right]^2\left[\frac{H_1}{H_2}\right]^{0.5}$$

$$= 90{,}000\left[\frac{1.5}{6.8}\right]^2\left[\frac{320}{400}\right]^{0.5} = 3920 \text{ gpm}$$

The model pump should therefore be run at a speed of 1825 rpm delivering 3920 gpm against a head of 320 feet.

To check these results, it will be noted that the specific speed of the prototype is:

$$NS_d = \frac{n\,(Q)^{0.5}}{H^{0.75}} = \frac{450(90{,}000)^{0.5}}{400^{0.75}} = 1510$$

and the specific speed of the model will be

$$NS_d = \frac{1825\,(3920)^{0.5}}{320^{0.75}} = 1510$$

Therefore, the specific speeds are the same as required.

2.6.12.2 Test of models at increased head

Under special and unusual circumstances, it may be desirable to carry out factory tests at higher heads than the prototype head. This, for example, may be due to the limitations of available test motors or electrical frequency. In this case, all of the above considerations continue to apply.

The choice of using a model is based on balancing the cost benefits of a smaller model versus the manufacturing and test accuracies.

It should be pointed out, however, that with a reduced-size model, and an increase in head, the increase in speed corresponding to the head increase tends to minimize the change in Reynolds number; that is, the product of flow velocity and linear dimensions of the model tends to approach equality with the same product in the prototype. This effect tends to restore dynamic similarity in model and prototype and to approach equality of efficiencies and other performance factors. With increased head, however, the preservation of the same suction specific speed value in the model as in the prototype must still be observed, and this value will assume increased importance, requiring an increase in submergence or reduction in suction lift in the factory test.

The last mentioned requirement may result in another reason for the use of an increased head in the factory test. Cases may arise in which the limitations of the factory test setup may preclude obtaining sufficient suction lift to reproduce the prototype suction specific speed. In such cases, the required value can be obtained by an increase in the pumping head instead of by a reduction in suction head or an increase in suction lift.