ANSI/HI 2.6-2000

Approved American National Standard

ANSI

American National Standard *for*

Vertical Pump Tests

ANSI/HI 2.6-2000

Hydraulic Institute

9 Sylvan Way
Parsippany, New Jersey
07054-3802
www.pumps.org

This page intentionally blank.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

ANSI/HI 2.6-2000

American National Standard for

Vertical Pump Tests

Secretariat

Hydraulic Institute

www.pumps.org

Approved March 15, 2000

American National Standards Institute, Inc.

Recycled paper

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

American National Standard

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

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

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

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

CAUTION NOTICE: This American National Standard may be revised or withdrawn at any time.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

Published By

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

www.pumps.org

Copyright © 2000 Hydraulic Institute
All rights reserved.

No part of this publication may be reproduced in any form, in an electronic retrieval system or otherwise, without prior written permission of the publisher.

Printed in the United States of America

ISBN 1-880952-34-3

Contents

Tables

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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

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

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

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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.

Black & Veatch LLP
Brown & Caldwell
Camp Dresser & McKee, Inc.
Cascade Pump Company
Chas. S. Lewis & Company, Inc.
Cheng Fluid Systems, Inc.
EnviroTech Pumpsystems
Exeter Energy Limited Partnership
Fairbanks Morse Pump Corp.
Ferris State Univ. Const. and Facilities Dept.
Floway Pumps
Flowserve Corporation
Fluid Sealing Association
Illinois Department of Transportation
Ingersoll-Dresser Pump Company
ITT Industrial Pump Group
J.P. Messina Pump and Hydr. Cons.
John Crane, Inc.
Krebs Consulting Service
Lawrence Pumps, Inc.
Malcolm Pirnie, Inc.
Marine Machinery Association
Moving Water Industries (MWI)
Pacer Pumps
Patterson Pump Company
Pinellas County, Gen. Serv. Dept.
The Process Group, LLC
Raytheon Engineers & Constructors
Reddy-Buffaloes Pump, Inc.
Settler Supply Company
South Florida Water Mgmt. Dist.
Sta-Rite Industries, Inc.
Stone & Webster Eng. Corp.
Sulzer Pumps (USA) Inc.
Summers Engineering, Inc.
Systecon, Inc.
Val-Matic Valve & Manufacturing Corp.
Zoeller Engineered Products

This page intentionally blank.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

2.6 Test

2.6.1 Scope

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.

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.*

2.6.1.1 Objective

This standard is intended 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;

Optional tests as follows when specified:

b) Hydrostatic test of pressure-containing components;

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

d) Mechanical test.

For airborne sound testing see HI 9.1–9.6-2000, *Pumps – General Guidelines.*

2.6.2.1 Test conditions

Unless otherwise specified, the rate of flow, head, efficiency, NPSHR and priming time are based on shop tests using water corrected to 20°C (68°F). 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% and 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 rate of flow, 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 rate of flow, head, speed, NPSH and power at which the pump will normally operate. It may be the same as the rated condition point.

2.6.3.6 Best efficiency point (BEP)

The rate of flow 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:

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

Table 2.11 — Symbols

Symbol	Term	Metric unit	Abbreviation	US Customary Unit	Abbreviation	Conversion factor[a]
A	Area	square millimeter	mm^2	square inches	in^2	645.2
β (beta)	Meter or orifice ratio	dimensionless	—	dimensionless	—	1
D	Diameter	millimeter	mm	inches	in	25.4
Δ (delta)	Difference	dimensionless	—	dimensionless	—	1
η (eta)	Efficiency	percent	%	percent	%	1
g	Gravitational acceleration	meter/second squared	m/s^2	feet/second squared	ft/sec^2	0.3048
γ (gamma)	Specific weight			pounds/cubic foot	lb/ft^3	
h	Head	meter	m	feet	ft	0.3048
H	Total head	meter	m	feet	ft	0.3048
n	Speed	revolutions/minute	rpm	revolutions/minute	rpm	1
NPSHA	Net positive suction head available	meter	m	feet	ft	0.3048
NPSHR	Net positive suction head required	meter	m	feet	ft	0.3048
Ns	Specific speed $N_s = nQ^{1/2}/H^{3/4}$	dimensionless	—	dimensionless	—	1.162
ν (nu)	Kinematic viscosity	millimeter squared/sec	mm^2/s	seconds Saybolt Universal	SSU	0.22
π	pi = 3.1416	dimensionless	—	dimensionless	—	1
p	Pressure	kilopascal	kPa	pounds/square inch	psi	6.895
P	Power	kilowatt	kW	horsepower	hp	0.7457
q	Rate of flow	cubic meter/hour	m^3/h	cubic feet/second	ft^3/sec	101.94
Q	Rate of flow	cubic meter/hour	m^3/h	US gallons/minute	gpm	0.2271
ρ (rho)	Density	kilogram/cubic meter	kg/m^3	pound mass/cubic foot	lbm/ft^3	16.02
s	Specific gravity	dimensionless	—	dimensionless	—	1
t	Temperature	degrees Celsius	°C	degrees Fahrenheit	°F	$(°F-32) \times {}^5/_9$
τ (tau)	Torque	Newton – meter	N·m	pound-feet	lb-ft	1.356
v	Velocity	meter/second	m/s	feet/second	ft/sec	0.3048
x	Exponent	none	none	none	none	1
Z	Elevation gauge distance above or below datum	meter	m	feet	ft	0.3048

[a] Conversion factor × US units = metric units.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

— Metric: cubic meter;

— US units: US gallon;

— US units: cubic foot.

The specific weight of water at a temperature of 20°C (68°F) shall be taken as 9.79 kN/m^3 (62.3 lb. per cu. ft.). For other temperatures, proper specific weight corrections shall be made using values from the ASME steam tables.

2.6.3.9 Rate of flow (capacity) (Q)

The rate of flow of a pump is the total volume throughput per unit of time at suction conditions. It assumes no entrained gases at the stated operating conditions.

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 2.63).

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 2.64).

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 meter (feet) 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:

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

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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

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

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 or liquid surface.

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 2.65).

The total suction head (h_s) at datum (see Figure 2.63) is the submergence in meters (feet) of water (Z_w). The average velocity head of the flow in the pit is small enough to be neglected and gauge head is zero.

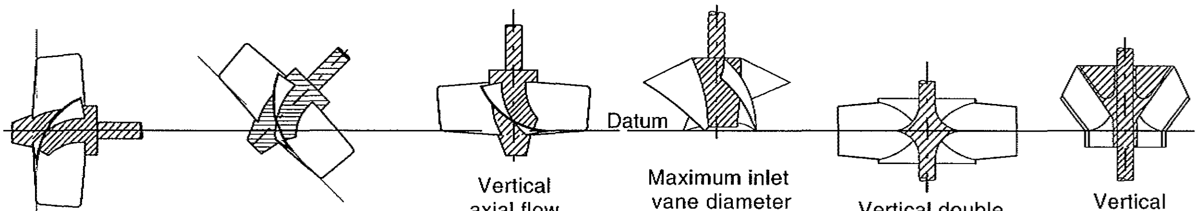


Figure 2.63 — Datum elevations for various pump designs

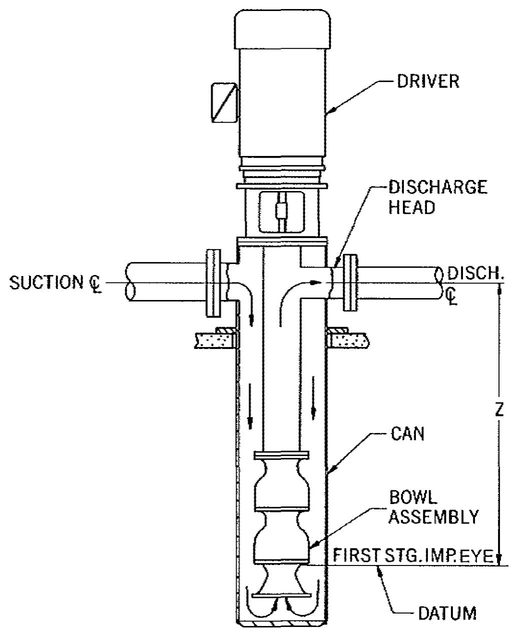


Figure 2.64 — First stage impeller datum closed suction – can pump

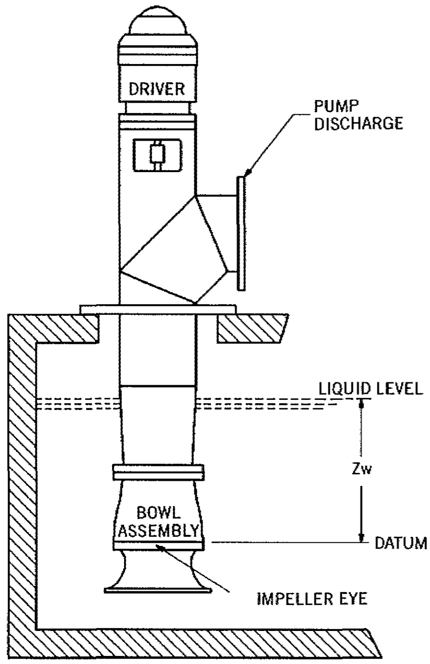


Figure 2.65 — Total suction head – open suction

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

$h_s = Z_w$

Where:

Z_w = vertical distance in meters (feet) from free water surface to datum.

NOTE: When absolute suction head is required for NPSH considerations, refer to Section 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 2.66).

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}) in meters (feet) 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 Section 2.6.6.4 for definition.

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 discharge elevation head (Z_d) from the discharge gauge centerline to the pump datum (see Figure 2.67).

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

For location of instruments for head measurements, see Section 2.6.9.

2.6.3.12.7 Total head (H)

This is the measure of work increase per unit weight 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.

$H = h_d - h_s$

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.

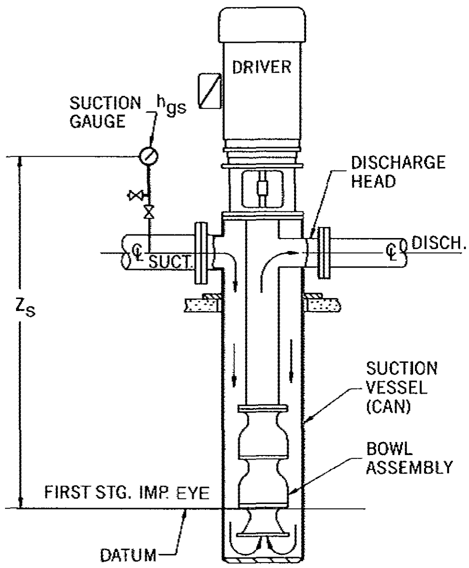


Figure 2.66 — Total suction head – closed suction

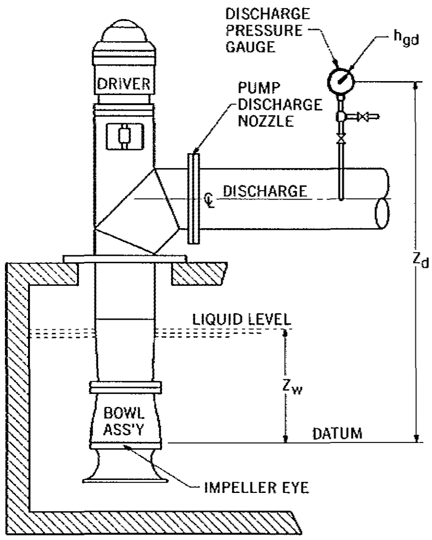


Figure 2.67 — Total head – open suction

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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_{vd}) 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 2.67).

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

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_{vd}) 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 2.68), 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 meters (feet).

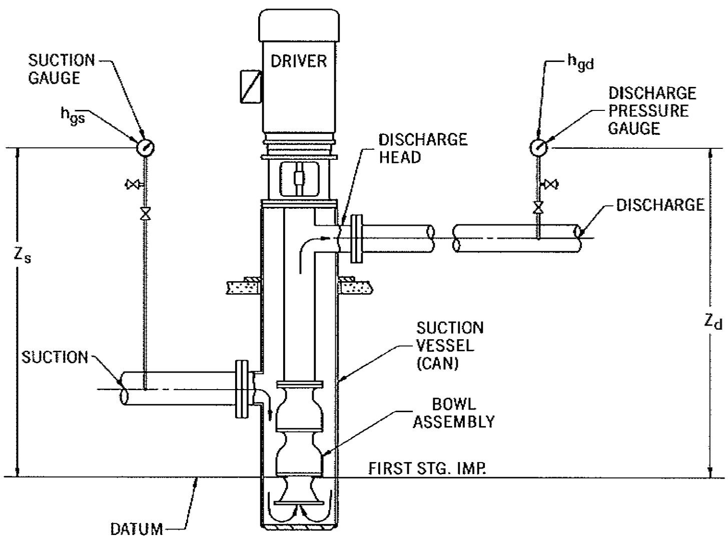


Figure 2.68 — Total head – closed suction

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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

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

or

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

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

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 rate of flow.

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 sometimes called *brake horsepower.*

2.6.3.13.2 Electric driver input power (P_{mot})

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

2.6.3.13.3 Bowl assembly input power (P_{ba})

The power 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.*

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

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

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 (ηp)

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 Section 2.6.5.8.3. 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

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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 pump output power (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 (optional)

2.6.4.1 Hydrostatic test 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 liquid means only prevention of its escape through the external surfaces of the pump, normally to the atmosphere.

2.6.4.2 Hydrostatic 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 32 mm^2/sec (150 SSU) 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 Hydrostatic 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.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

2.6.4.4 Hydrostatic test 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 Performance test 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 Section 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 Performance test witnessing

The purchaser or purchaser's designated representative may witness the test when requested by the purchaser in the purchase order.

2.6.5.3 Performance test acceptance tolerances

In making tests under this standard no minus tolerance or margin shall be allowed with respect to rate of flow, total head, or efficiency at rated or specified conditions.

Acceptance of the pump test results will be judged at rated rate of flow and rpm with applicable total head and efficiency as follows:

Total head	Tolerance
Under 60 m (200 ft) and 680 m^3/h (2999 gpm)	+ 8%, – 0
Under 60 m (200 ft) and 681 m^3/h (3000) gpm and over	+ 5%, – 0
From 61 m (201 ft) to 150 m (500 ft), any rate of flow	+ 5%, – 0
151 m (501 ft) and over, any rate of flow	+ 3%, – 0

NOTE: Minimum efficiency at rated rpm and rate of flow shall be contract efficiency.

Alternately, the pump test results may be judged at rated total head and rpm versus rate of flow as follows:

— Rate of flow 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 rate of flow 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 rate of flow 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 the test and calculated results for reduced diameter impeller may be submitted as a final acceptance test without further testing.

2.6.5.4 Performance test instrumentation

Test instrumentation shall be selected so that it can provide measurements with accuracy shown in Section 2.6.5.4.1 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.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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.

Table 2.13 — Recommended instrument calibration interval[a]

Rate of flow		Power (continued)	
Quantity meter		Torque bar	1 yr
Weighing tanks	1 yr	Calibrated motor	Not req'd[b]
Volumetric tank	10 yr	KW transducer	3 yr
Rate meters		Watt-amp-volt, portable	1 yr
Venturi	c	Watt-amp-volt, permanent	1 yr
Nozzle	c	Strain gauges	6 mo
Orifice plate	c	Transmission gears to 500 HP	10 yr
Weir	c	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[b]
Ultrasonic	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		

[a] Use instrument manufacturer's recommendation if shorter than listed above.

[b] Unless electrical or mechanical failure.

[c] Calibration is not required unless it is suspected there are critical dimensional changes.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

	Acceptable fluctuation of test reading in ± %	Required accuracy of the instrument in ± % of the specified values being observed
Rate of flow	2	1.0
Differential pressure or head	2	1.0
Discharge head	2	0.5
Suction head	2	0.5
Input power	2	0.75
Speed	0.3	0.3

2.6.5.5 Performance test setup

This section contains general guidelines for performance testing to ensure accurate and repeatable test results.

The performance 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:

   i) Open suction bowl assembly test, (see Figure 2.69). 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 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;

   ii) Closed suction and closed loop, pump or bowl assembly performance test (see Figures 2.70 and 2.71). These types of pump test setups are used when both NPSH and performance testing are 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;

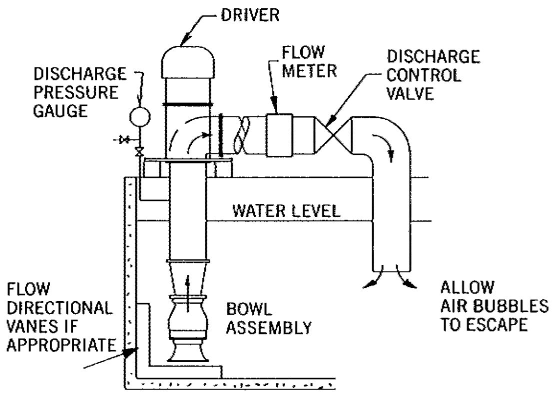


Figure 2.69 — Bowl assembly performance test – open sump

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

iii) 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 liquid 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 rate of flow, such as by weight, by volume or by rate meters;

13) Test setups for NPSHR 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

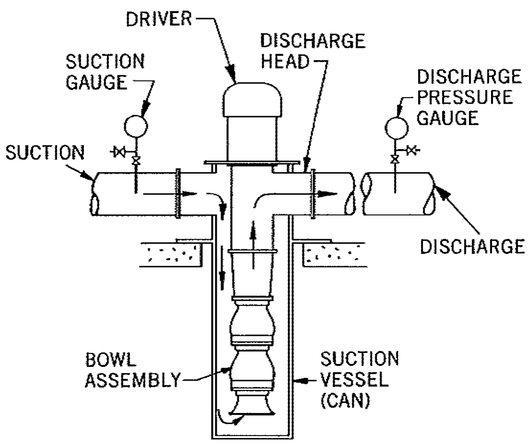


Figure 2.70 — Pump performance test – closed suction

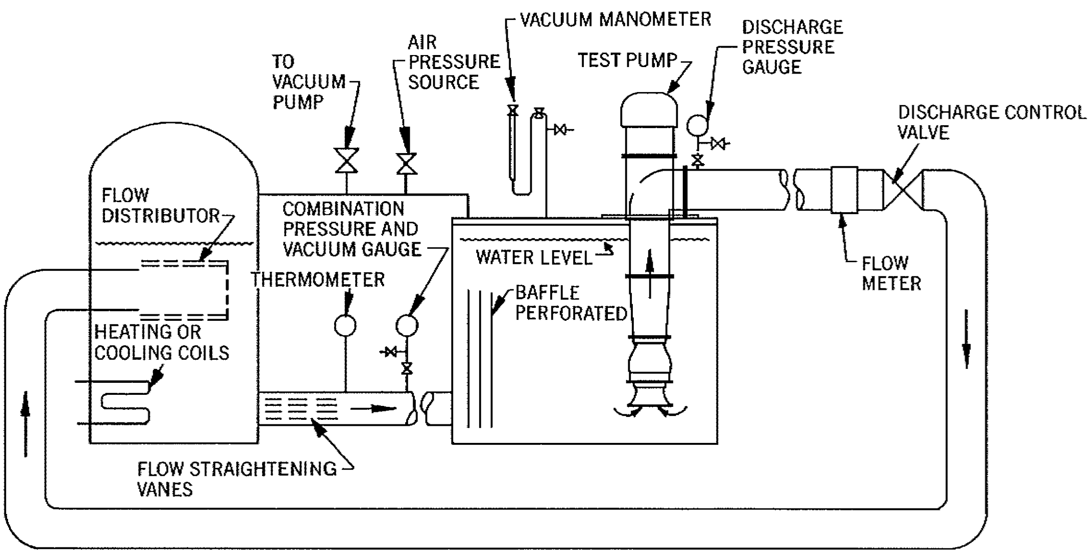


Figure 2.71 — Pump performance test – closed loop

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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 Performance test 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 14):

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 NPSHR or for high power 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;

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 Performance test 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.

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 Performance test calculations

2.6.5.8.1 Calculations of total suction head (h_s)

For a closed system (can pump):

$$h_s = h_{gs} + Z_s + \frac{V_s^2}{2g} \text{ (see Figure 2.66)}$$

For an open system (pump in open pit):

$$h_s = Z_w \text{ (see Figure 2.65)}$$

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 and the gauge head is zero.

2.6.5.8.2 Calculation of total discharge head (h_d)

For closed suction (can pump) (see Figure 2.68) and open suction, wet pit (see Figure 2.67):

$$h_d = h_{gd} + Z_d + \frac{V_d^2}{2g}$$

The discharge pressure gauge is located downstream of the pump's discharge head, and all internal pump hydraulic losses are therefore included.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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: (can pump) nominal ____________ actual ____________
8. Size discharge: nominal ____________ actual ____________

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 ____________
8. Driver efficiency ____________

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. Rate of flow ____________
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 ____________

Rate of flow:

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 ____________

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

2.6.5.8.3 Calculation of bowl assembly total head (H_{ba})

$$H_{ba} = h_{gd} + (Z_d - Z_w) + \frac{v_d^2}{2g} \text{ (see Figure 2.85)}$$

The discharge gauge pressure tap is located a minimum of two 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:

(Metric units) $P_p = \frac{n\,\tau}{60{,}000}$

(US units) $P_p = \frac{2\pi LWn}{33{,}000} = \frac{n\,\tau}{5250}$

Where:

L = Length of lever arm in m (ft.);

W = Net force in N (lbs.);

n = Speed in rpm;

τ = Torque in N•m (pound feet)

The input power to an electric motor in horsepower is given by:

(Metric) $P_{mot} = kW$

(US Units) $P_{mot} = \frac{kW}{0.746}$

Where:

kW = Electrical input power in kilowatts

The input power to a pump driven by an electric motor is:

$$P_p = P_{mot} \times \frac{\eta_{mot}}{100}$$

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) Metric units: $P_w = \frac{Q \times H \times s}{366}$

b) 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:

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

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.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

2.6.5.8.7.1 Calculation of bowl assembly efficiency (η_{ba})

This efficiency value excludes all losses outside the bowl assembly proper:

$$\text{(Metric units)}\quad \eta_{ba} = \frac{Q \times H_{ba} \times s}{366 \times P_{ba}} \times 100$$

$$\text{(US units)}\quad \eta_{ba} = \frac{Q \times H_{ba} \times s}{3960 \times P_{ba}} \times 100$$

NOTE: Refer to the *Hydraulic Institute Engineering Data Book* for column pipe friction losses and driveshaft bearing losses.

2.6.5.8.7.2 Calculation of 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 Calculation of 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 $\times$ 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.

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 performance test results

The head, efficiency and input power are plotted as ordinates on the same sheet with rate of flow as the abscissa (see Figure 2.72). 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 Performance test at non-rated conditions

2.6.5.10.1 Performance 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 rate of flow 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 rate of flow;

Q_2 = Rated rate of flow;

H_1 = Test head;

H_2 = Rated head;

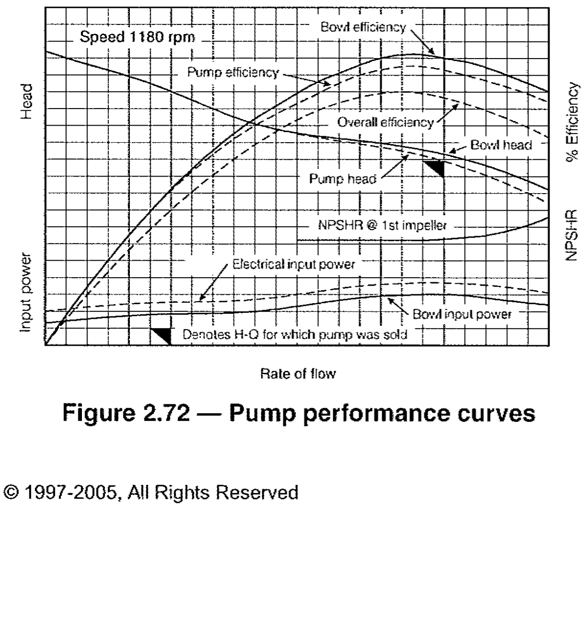


Figure 2.72 — Pump performance curves

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

P_1 = Test power;

P_2 = Rated power;

$NPSHR_1$ = Test NPSHR;

$NPSHR_2$ = Rated NPSHR.

EXAMPLE (Metric): A four-stage bowl assembly is rated at 100 m^3/h against a bowl head of 75 meters, NPSHR of 5 meters 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, rate of flow 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:

$$H_1 = H_2\left(\frac{n_1}{n_1}\right)^{½} = 75\left(\frac{1770}{2950}\right)^2 = 27 \text{ meters}$$

The equivalent rate of flow for the factory test is:

$$Q_1 = Q_2\frac{n_1}{n_1} = 100\frac{1770}{2950} = 60\ m^3/h$$

The NPSH required for the factory test is:

$$NPSHR_1 = NPSHR_2\left(\frac{n_1}{n_1}\right)^2 = 5\left(\frac{1770}{2950}\right)^2 = 1.8 \text{ meters}$$

EXAMPLE (US Units): 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, rate of flow 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:

$$H_1 = H_2\left(\frac{n_1}{n_1}\right)^{½} = 240\left(\frac{1770}{2950}\right)^2 = 86.4 \text{ feet}$$

The equivalent rate of flow for the factory test is:

$$Q_1 = Q_2\frac{n_1}{n_1} = 400\frac{1770}{2950} = 240 \text{ gpm}$$

The NPSH required for the factory test is:

$$NPSHR_1 = NPSHR_2\left(\frac{n_1}{n_1}\right)^2 = 14\left(\frac{1770}{2950}\right)^2 = 5 \text{ feet}$$

2.6.5.10.2 Performance 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 performance test speed variations

The pump test speed will vary with operating conditions.

For purposes of plotting the test results, rate of flow, 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 Section 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 and in input power; therefore, the efficiency is not changed.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

Reduced viscosity of water due to a temperature increase will impact efficiency. For pumps in the lower range of specific speed, typically below 1750 (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:

$$\eta_{ot} = 1-(1-\eta_t)\left(\frac{\upsilon_{ot}}{\upsilon_t}\right)^x$$

Where:

η_{ot} = Efficiency at operating temperature, decimal value;

η_t = Efficiency at test temperature, decimal value;

υ_{ot} = Kinematic viscosity at operating temperature;

υ_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 38°C (100°F) resulted in an efficiency of 80 percent. What will be the projected efficiency at 177°C (350°F)?

$$\eta_{ot} = 1-(1-\eta_t)\left(\frac{\upsilon_{ot}}{\upsilon_t}\right)^x$$

$$\eta_{ot} = 1-(1-.80)\left(\frac{.00000185}{.0000076}\right)^{0.1}$$

$$\eta_{ot} = .826 = 82.6\%$$

2.6.5.10.5 Correcting for 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:

$$(P_p)_2 = (P_p)_1 \times \frac{\Upsilon_2}{\Upsilon_1}$$

There is no change in efficiency.

2.6.5.10.6 Correcting for viscosity variations

Viscosity has a significant effect on pump performance with respect to head, rate of flow, efficiency and input power. 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 ANSI/HI 2.3-2000, *Vertical Pump Operation.*

2.6.5.10.7 Correcting for 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 liquids are not part of this test standard.

2.6.5.11 Report of performance test

Parties to the test shall be furnished a copy of the performance curve at constant speed, as drawn in accordance with Section 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 (optional)

2.6.6.1 NPSHR test objective

To determine the NPSH required (NPSHR) by the pump.

2.6.6.2 NPSHR test arrangement

Four typical test setups are shown for determining the NPSHR characteristics of pumps.

In the first arrangement, shown in Figure 2.73, the pump is supplied from a sump through a throttle valve, which is followed by a section of pipe containing a screen and straightening vanes. This minimizes the

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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 3 meters (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.

In the second arrangement, Figure 2.74, 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 excess of atmospheric pressure. Care must be taken to prevent vortexing when the liquid level is varied.

In the third arrangement, Figure 2.75, 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, by varying the temperature of the liquid, or both.

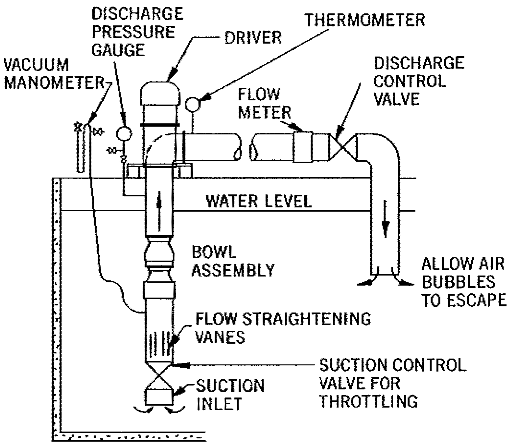


Figure 2.73 — Suction throttling NPSH test constant sump level

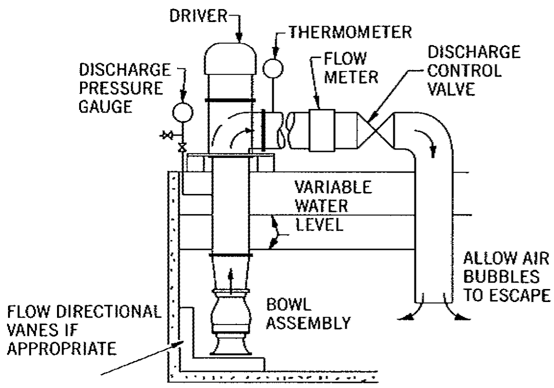


Figure 2.74 — Level control NPSH test

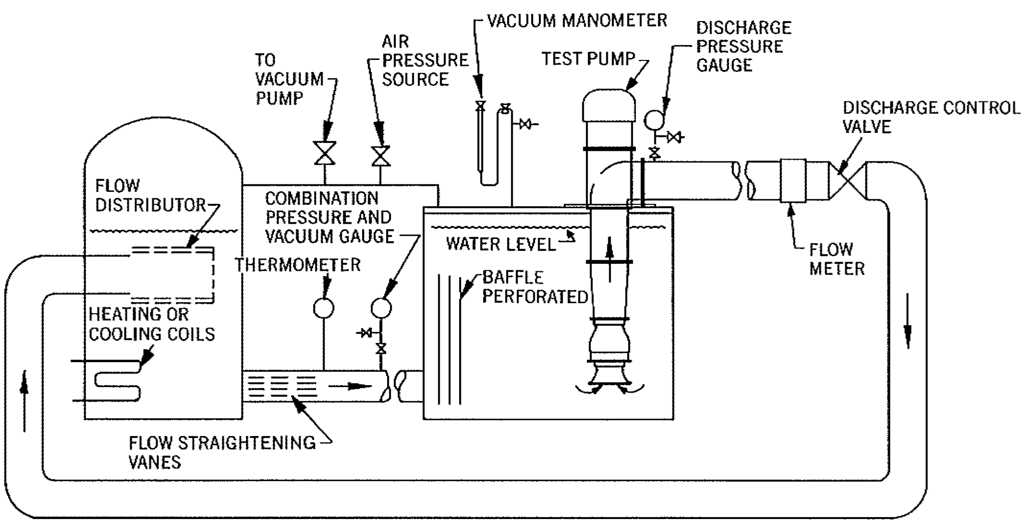


Figure 2.75 — Closed loop NPSH test

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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. This arrangement is more effective for high specific speed mix flow and propeller pumps.

The fourth arrangement, Figure 2.76, shows a typical NPSHR test for a can pump. This arrangement is used when the suction condition approaches zero meters (feet) NPSHA at the pump (can) suction centerline 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 pump (can) 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*: Liquid aeration shall be minimized by taking the following precautions:

  - Intake structure designed to avoid vortexing. (See ANSI/HI 9.8-1998, *Pump Intake Design*);
  - Submerged lines when pressure is below atmospheric, if practical;
  - Reservoir sized for long retention to allow air to escape. Inlet to sump located to prevent vortexing;

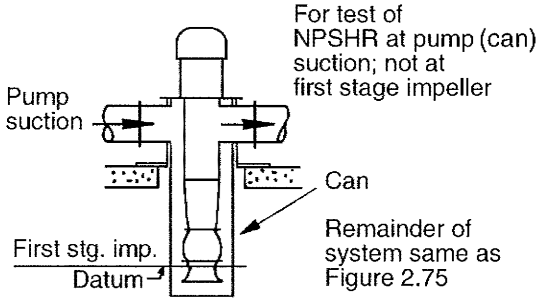


Figure 2.76 — Closed loop NPSH test – alternate arrangement for can pump

  - 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 Section 2.6.13.

2.6.6.3 NPSHR test procedure

Unless otherwise agreed between the purchaser and the manufacturer, the test shall be run for the range of ± 20% of rated flow rate with three test flow rates 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 rate of flow 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 2.77) 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 2.77 shows the results typical of tests at flow rates 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 rate of flow. The test is repeated for various suction head values and the total head plotted against rate of flow. Such tests will result in a family of curves, as shown in Figure 2.79. Where the pump head for any suction head (h_S) breaks away from the normal head curve by 3%, NPSHR (NPSH required) is established.

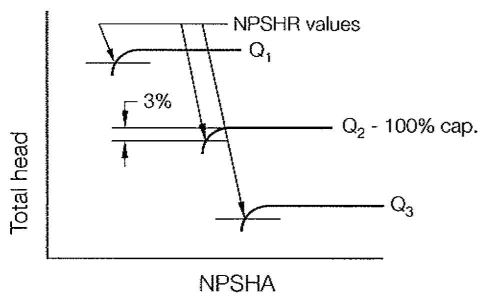


Figure 2.77 — NPSH at constant rate of flow

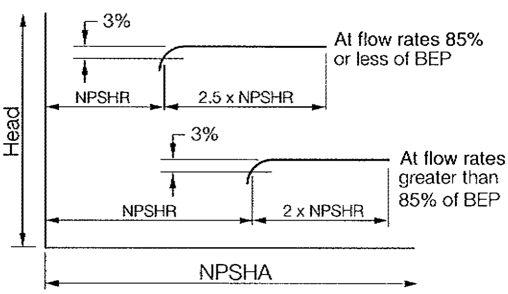
Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

Accurate determination of the 3% head drop 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 rate of flow or a change in sound or vibration—may indicate the presence of excess cavitation. With the difficulty in determining just when the change starts, a drop in head of 3% at a given rate of flow or NPSH is generally accepted as evidence that excess 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 flow rates greater than 85% of BEP, and at least two and one half times the predicted NPSHR for flow rates below 85% of BEP, is recommended for assurance that full performance conditions exist (see Figure 2.78).

Tests performed to establish NPSHR for a specific pump must begin with a full performance 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.



NPSH test at constant rate of flow

Recommended NPSHA range for NPSHR test when no previous data on pumps full performance is available.

Figure 2.78 — NPSH at constant rate of flow

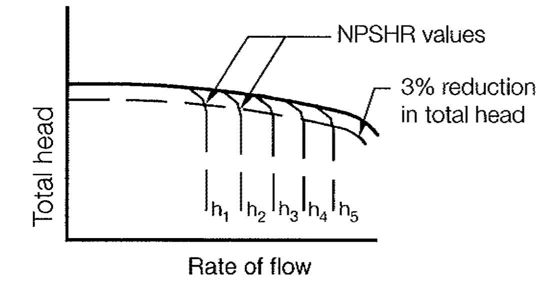


Figure 2.79 — NPSH at varying rate of flow

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.

2.6.6.3.1 Correction to rated speed from test speed for NPSHR

$$NPSHR_2 = \left(\frac{n_2}{n_1}\right)^2 \times NPSHR_1$$

and $Q_2 = \dfrac{n_2}{n_1} \times Q_1$

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 = Rate of flow at test speed;

Q_2 = Rate of flow at rated speed.

2.6.6.3.2 NPSHR experimental deviation from the square law.

The affinity relationships define the manner in which head, rate of flow, 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.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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 NPSHR test suction conditions

The suction lift or suction head is to be measured as specified in Section 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 meters (feet) of liquid absolute, determined at the first stage impeller eye (datum), less the absolute vapor pressure in meters (feet) of the liquid pumped:

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

Where:

h_{sa} = Total suction head in meters (feet) absolute = $h_{atm} + h_s$;

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

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 NPSHR test records

Complete written or computer records shall be kept by the test facility of all data relevant to the NPSHR test for a minimum of two years. (See sample data sheet on page 14.)

These records must include:

1) Specified NPSHR and 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, rate of flow;
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 NPSHR test

All parties to the test shall be furnished a copy of the NPSHR curve or curves, as described in Section 2.6.6.3.

2.6.7 Mechanical test (optional)

2.6.7.1 Mechanical test objective

To demonstrate the satisfactory mechanical operation of a pump, at the rated conditions, including: vibration levels; lack of leakage from shaft seals, gaskets, and lubricated areas; and free running operation of rotating parts. When specified, bearing temperature stabilization will be recorded.

2.6.7.2 Mechanical test setup

The test setup shall conform to the requirements of Section 2.6.5.5 where applicable, and the test liquid shall be clear water. In addition, instrumentation shall be added to measure the following:

a) Vibration at the pump motor base, in two directions perpendicular to the shaft plus the axial direction.

b) Temperature of bearings or bearing housing in discharge head.

c) Leakage from mechanical seals, gaskets, and bearing lubricant. Visual observation is sufficient for all leakage.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

2.6.7.3 Mechanical test operating conditions

The mechanical test shall be conducted under the following operating conditions:

a) Shaft speed - as required to meet rated conditions as specified in the customer's order. Facility 60 or 50 hertz speeds may be used when customers hertz are not available, or as agreed to by customer.

b) Rate of flow - the rated flow rate for which the pump is sold, or as adjusted to a speed other than contract by Section 2.6.5.10.1.

c) Suction pressure - as available from the test facility.

d) Liquid temperature - at ambient condition.

e) Ambient air temperature.

2.6.7.4 Mechanical test instrumentation

2.6.7.4.1 Vibration instruments can be either hand held or rigidly attached to the pump. The sensor(s) shall be velocity type designed to read the nominal RMS velocity without filtering to specific vibration frequencies. Readings can be taken manually or with recording instruments.

2.6.7.4.2 Temperature instruments can be any recognized temperature sensors such as pyrometers, thermometers, thermocouples and the like. They shall be capable of measuring the metal temperature on the outside of the bearing housing, and may be hand held or rigidly attached to the bearing housing. Where temperature sensors are built into the pump, they shall be used instead of sensors on the bearing housing. If built in, they must be at a location where temperature is of interest.

2.6.7.5 Mechanical test procedure

The pump rate of flow and suction pressure shall be set per Section 2.6.7.3. The pump shall be operated for a minimum of 10 minutes, and the following observations made and data recorded:

a) Leakage from shaft seals, gaskets, mechanical seal piping, and bearing housing (s).

b) Vibration level on motor base, in two directions perpendicular to the shaft plus the axial direction. Only the unfiltered RMS velocity values need be recorded.

c) Bearing temperatures shall be recorded. When specified, the pump shall be operated until the bearing temperature stabilizes.

d) Rubbing of rotating parts shall be checked for by listening for unusual or excessive noise, and observing the coast down of the pump when power is cut off. Torque readings or other changes in similar instrument readings can also indicate rubbing.

e) Liquid temperature and ambient air temperature shall be taken manually or with recording instruments.

2.6.7.6 Mechanical test acceptance levels

The mechanical performance is considered acceptable when each of the following is achieved:

a) Vibration levels on the pump driver in any direction do not exceed the allowable limits specified on the order document.

b) Temperature of bearing housing surface does not exceed the pump manufacturer's standard for the product.

c) Mechanical seals may have an initial small leakage, but shall have no visible leakage when running at test operating conditions for a minimum of 10 minutes. When shut down, there shall be no visible leakage from seals for five minutes with the test suction pressure applied. The purpose of this test is to ensure that the entire seal (cartridge) has been properly installed.

Soft packing shall have no more than 12 drops per minute leakage for a 25-mm (1-inch) shaft up to 3500 rpm. For larger shafts or higher test speeds and pressures, allowable leakage shall be increased proportionately with shaft diameter speed and pressure or as agreed to by the purchaser.

There shall be no visible leakage through pressure containment parts, gaskets, seal recirculation piping, bearing housing, etc. Minor leakage at pump discharge flange shall not be cause for rejection because these joints are disconnected and reconnected in the field.

d) Rubbing of rotating parts shall not be apparent from excessive noise during operation nor abrupt stopping of the pump when power is cut off.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

2.6.7.7 Mechanical test records

The following data shall be recorded in either written or computer form and kept on file, available to the purchaser by the test facility, for two years.

a) The manufacturer's serial number, pump type and size, or other means of identification of the pump.

b) Vibration levels on motor base in two directions perpendicular to the shaft plus the axial direction.

c) Temperature at bearing.

d) Ambient air temperature.

e) Leakage from the pump as observed at the following:

  - Pump pressure-containment components
  - Pump gaskets
  - Mechanical seal piping
  - Mechanical seal(s) or packing
  - Bearing housing(s)

f) Free-running rotating parts

g) Date of test

h) Name of test technician

2.6.8 Measurement of rate of flow

2.6.8.1 Introduction

Any flow measuring system may be used for measuring pump rate of flow. 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 rate of flow with an accuracy of ±1.0% at BEP.

Rate of flow instruments are classified into two functional groups. One group primarily measures batch quantity, and the other primarily measures rate of flow.

2.6.8.2 Rate of flow measurement by weight

Measurement of rate of flow 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.

2.6.8.3 Rate of flow measurement by volume

Measurement of rate of flow by volume 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.

2.6.8.4 Rate of flow measurement by head type rate meters

Measurement of rate of flow by head type meters 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 rate of flow. The meters discussed in Sections 2.6.8.4.1, 2.6.8.4.2 and 2.6.8.4.3 use this principle.

Meters falling within this classification, and acceptable for rate of flow 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.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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 ensure 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 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 2.80 and 2.81).

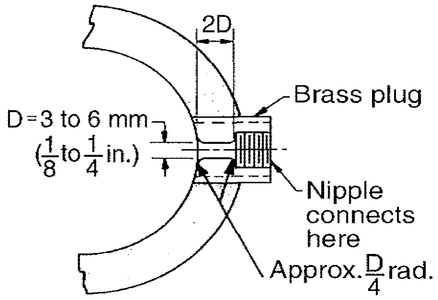


Figure 2.80 — Pressure tap opening

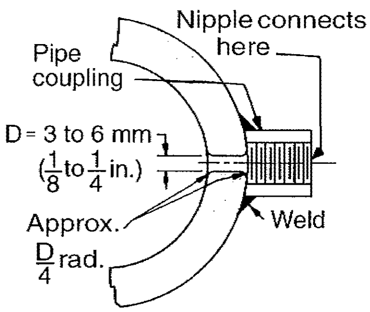


Figure 2.81 — Welded-on pressure tap opening

2.6.8.4.1 Rate of flow measurement by venturi meter

To ensure accurate results in the measurement of flow rates 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.

2.6.8.4.2 Rate of flow measurement by nozzles

To ensure accurate results in the measurement of flow rates 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 discharging directly into a venturi meter should have at least 10 diameters of straight pipe between it and the meter.

2.6.8.4.3 Rate of flow 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 Section 2.6.8.4, and shall, in addition, indicate the exact location and size of pressure taps, which are then to be duplicated in the test installation.

To ensure accurate results in the measurement of flow rates 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.

2.6.8.5 Rate of flow measurement by weirs

Measurement of rate of flow by weirs 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 rate of flow.

The rectangular sharp-crested weir with smooth vertical crest wall, complete crest contraction, free over-fall and with end contraction suppressed is acceptable for rate of flow determination under this standard. It may be used for either factory or field testing.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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

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

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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

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.

2.6.8.6 Rate of flow measurement by pitot tubes

A pitot tube is a double tube, one within the other. Rate of flow is measured by inserting the tube so that it points upstream in 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 velocity head which in turn determines rate of flow.

Where it is impracticle 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.

2.6.8.7 Rate of flow measurement by other methods

When the methods of rate of flow 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 Section 2.6.5.4.1 can be demonstrated.

2.6.9 Head measurement

The units of head and the definition of total head and its component parts are covered in Section 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 2.69).

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 Section 2.6.3.12 for definition of the total discharge head and total suction head.)

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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 300 mm (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 3 to 6 mm (1/8 to 1/4 inch) 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 2.80 and 2.81 show suggested arrangements of taps in conformity with the above.

A single tap connection (Figure 2.82) is used under normal conditions when the test arrangement is in compliance with this standard.

Multiple tap connections (Figure 2.83) are used where abnormal velocity profiles are suspected or conditions preclude compliance with test arrangements in this standard.

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.

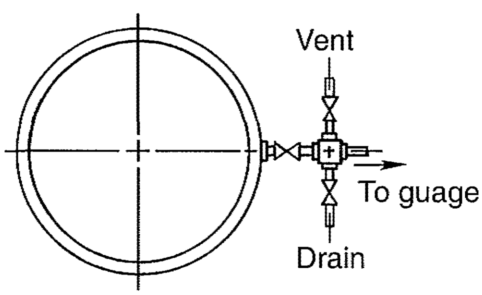


Figure 2.82 — Single tap connection

All connections and lines from the orifice tap shall be free of liquid leakage and as short and direct as possible. For the dry-tube type of lines, suitable 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 2.84 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

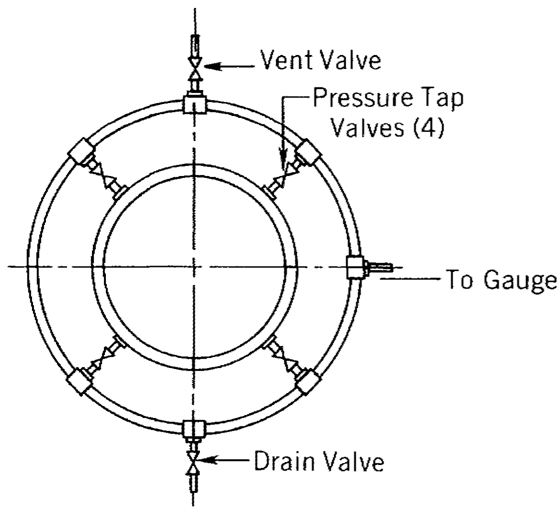


Figure 2.83 — Loop manifold connecting pressure tabs

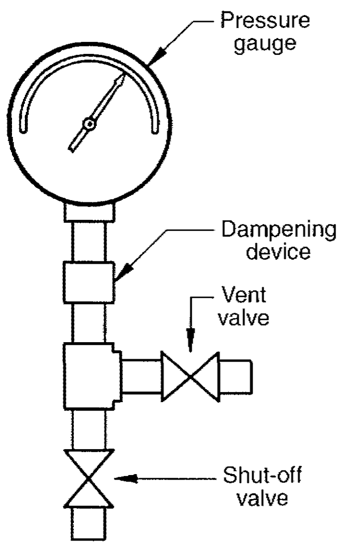


Figure 2.84 — Gauge/valve arrangement

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

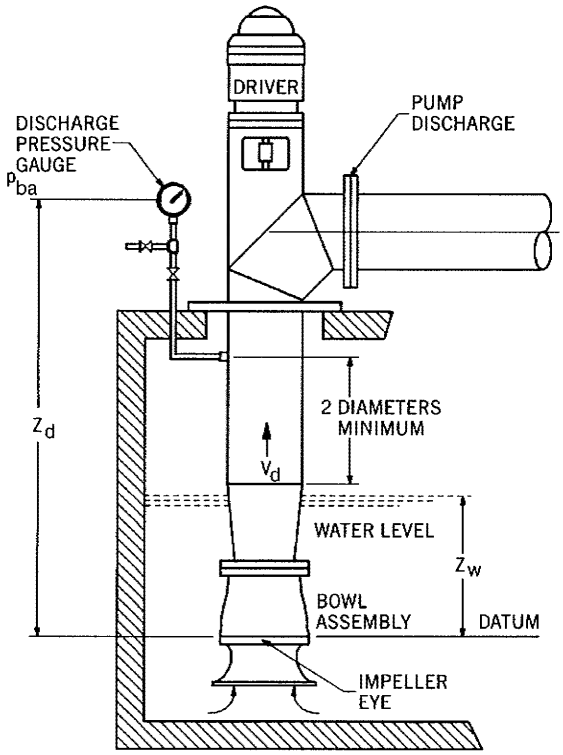


Figure 2.85 — Bowl assembly head measurement

valve is required to protect the gauge against pressure surges during startup and shut-down of the pump.

2.6.9.1 Head measurement by pressure gauges

The definitions in Section 2.6.3 apply to nomenclature in this section.

2.6.9.1.1 Bowl assembly total head measurement (see Figure 2.85)

When the connecting tube to the pressure gauge is filled with water, then:

$$\text{(Metric)}\quad H_{ba} = \frac{p_{gba}}{9.8s} + Z_d - Z_w + \frac{V_d^2}{2g}$$

$$\text{(US Units)}\quad H_{ba} = \frac{2.31}{s}p_{gba} + Z_d - Z_w + \frac{V_d^2}{2g}$$

2.6.9.1.2 Total head measurement, closed suction above atmospheric pressure (see Figure 2.86)

When the gauge pressures are above atmospheric pressure and the connecting tubes are filled with water, then:

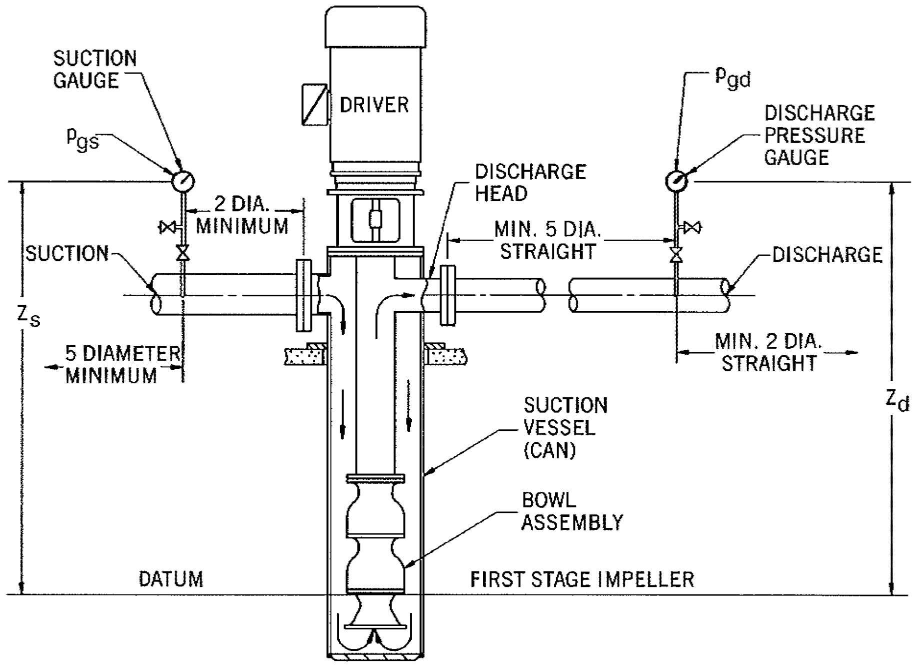


Figure 2.86 — Total head measurement – can pump

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

(Metric): $H = \left(\frac{p_{gd}}{9.8s} + Z_d + \frac{V_d^2}{2g}\right) - \left(\frac{p_{gs}}{9.8s} + Z_s + \frac{V_s^2}{2g}\right)$

(US Units):

$$H = \left(\frac{2.31}{s}p_{gd} + Z_d + \frac{V_d^2}{2g}\right) - \left(\frac{2.31}{s}p_{gs} + Z_s + \frac{V_s^2}{2g}\right)$$

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.

2.6.9.1.3 Total head measurement – open suction above atmospheric pressure (see Figure 2.87)

When the gauge pressure is above atmospheric pressure and the connecting tube is filled with water, then:

(Metric): $H = \frac{p_{gd}}{9.8s} + Z_d - Z_w + \frac{V_d^2}{2g}$

(US Units): $H = \frac{2.31}{s}p_{gd} + Z_d - Z_w + \frac{V_d^2}{2g}$

2.6.9.1.4 Head measurement with bourdon gauge below atmospheric pressure (see Figure 2.88)

If the pressure at the suction gauge connection (a) is below atmospheric pressure and the connecting tube is completely air-filled, then:

(Metric): $h_s = \frac{p_{gs}}{9.8s} + \frac{V_s^2}{2g} + Z_s$

(US Units): $h_s = \frac{2.31}{s}p_{gs} + \frac{V_s^2}{2g} + Z_s$

NOTE: There is no elevation correction to datum since the gauge line is filled with air.

2.6.9.2 Head measurement with fluid gauge below atmospheric pressure (see Figure 2.89)

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

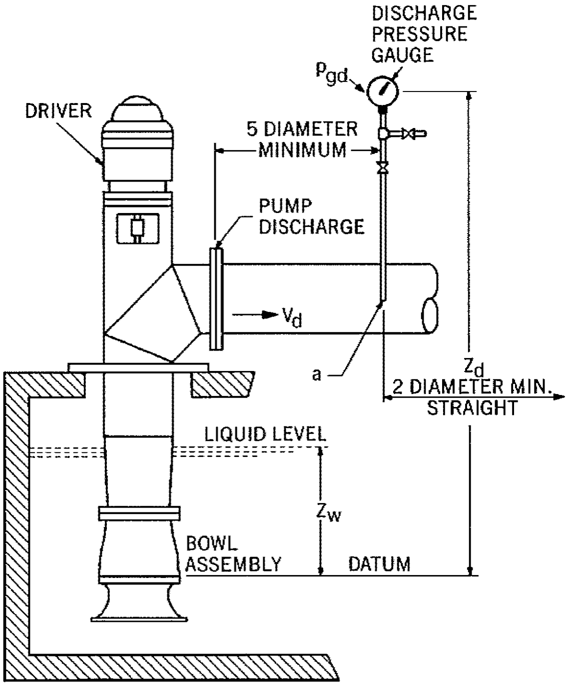


Figure 2.87 — Total head measurement – wet pit

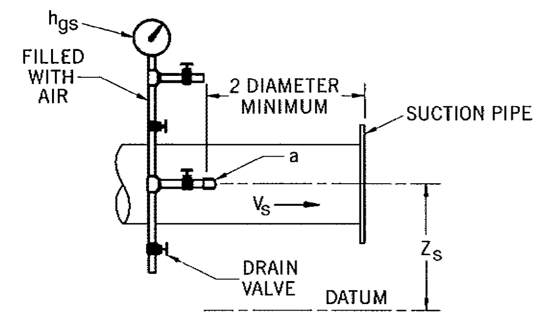


Figure 2.88 — Gauge below atmospheric pressure

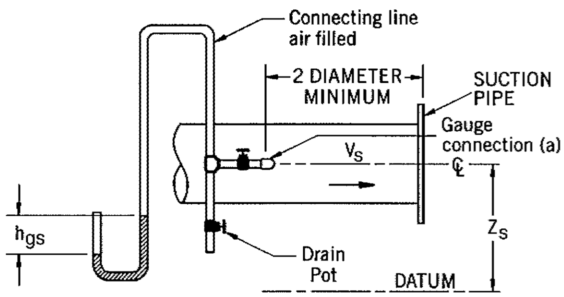


Figure 2.89 — Fluid gauge with air leg below atmospheric pressure

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

water from passing to the fluid measuring column, then (h_s) is calculated as follows:

$$h_s = h_{gs}\frac{\gamma\ \text{measurement fluid}}{\gamma\ \text{water}} + \frac{v_s^2}{2g} + Z_s$$

2.6.10 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 ± 0.75% at the specified condition.

Readings of power shall be taken at the same time that rate of flow 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 — 1995.*

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 during 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.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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.

2.6.11 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 ±0.3%. 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.

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 handheld 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 50 or 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).

2.6.12 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 rate of flow.

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.

2.6.13 Model tests

2.6.13.1 Model test procedure

In many installations involving large pumps, model tests are often necessary. 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 300 mm (12 inches) 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 pump not only provides advance assurance of performance but makes design alterations possible in time for incorporation into the prototype pump.

Not all installations lend themselves to a practical model investigation. The pumping of water carrying considerable quantities of sand or other foreign material

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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 rate of flow 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 (Metric): A single stage pump designed to deliver 20,000 m^3/h against a head of 122 meters at 450 rpm and have an impeller diameter of 2 meters. 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 97 meters. The model impeller is to be 0.457 meters in diameter.

Determine speed and rate of flow 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

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

$$n_1 = n_2\left[\frac{D_2}{D_1}\right]\left[\frac{H_1}{H_2}\right]^{0.5}$$

$$n_1 = 450\left[\frac{2.0}{0.457}\right]\left[\frac{97}{122}\right]^{0.5} = 1756 \text{ rpm}$$

$$\frac{Q_1}{Q_2} = \left[\frac{D_1}{D_2}\right]^2\left[\frac{H_1}{H_2}\right]^{0.5}$$

or

$$Q_1 = Q_2\left[\frac{D_1}{D_2}\right]^2\left[\frac{H_1}{H_2}\right]^{0.5}$$

$$Q_1 = 20{,}000\left[\frac{0.457}{2.0}\right]^2\left[\frac{97}{122}\right]^{0.5} = 931 \text{ m}^3\text{/h}$$

The model pump should therefore be run at a speed of 1756 rpm delivering 4075 m^3/h against a head of 97 meters.

To check these results, it will be noted that the specific speed of the prototype is:

$$NS_d = \frac{n(Q)^{0.5}}{H^{0.75}} = \frac{450(20{,}000)^{0.5}}{122^{0.75}} = 1734$$

and the specific speed of the model will be

$$NS_d = \frac{1756 \times 931^{0.5}}{97^{0.75}} = 1734$$

Therefore, the specific speeds are the same as required.

Example (US Units): 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 in diameter.

Determine speed and rate of flow 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

$$n_1 = n_2\left[\frac{D_2}{D_1}\right]\left[\frac{H_1}{H_2}\right]^{0.5}$$

$$n_1 = 450\left[\frac{6.8}{1.5}\right]\left[\frac{320}{400}\right]^{0.5} = 1825 \text{ rpm}$$

$$\frac{Q_1}{Q_2} = \left[\frac{D_1}{D_2}\right]^2\left[\frac{H_1}{H_2}\right]^{0.5}$$

or

$$Q_1 = Q_2\left[\frac{D_1}{D_2}\right]^2\left[\frac{H_1}{H_2}\right]^{0.5}$$

$$Q_1 = 90{,}000\left[\frac{1.5}{6.8}\right]^2\left[\frac{320}{400}\right]^{0.5} = 3920 \text{ gpm}$$

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.

2.6.13.2 Model test 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.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

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.

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

Appendix A

References

American Society of Mechanical Engineers
United Engineering Center
345 East 47th Street
New York, NY 10017

Institute of Electrical and Electronics Engineers
1828 L Street, N.W., Suite 1202
Washington, DC 20036-5104

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved

Appendix B

Index

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

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

Hydraulic Institute Standards, Copyright © 1997-2005, All Rights Reserved