Approved American National Standard

ANSI

American National Standard for

Rotodynamic
Pumps

for Hydraulic Performance
Acceptance Tests

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Rotodynamic Pumps

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Approved August 9, 2011
American National Standards Institute, Inc.

American National Standard

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Contents

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Foreword (Not part of Standard)

This new ANSI/HI 14.6 pump test standard is a substantially revised and updated standard replacing ANSI/HI 1.6 *Centrifugal Pump Tests* and ANSI/HI 2.6 *Vertical Pump Tests*. It is in harmony with the revised ISO 9906 *Rotodynamic pumps - Hydraulic performance acceptance test* standard and ANSI/HI 11.6 *Submersible Pump Tests*. These three standards now have identical pump acceptance test criteria with worldwide acceptance. This means that users in all parts of the world, whether they are using this standard, ANSI/HI 11.6, or ISO 9906, when specifying a pump hydraulic performance acceptance test, will be working with identical technical requirements and acceptance grades.

Although quite different in both format and layout, the changes in this standard from ANSI/HI 1.6 and 2.6 are mostly in the area of acceptance requirements and informational details rather than in pure technical issues and test procedures. Test methods and procedures are similar in both the new and old ANSI/HI standards.

This standard is normative, meaning that what is written in the standard must be adhered to in order to comply with the standard. The appendices of this standard are either normative or informative; they are individually marked to clearly show their status. The normative appendices must be adhered to in order to comply with the standard, whereas informative appendices are written to inform and educate the user and do not require compliance.

Differences between the old (ANSI/HI 1.6 and ANSI/HI 2.6) and new (ANSI/HI 14.6) standards include:

- Old: Has two test acceptance levels, A and B. New: Has three levels of acceptance: Grade 1 with tighter tolerance band that can be applied in three acceptance grades (1U, 1E, 1B); Grade 2 with a broader tolerance band can be applied in two acceptance grades (2B, 2U); and Grade 3 with even broader tolerance band. Acceptance grades 1U and 2U have no negative tolerance.

- The new standard spells out measurement uncertainty and allowable measurement fluctuations in greater detail than the old standard.

- The new standard goes into a more detailed discussion of the tolerance band and what constitutes acceptance.

- The new standard allows a wider efficiency tolerance for pumps with an input power below 10 kW.

- The new standard defines industry-specific default test acceptance grades for cases where the user has not defined an acceptance grade.

- The new standard has an informative appendix that provides information about various types of measurement equipment.

- The new standard has an informative appendix that provides information about performing string tests.

- The new standard has an informative appendix that provides educational information about testing parameters and variations.

Purpose and aims of the Hydraulic Institute

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

a) To develop and publish standards for pumps;

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

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

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

e) To support educational and research activities;

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

Purpose of Standards

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

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

Definition of a Standard of the Hydraulic Institute

Quoting from Article XV, Standards, of the By-Laws of the Institute, Section B: "An Institute Standard defines the product, material, process or procedure with reference to one or more of the following: nomenclature, composition, construction, dimensions, tolerances, safety, operating characteristics, performance, quality, rating, testing, and service for which designed."

Comments from users

Comments from users of this standard will be appreciated to help the Hydraulic Institute prepare even more useful future editions. Questions arising from the content of this standard may be sent to the Technical Director of the Hydraulic Institute. The inquiry will then be directed to the appropriate technical committee for provision of a suitable answer.

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

Revisions

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

Units of measurement

In this standard all principal quantities and formulae, as well as charts and graphs, are given in coherent metric units. Corresponding US customary units appear in brackets. Example calculations are given in metric and US customary units, as appropriate.

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

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

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

2guysfromTU
A.R. Wilfley & Sons, Inc.
A.W. Chesterton Company
Brown and Caldwell
E.I. DuPont Company
ekwestrel corp
Fluid Sealing Association
GIW Industries, Inc.
Healy Engineering, Inc.
ITT - Industrial Process
ITT - Water & Wastewater
J.A.S. Solutions Ltd.
John Crane Inc.

Kemet Inc.
LVVWD - Las Vegas Valley Water District
National Pump Company
Peerless Pump Company
Pentair Water - Engineered Flow GBU
Powell Kugler, Inc.
Pump Design, Development & Diagnostics, LLC
Sulzer Pumps (US) Inc.
TACO, Inc.
Wasserman, Horton - Consultant
Weir Floway, Inc.
Weir Minerals North America
Weir Specialty Pumps

Committee list

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

Chair (2000 - 2011) - Stefan Abelin, ITT - Water & Wastewater
Chair (2011) - Al Iseppon, Pentair Water
Vice-chair (2000 - 2011) - Roger Turley, formerly of Flowserve
Vice-chair (2011) - Michael Coussens, Peerless Pump Company

Committee Members	Company
Graeme Addie	GIW Industries, Inc.
John Anspach	John Anspach Consulting
Charles Cappellino	ITT - Industrial Process
Jack Claxton	Patterson Pump Company
Michael Cropper	Sulzer Pumps (US) Inc.
James Healy	Healy Engineering, Inc.
Greg Highfill	A.R. Wilfley & Sons, Inc.
Allen Hobratschk	National Pump Company
David McKinstry	IMO Colfax
Rodney Mrkvicka	Smith & Loveless, Inc.
Michael Mueller	Flowserve Corporation
James Roberts	ITT - Residential & Commercial Water
Mark Rosebraugh	Yeomans Chicago Corporation
Aleksander Roudnev	Weir Minerals North America
Paul Ruzicka	ITT Residential and Commercial Water
Joseph Salah	Sulzer Process Pumps (US) Inc.
Arnold Sdano	Pentair Water
Ernest Sturtz	CDM
Greg Towsley	Grundfos North America
Fred Walker	Weir Floway, Inc.

14.6 Hydraulic performance acceptance tests

14.6.1 Scope

This standard covers hydraulic performance tests for acceptance of rotodynamic pumps (centrifugal, mixed flow, and axial flow pumps), in this document referred to as *pumps*.

ANSI/HI Standard 14.6 is intended to be used for pump acceptance testing at pump test facilities, such as manufacturers' pump test facilities or laboratories only. Industry experience shows that it is very difficult to perform measurements accurate enough to satisfy the acceptance requirements in this standard when testing is performed in the field.

Information in the standard may be applied to pumps of any size and to any pumped liquids behaving as clear water.

The standard includes three grades of accuracy of measurement: grade 1 for higher accuracy, and grades 2 and 3 for lower accuracy. These grades include different values for tolerance factors for allowable fluctuations and uncertainties of measurement.

This standard applies to a pump by itself without any fittings. The pump may also be tested with a combination of upstream and/or downstream fittings by prior agreement and agreed on contractually.

References to ANSI/HI 1.6 *Centrifugal Pump Tests* or ANSI/HI 2.6 *Vertical Pump Tests* in procurement documents and test specifications shall refer to ANSI/HI 14.6 for all applicable parts of the standard.

There are other pump acceptance test standards for submersible and sealless pumps, as defined in their respective documents, that take into account the unique features that those products exhibit. The Hydraulic Institute recommends that the user of this standard consult those respective standards (ANSI/HI 11.6 *Submersible Pump Tests* and ANSI/HI 4.1-4.6 *Sealless, Magnetically Driven Rotary Pumps for Nomenclature, Definitions, Application, Operation, and Test*) to determine if they are more appropriate for the products being considered for testing.

14.6.2 Terms and definitions

14.6.2.1 Introduction

For the purposes of this standard, the quantities, definitions, symbols, and units given here apply.

Pump performance acceptance grades for flow, head, efficiency, or power are used in this standard when evaluating acceptance of a pump for a guarantee point. A guarantee point, as defined in Section 14.6.3.1, is a flow/head (Q/H) point that a tested pump shall meet, within the tolerances of the agreed acceptance class, to be accepted according to this standard.

Table 14.6.2.1 gives definitions of quantities used in this standard. The definitions, particularly those given for head and net positive suction head (NPSH), may not be appropriate for general use in hydrodynamics, and are for the purpose of this standard only.

Table 14.6.2.2a gives an alphabetical list of symbols used, and Table 14.6.2.2b gives a list of subscripts.

In this standard all formulae are given in coherent metric units. For conversion of other units to metric units see Appendix L.

Table 14.6.2.1 — List of quantities

Row	Quantity	Definition[a]	Symbol	Dimension	Unit
3.1.1	Mass	The inertial resistance of a body to acceleration. A quantitative measure of inertia.	m	M	kg (lbm)
3.1.2	Length		l	L	m (ft)
3.1.3	Time	The measure of a sequence of events.	t	T	s
3.1.4	Temperature	A measurement of the average kinetic energy of the molecules in an object or system and can be measured with a thermometer or calorimeter. It is a means of determining the internal energy contained within the system.	Θ	Θ	°C (°F)
3.1.5	Area		A	L^2	m^2 (ft^2)
3.1.6	Volume	The quantity of three-dimensional space occupied by a liquid, solid, or gas.	V	L^3	m^3 (ft^3)
3.1.7	Angular velocity (Rotation)	Number of radians of a shaft per unit time, $\omega = 2\pi n$.	ω	T^{-1}	rad/s
3.1.8	Velocity		v	LT^{-1}	m/s (ft/s)
3.1.9	Acceleration due to gravity	See Note [b]. The acceleration of a body due to the influence of the pull of gravity alone.	g	LT^{-2}	m/s^2 (ft/s^2)
3.1.10	Speed of rotation	Number of rotations per unit time.	n	T^{-1}	s^{-1}, min^{-1}
3.1.11	Density	Mass per unit volume.	ρ	ML^{-3}	kg/m^3 (lbm/ft^3)
3.1.12	Pressure	Force per unit area. In this standard, all pressures are gauge pressures, i.e., measured with respect to the atmospheric pressure, except the atmospheric pressure and the vapor pressure, which are absolute pressures.	p	$ML^{-1}T^{-2}$	Pa (psi)
3.1.13	Kinematic viscosity	A coefficient defined as the ratio of the dynamic viscosity of a fluid to its density.	ν	L^2T^{-1}	m^2/s (ft^2/s)
3.1.14	Energy	The capability of doing work.	E	ML^2T^{-2}	J (ft·lbf)
3.1.15	Power (general term)	Energy transferred per unit time.	P	ML^2T^{-3}	W (hp)
3.1.16	Reynolds number	Defined, especially for this standard by the relation: $$Re = \frac{UD}{\nu}$$	Re	pure number	
3.1.17	Diameter		D	L	m (ft)

Table 14.6.2.1 — List of quantities (*continued*)

Row	Quantity	Definition[a]	Symbol	Dimension	Unit
3.1.18	Mass rate of flow	The mass rate of flow designates the external mass rate of flow of the pump, i.e., the rate of flow from the pump discharge. NOTE: Losses or abstractions inherent to the pump, such as: • discharge necessary for hydraulic balancing of axial thrust, • cooling of bearings of the pump itself, • flush water to the seal or packing, • leakage from the fittings, internal leakage, etc., are not to be reckoned in the rate of flow. On the contrary, all derived flows for other purposes, such as: • cooling of the motor bearings, • cooling of a gearbox (bearings, oil cooler), etc., are to be reckoned in the rate of flow. Whether and how these flows must be taken into account depends on the location of their derivation and of the section of flow measurement, respectively.	q^c	MT^{-1}	kg/s (lbm/s)
3.1.19	Volume rate of flow	The outlet volume rate of flow has the following value: $Q = \dfrac{q}{\rho}$ In this standard this symbol may also designate the volume rate of flow in any given section[d]. It is the quotient of the mass rate of flow in this section by the density. (The section may be designated by subscripts.)	Q^e	L^3T^{-1}	m^3/s (ft^3/s)
3.1.20	Mean velocity	The mean axial velocity of flow equal to the volume rate of flow[d] divided by the pipe cross-section area $U = \dfrac{Q}{A}$	U	LT^{-1}	m/s (ft/s)
3.1.21	Local velocity	Local velocity of flow at any point.	v	LT^{-1}	m/s (ft/s)
3.1.22	Head (general term)	The energy per unit mass of fluid, divided by acceleration due to gravity, g.	h	L	m (ft)

Table 14.6.2.1 — List of quantities (*continued*)

Row	Quantity	Definition[a]	Symbol	Dimension	Unit
3.1.23	Height above reference plane	The height of the considered point above the reference plane. Its value is: • positive, if the considered point is above the reference plane; • negative, if the considered point is below the reference plane. Reference plane: Any horizontal plane used as a datum for height measurement. For practical reasons it is preferable not to specify an imaginary reference plane.	z	L	m (ft)
		• Difference between NPSH datum plane (see 3.1.36) and reference plane.	z_D	L	m (ft)
3.1.24	Gauge pressure	The pressure relative to atmospheric pressure. Its value is: • positive, if this pressure is greater than the atmospheric pressure; • negative, if this pressure is less than the atmospheric pressure. NOTE: All pressures in this standard are gauge pressures read from a manometer or similar pressure-sensing instrument, except atmospheric pressure and the vapor pressure of the liquid, which are expressed as absolute pressures.	p	$ML^{-1}T^{-2}$	Pa (psi)
3.1.25	Atmospheric pressure (absolute)	The pressure exerted by the weight of the air above it at any point on the earth's surface.	p_{amb}	$ML^{-1}T^{-2}$	Pa (psi)
3.1.26	Vapor pressure (absolute)	The pressure exerted when a solid or liquid is in equilibrium with its own vapor. The vapor pressure is a function of the substance and the temperature.	p_v	$ML^{-1}T^{-2}$	Pa (psi)
3.1.27	Velocity head	The kinetic energy per unit mass of the liquid in movement, divided by g. It is expressed by $\dfrac{U^2}{2g}$	h_v	L	m (ft)
3.1.28	Total head	In any section, the total head is given by: $$H_x = z_x + \frac{p_x}{\rho g} + \frac{U_x^2}{2g}$$ where z is the height of the center of the cross section above the reference plane and p is the gauge pressure related to the center of the cross section. The absolute total head in any section is given by: $$H_{x(abs)} = z_x + \frac{p_x}{\rho g} + \frac{p_{amb}}{\rho g} + \frac{U_x^2}{2g}$$	H_x	L	m (ft)

Table 14.6.2.1 — List of quantities (*continued*)

Row	Quantity	Definition[a]	Symbol	Dimension	Unit
3.1.29	Inlet total head	Total head in the inlet section of the pump: $h_1 = z_1 + \dfrac{p_1}{\rho g} + \dfrac{U_1^2}{2g}$	h_1	L	m (ft)
3.1.30	Outlet total head	Total head in the outlet section of the pump: $h_2 = z_2 + \dfrac{p_2}{\rho g} + \dfrac{U_2^2}{2g}$	h_2	L	m (ft)
3.1.31	Pump total head	Algebraic difference between the outlet total head h_2 and the inlet total head h_1: $H = h_2 - h_1$ if compressibility is negligible. If the compressibility of the pumped liquid is significant, the density ρ should be replaced by the mean value: $\rho_m = \dfrac{p_1 + p_2}{2}$ and the pump total head should be calculated by the formula: $H = z_2 - z_1 + \dfrac{p_2 - p_1}{\rho_m \cdot g} + \dfrac{U_2^2 - U_1^2}{2g}$ For vertical turbine pumps, the pump total head includes the bowl assembly head minus all the pump internal hydraulic friction losses.	H	L	m (ft)
3.1.31.1	Bowl assembly total head	The bowl assembly total head is the gauge head measured at the gauge connection located on the column pipe downstream from the bowl assembly, plus the velocity head at the point of the gauge connection, plus any losses between top of bowl assembly to tap location, plus datum from gauge plus liquid level. Friction losses of suction piping, can (barrel), and strainers must also be added if significant. (h_j includes all losses before and after the bowl assembly.)	H_{ba}	L	m (ft)
3.1.32	Specific energy	Energy per unit mass of liquid. It is given by the equation: $y = gH$.	y	L^2T^{-2}	J/kg (Btu/lbm)
3.1.33	Loss of head at inlet	The difference between the total head of the liquid at the measuring point and the total head of the liquid in the inlet section of the pump.	h_{J1}	L	m (ft)
3.1.34	Loss of head at outlet	The difference between the total head of the liquid in the outlet section of the pump and the total head of the liquid at the measuring point.	h_{J2}	L	m (ft)
3.1.35	Pipe friction loss coefficient	Coefficient for the head loss by friction in the pipe.	λ	pure number	

Table 14.6.2.1 — List of quantities (*continued*)

Row	Quantity	Definition[a]	Symbol	Dimension	Unit
3.1.36	Net positive suction head, NPSH	Net positive suction head is the absolute inlet total head above the head equivalent to the vapor pressure referred to the NPSH datum plane. $$(NPSH) = h_1 - z_D + \frac{p_{amb} - p_v}{\rho_1 g}$$ This NPSH is referred to the *NPSH datum plane*, whereas inlet total head is referred to the *reference plane*. NPSH datum plane: The horizontal plane through the center of the circle described by the external points of the entrance edges of the impeller blades; in the first stage in the case of multistage pumps. In the case of double inlet pumps with vertical or inclined axis, it is the plane through the higher center. The manufacturer should indicate the position of this plane with respect to precise reference points on the pump. Examples: See Figure 14.6.2.1.	*NPSH*	L	m (ft)
3.1.36.1	NPSH Available (NPSHA)	The NPSH available as determined by the conditions of the installation for a specified rate of flow.	*NPSHA*	L	m (ft)
3.1.36.2	NPSH Required (NPSHR)	A minimum NPSH given by the manufacturer/ supplier for a pump achieving a specified performance at the specified rate of flow, speed, and pumped liquid (occurrence of visible cavitation, increase of noise and vibration due to cavitation, beginning of head or efficiency drop, head or efficiency drop of a given amount, limitation of cavitation erosion).	*NPSHR*	L	m (ft)
3.1.36.3	NPSH3	NPSH3 is defined as the value of NPSHR at which the first-stage total head drops by 3% due to cavitation.	*NPSH3*	L	m (ft)
3.1.37	Type number[f]	Dimensionless quantity calculated at the point of best efficiency, which is defined by the following formula: $$K = \frac{2\pi n Q'^{0.5}}{(gH')^{0.75}} = \frac{\omega Q'^{0.5}}{(y')^{0.75}}$$ where Q' is the volume rate of flow per eye and H' is the head of the first stage. NOTE: The type number (specific speed)[f] is to be taken at maximum diameter of the first-stage impeller.	*K*	pure number	

Table 14.6.2.1 — List of quantities (*continued*)

Row	Quantity	Definition[a]	Symbol	Dimension	Unit
3.1.38	Pump power input	The power transmitted to the pump by its driver. For vertical turbine pumps, the pump power input is the power required to drive the complete pump assembly, including the bowl assembly input power, the lineshaft bearing power loss, stuffing-box loss, and thrust bearing loss.	P	ML^2T^{-3}	W (hp)
3.1.38.1	Bowl assembly power input	Power delivered to the bowl assembly. This is the pump power input minus any losses external to the bowl assembly.	P_{ba}	ML^2T^{-3}	W (hp)
3.1.39	Pump power output	The mechanical power transferred to the liquid as it passes through the pump, also known as *pump hydraulic power*. $P_u = \rho Q g H = \rho Q y$	P_u	ML^2T^{-3}	W (hp)
3.1.40	Driver power input	The power absorbed by the pump driver.	P_{gr}	ML^2T^{-3}	W (hp)
3.1.41	Rated shaft power	The maximum pump shaft power, as set by the manufacturer, that is adequate to drive the pump over the specified operating conditions.	P_{max}	ML^2T^{-3}	W (hp)
3.1.42	Pump efficiency	$\eta = \dfrac{P_u}{P} = \dfrac{Pump\ power\ output}{Pump\ power\ input}$	η	pure number	
3.1.42.1	Bowl efficiency	$\eta_{ba} = \dfrac{P_{u_{ba}}}{P_{ba}} = \dfrac{Bowl\ power\ output}{Bowl\ power\ input}$	η_{ba}	pure number	
3.1.43	Overall efficiency	$\eta_{gr} = \dfrac{P_u}{P_{gr}} = \dfrac{Pump\ power\ output}{Driver\ power\ input}$	η_{gr}	pure number	

[a] In order to avoid any error of interpretation, it is deemed desirable to reproduce the definitions of quantities and units as given in ISO 31 and to supplement these definitions by some specific information on their use in this standard.

[b] In principle, the local value of g should be used. Nevertheless, for grade 2 it is sufficient to use a value of 9.81 m/s². For the calculation of the local value g = 9.7803 (1 + 0.0053 $\sin^2 \varphi$) – 3 · 10⁻⁶ · z, where φ = latitude and z = altitude.

[c] An optional symbol for mass rate of flow is q_m.

[d] Attention is drawn to the fact that in this case Q may vary for different reasons across the circuit.

[e] An optional symbol for volume rate of flow is q_v

[f] See Appendix L, Table L.1, for conversion to specific-speed terms.
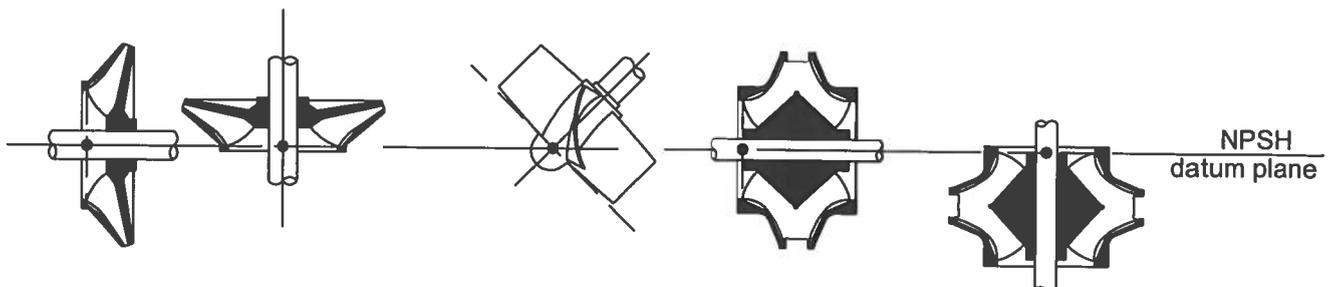
Figure 14.6.2.1 — Datum elevation for various pump designs at eye of first-stage impeller

14.6.2.2 Lists of basic letters and subscripts

Table 14.6.2.2a — Alphabetical list of basic letters used as symbols

Symbol	Quantity	Unit
A	Area	m^2 (ft^2)
D	Diameter	m (ft)
E	Energy	J (ft·lbf)
e	Overall uncertainty, relative value	(%)
f	Frequency	s^{-1}, Hz
g	Acceleration due to gravity	m/s^2 (f/s^2)
H	Pump total head	m (ft)
h_J	Losses in terms of head of liquid	m (ft)
k	Equivalent uniform roughness	m (ft)
K	Type number	
l	Length	m (ft)
m	Mass	kg (lbm)
n	Speed of rotation	s^{-1}, min^{-1}
$NPSH$	Net positive suction head	m (ft)
p	Pressure	Pa (psi)
P	Power	W (hp)
q	Mass rate of flow	kg/s (lb/s)
Q	(Volume) rate of flow	m^3/s (ft^3/s)
Re	Reynolds number	
t	Tolerance factor, relative value	(%)
t	Time	s
T	Torque	N·m (lbf·ft)
U	Mean velocity	m/s (ft/s)
v	Local velocity	m/s (ft/s)
V	Volume	m^3 (ft^3)
y	Specific energy	J/kg (Btu/lbm)
z	Height above reference plane	m (ft)
η	Efficiency	
θ	Temperature	°C (°F)
λ	Pipe friction loss coefficient	
ν	Kinematic viscosity	m^2/s (ft^2/s)
ρ	Density	kg/m^3 (lbm/ft^3)
ω	Angular velocity	rad/s

Table 14.6.2.2b — List of letters and numbers used as subscripts

Subscript	Meaning
1	Inlet
1'	Inlet measuring section
2	Outlet
2'	Outlet measuring section
abs	Absolute
amb	Ambient
ba	Bowl assembly
D	Difference, datum
f	Fluid in measuring pipes
G	Guaranteed
gr	Combined motor/pump unit (overall)
H	Pump total head
L	Liquid level
m	Mean
M	Manometer
n	Speed of rotation
P	Power
Q	Rate of flow (volume)
sp	Specified
T	Translated, torque
u	Useful
v	Vapor (pressure)
η	Efficiency
x	At any section

14.6.2.3 Factory performance tests

14.6.2.3.1 General

Pump tests are performed to verify the initial performance of new pumps as well as checking for repeatability of production units, accuracy of impeller trim calculations, performance with special materials, etc. A typical performance test consists of measurement of flow, head, and power input to the pump or pump test motor. Additional optional tests, such as NPSH, may be included as agreed on. A factory test is understood to mean testing at a dedicated test facility, often at a pump manufacturer's plant or at an independent pump test facility.

14.6.2.3.2 Nonwitnessed pump test

14.6.2.3.2.1 Factory test

Nonwitnessed factory tests are performed without the presence of a purchaser representative. The pump manufacturer is responsible for the data collection and judgment of pump acceptance. The advantages of this test are cost savings and accelerated pump delivery to the pump user. In many cases, if the purchaser is familiar with the performance of the pump (e.g., an identical pump model order), then a factory nonwitnessed test may be acceptable.

14.6.2.3.2.2 Certified factory test

Nonwitnessed certified factory tests are performed without the presence of a purchaser representative. The pump manufacturer is responsible for compliance with this test standard. The pump manufacturer conducts the test, passes judgment of pump acceptance, and produces a certified and signed pump test document. The advantages of this test are the same as the nonwitnessed factory test, i.e., the test is substantially less expensive and often leads to accelerated pump delivery to the end user.

14.6.2.3.3 Witnessed pump test

The witnessing of a pump test by a representative of the pump purchaser can serve many useful functions. There are various ways of witnessing a test.

14.6.2.3.3.1 Witnessing by the purchaser's representative

A representative of the purchaser physically attends the testing. The representative signs off on the raw test data to certify that the test is performed satisfactorily. Final acceptance of the pump performance may or may not be determined by the witness. The benefit of witness testing depends largely on the effectiveness and expertise of the witness. A witness cannot only ensure the test is conducted properly, but also observe operation of the pump during testing prior to pump shipment to the jobsite. A disadvantage to witness testing can be extended delivery times and substantially higher cost. With today's "just-in-time" manufacturing methods, scheduling of witness testing requires flexibility on the part of the witness, and may lead to additional costs if the schedule of the witness causes delays in manufacturing.

14.6.2.3.3.2 Remote witness by the purchaser's representative

Pump performance testing may be remotely witnessed by the purchaser or its representative. With a remote camera system, the purchaser can monitor the entire testing remotely in real time. The raw data as recorded by the data acquisition system can be viewed and analyzed during the test, and the results can be discussed and submitted for approval. The advantages of this type of testing are savings in travel costs and accelerated pump delivery.

14.6.3 Pump acceptance tests

14.6.3.1 General

The specified and contractually agreed on rated point (duty point), hereafter called the *guarantee point*, shall be evaluated against one acceptance tolerance grade. For a pump performance test, this guarantee point must always specify guaranteed flow Q_G and guaranteed head H_G, and may also specify guaranteed efficiency, guaranteed shaft power, or guaranteed NPSHR. When applicable, these optional guarantee parameters need to be specified for those tests, see respective test Sections 14.6.3.2 and 14.6.5.8.

The acceptance tolerance grade applies to the guarantee point only. For cases where a guarantee point is given, but no acceptance grade is specified, this standard reverts to a default test acceptance grade, as specified in Table 14.6.4. Other specified duty points, including their tolerances, shall be per separate agreement between the manufacturer and purchaser. If other specified duty points are agreed on, but no tolerance is given for these points, then the default acceptance grade for these points shall be grade 3B.

A guarantee point can be detailed in a written contract, in a customer-specific pump performance curve, or in similar written and project-specific documentation.

If not otherwise agreed on between the manufacturer/supplier and the purchaser, the following shall apply:

a) Acceptance grade tolerances according to the grades given in Table 14.6.3.4.

b) Tests shall be carried out on the test stand of the manufacturer's pump test facilities or laboratories with clear water by the methods and in the test arrangements specified in this standard.

c) The pump performance shall be guaranteed between the pump's inlet flange and discharge flange.

d) Pipe and fittings (bends, reducers, and valves) outside of the pump are not a part of the guarantee.

The combination of manufacturing and measurement tolerances in practice necessitates the usage of tolerances on tested values. The tolerances given in Table 14.6.3.4 have been established as a means to mitigate the effects of both manufacturing and measurement tolerances. See Appendix C for details.

The performance of a pump varies substantially with the nature of the liquid being pumped. Although it is not possible to give general rules whereby performance with clear water can be used to predict performance with other liquids, it is desirable for the parties to agree on empirical rules to suit the particular circumstances.

When a number of identical pumps are to be purchased, the number of pumps to be tested shall be agreed on by the purchaser and manufacturer/supplier.

Both purchaser and manufacturer/supplier shall be entitled to witness the testing, by prior agreement. When tests are not carried out at the manufacturer's test stand, opportunity shall be allowed for verification of pump installation and instrumentation adjustments by both parties.

14.6.3.2 Guarantees

The manufacturer guarantees that, for the guarantee point and at the rated speed (or in some cases frequency and voltage), the measured pump curve will touch or pass through a tolerance band surrounding the guarantee point as defined by the applicable acceptance grade, see Table 14.6.3.4 and Figures 14.6.3.4.2a and b.

A guarantee point shall be defined by a guarantee flow Q_G and a guarantee head H_G.

In addition, one or more of the following quantities may be guaranteed at the specified conditions and the rated speed:

a) 1) The minimum pump efficiency η_G or the maximum pump input power P_G, or

     2) In the case of a combined pump and motor unit, the minimum combined efficiency η_{grG} or the maximum pump motor unit input power P_{grG}.

                                        (Defined in Section 14.6.3.4.3 and Figures 14.6.3.4.3a, b, and c.)

b) The maximum required net positive suction head (NPSHR) at the guarantee flow.

The maximum power input may be guaranteed for the guarantee point or for a range of points along the pump curve. This, however, may require larger tolerance bands to be agreed on by the purchaser and manufacturer/supplier.

14.6.3.3 Measurement uncertainty

14.6.3.3.1 General

Every measurement is inevitably subject to some uncertainty, even if the measuring procedures, instruments used, and methods of analysis fully comply with good practice and with the requirements of this standard.

14.6.3.3.2 Permissible fluctuations (random fluctuations)

For each quantity to be measured, Table 14.6.3.3.2 gives the permissible amplitude of fluctuations per acceptance grades 1, 2, and 3 (see Section 14.6.3.4).

Where the construction or operation of a pump is such that fluctuations of great amplitude are present, measurements may be carried out by providing a damping device in the measuring instruments, their connecting lines, or by electronic data averaging. This can reduce the amplitude of the fluctuations to within the values given in Table 14.6.3.3.2.

Since it is possible that damping will affect the accuracy of the readings, use shall be made of symmetrical and linear damping devices, for example, a capillary tube, which must provide an integration over at least one complete cycle (peak-to-peak) of fluctuations.

Table 14.6.3.3.2 — Permissible amplitude of fluctuation as a percentage of mean value of quantity being measured

Measured quantity	Permissible amplitude of fluctuations per grade		
	Grade 1	Grade 2	Grade 3
Rate of flow	± 2%	± 3%	± 6%
Differential head	± 3%	± 4%	± 10%
Discharge head	± 2%	± 3%	± 6%
Suction head	± 2%	± 3%	± 6%
Input power	± 2%	± 3%	± 6%
Speed of rotation	± 0.5%	± 1%	± 2%

Table 14.6.3.3.2 — Permissible amplitude of fluctuation as a percentage of mean value of quantity being measured (*continued*)

Measured quantity	Permissible amplitude of fluctuations per grade		
	Grade 1	Grade 2	Grade 3
Torque	± 2%	± 3%	± 6%
Temperature	± 0.3 °C	± 0.3 °C	± 0.3 °C

14.6.3.3.3 Maximum permissible measurement device uncertainty (systematic)

The uncertainty of a measurement depends on the residual uncertainty of the measurement device and on the method of measurement used. After all known errors have been removed by zero adjustment, calibration, careful measurement of dimensions, proper installation, etc., this remains an uncertainty that never disappears. This uncertainty cannot be reduced by repeating the measurements if the same instrument and the same method of measurement is used.

Table 14.6.3.3 shows maximum permissible measurement device uncertainty for the different acceptance grades. It is important to note that these maximum uncertainty values pertain to the measurements at the guarantee point. Many measurement devices have their uncertainty based on their full-scale capability and, when practically applied, the actual measurement uncertainty can be two to four times higher. This means that the measurement device must often have a correspondingly higher accuracy (lower uncertainty).

Appendix I describes different methods of measurement and devices that typically are used to determine rate of flow, pump total head, speed of rotation, and pump power input in the range of accuracy required for tests according to grades 1, 2, and 3.

After having selected an appropriate measurement device and setup, the best assurance of an accurate measurement is obtained by ensuring that a zero adjustment is performed regularly and that device calibration is performed at proper intervals.

Table 14.6.3.3.3 — Maximum permissible measurement device uncertainty at guarantee point

Measured Quantity	Maximum permissible measurement device uncertainty at guarantee point per grade	
	Grade 1	Grade 2 and 3
Rate of flow	± 1.5%	± 2.5%
Differential head	± 1.0%	± 2.5%
Discharge head	± 1.0%	± 2.5%
Suction head	± 1.0%	± 2.5%
Suction head for NPSH testing	± 0.5%	± 1.0%
Driver power input	± 1.0%	± 2.0%
Speed of rotation	± 0.35%	± 1.4%
Torque	± 0.9%	± 2.0%

14.6.3.3.4 Overall measurement uncertainty

The fluctuation due either to the characteristics of the measuring system or to variations of the measured quantity, or both, appears directly as a scatter of the measurements. Unlike the systematic uncertainty, the fluctuation can be reduced by increasing the number of measurements of the same quantity under the same conditions.

The overall measurement uncertainty is calculated by the square root of the sum of the squares of the systematic and random uncertainties (fluctuations).

14.6.3.4 Performance test acceptance grades and tolerances

Six pump performance test acceptance grades: 1B, 1E, 1U, 2B, 2U, and 3B are defined below.

Grade 1 is the most stringent, with acceptance grade 1U having a unilateral tolerance band and 1B having a bilateral tolerance band. Acceptance grade 1E is also bilateral and is important to those concerned with energy efficiency. Grades 2 and 3 have wider tolerance bands, with 2B and 3B being bilateral and 2U unilateral. Note that all grade 1 tolerances have the same tolerance bandwidth for flow and head; the same is true for grades 2 and 3.

The purchaser and manufacturer can agree to use any grade to judge if a specific pump will meet a guarantee point. If a guarantee point is given, but no acceptance grade is specified, then this standard reverts to a default test acceptance grade, as described in Section 14.6.4.

Guarantee point acceptance grades for pump head, flow, power, and efficiency are provided in Table 14.6.3.4.

Table 14.6.3.4 — Pump test acceptance grades and corresponding tolerance band

Test parameter	Guarantee requirement	Grade	Grade 1			Grade 2		Grade 3
		Δt_Q	10%			16%		18%
		Δt_H	6%			10%		14%
		Symbol	Acceptance grade					
			1B	1E	1U	2B	2U	3B
Rate of flow	Mandatory	t_Q (%)	± 5%	± 5%	0% to + 10%	± 8%	0% to +16%	± 9%
Total head	Mandatory	t_H (%)	± 3%	± 3%	0% to + 6%	± 5%	0% to +10%	± 7%
Power[a]	Optional (either/or)	t_P (%)	+ 4%	+ 4%	+ 10%	+ 8%	+ 16%	+ 9%
Efficiency[a]		t (%)	− 3%	− 0%	− 0%	− 5%	− 5%	− 7%

[a] The power and efficiency tolerances are not the result of an exact calculation using the maximum values of a related column. They are instead reflecting real life experience. For grade 1E and 1U, no negative tolerance on efficiency is allowed.

NOTE: All tolerances are percentages of values guaranteed.

Other specified duty points, including their tolerances, shall be per separate agreement between the supplier and buyer. If other specified duty points are agreed on, but no tolerance is given for these points, then the default acceptance grade for these points shall be grade 3B.

14.6.3.4.1 Tolerances for pumps with an input power of 10 kW (13 hp) and below

For pumps with a shaft power input up to 10 kW (13 hp) but larger than 1 kW (1.3 hp), the tolerance factors given in Table 14.6.3.4 may be too stringent. If not otherwise agreed on between manufacturer and purchaser, the tolerance factors shall be as follows for pumps falling within this range:

rate of flow $\quad t_Q = \pm\, 10\%$

pump total head $\quad t_H = \pm\, 8\%$

If efficiency is guaranteed, the tolerance factor on efficiency t_η shall be calculated as follows:

$$t_\eta = -\left[10\left(1 - \frac{P}{10}\right) + 7\right]\%$$

where P is the maximum shaft power in kW over the range of operation.

If power is guaranteed, the tolerance factor on power t_P will be calculated as follows:

$$t_P = \sqrt{(7\%)^2 + t_\eta^2 \%}$$

For pumps with shaft power input of 1 kW (1.3 hp) and below, special agreements should be made between the parties.

14.6.3.4.2   Evaluation of flow and head

Guarantee point evaluation shall be performed at the rated speed. Test points do not have to be recalculated based on speed in cases where the test speed is identical to the rated speed and for tests with a combined motor and pump (i.e., submersible pumps, close-coupled pumps, and all pumps tested with a job motor). For tests in which the test speed is different from the rated speed, each test point has to be recalculated to the rated speed, using the affinity rules.

The tolerances for flow and head shall be applied in the following manner:

- The pump flow tolerance shall be applied to the guarantee flow Q_G, at the guaranteed head H_G

- The pump head tolerance shall be applied to the guarantee head H_G, at the guaranteed flow Q_G

Acceptance is achieved when either flow or head (or both) are found to be within the applicable tolerance band(s), see Figures 14.6.3.4.2a and b.

14.6.3.4.3   Evaluation of efficiency or power

If efficiency or power has been guaranteed, then it shall be evaluated against the applicable acceptance grade tolerance factor, i.e., the same as for Q/H in the following manner:

After a best-fit test curve (Q-H-/Q-η/or Q-P-curves) has been drawn and smoothly fitted through the measured test points, an additional straight line shall be drawn between the origin (0 rate of flow, 0 head) and the guarantee point (rate of flow/head). If necessary this line shall be extended until it crosses the fitted test curve. The intersection between the smoothly fitted test curve and this straight line shall form the new rate of flow/head point that is used for evaluation of efficiency or power. The measured input power or calculated efficiency at this point shall be compared to the guaranteed value and the applicable power or efficiency tolerance factors. See Figures 14.6.3.4.3a, b, and c.

> **NOTE 1:** The reason for using the "line from origin" method when evaluating the guaranteed efficiency or power is that it best retains the pump characteristics if the impeller diameter is changed. Additionally this method will always give one single point of reference for evaluation, and the line can be seen as a fair approximation of the system curve that the pump will eventually operate against.

> **NOTE 2:** If power is guaranteed, then the actual upper flow and/or head tolerance limit may be reduced depending on the power guarantee.

> **NOTE 3:** If the water temperature during the test is substantially different from the specified guarantee condition, the manufacturer and the purchaser shall beforehand mutually agree on any efficiency correction that may be appropriate; see Appendix K (model testing).
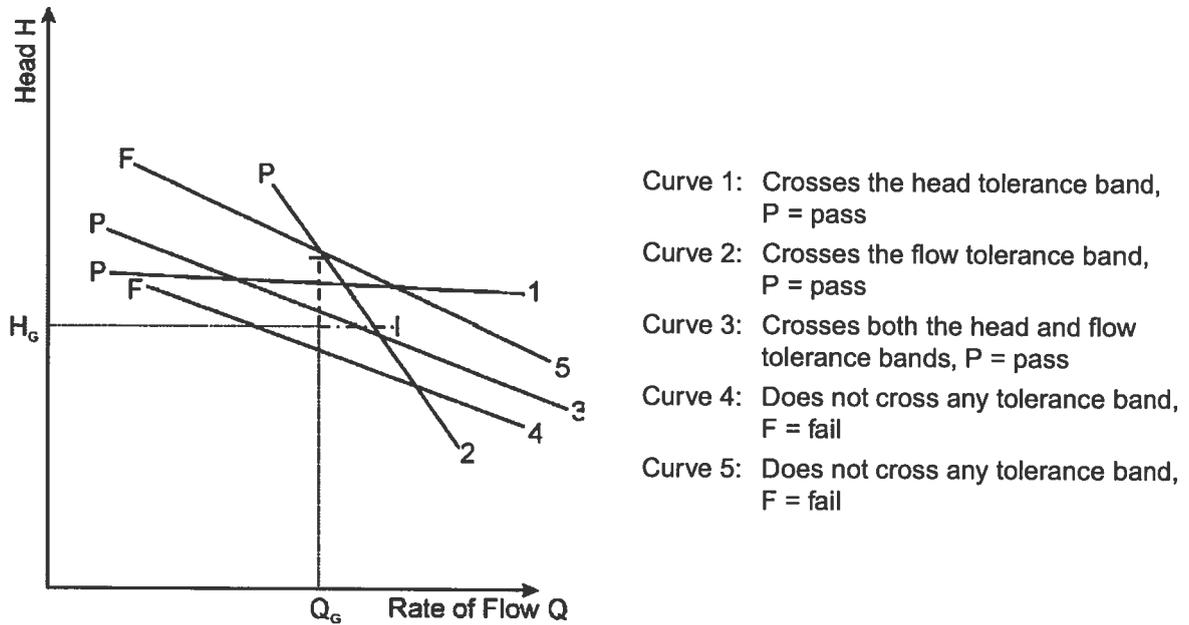
Figure 14.6.3.4.2a — Unilateral tolerance acceptance

Curve 1: Crosses the head tolerance band, P = pass

Curve 2: Crosses the flow tolerance band, P = pass

Curve 3: Crosses both the head and flow tolerance bands, P = pass

Curve 4: Does not cross any tolerance band, F = fail

Curve 5: Does not cross any tolerance band, F = fail
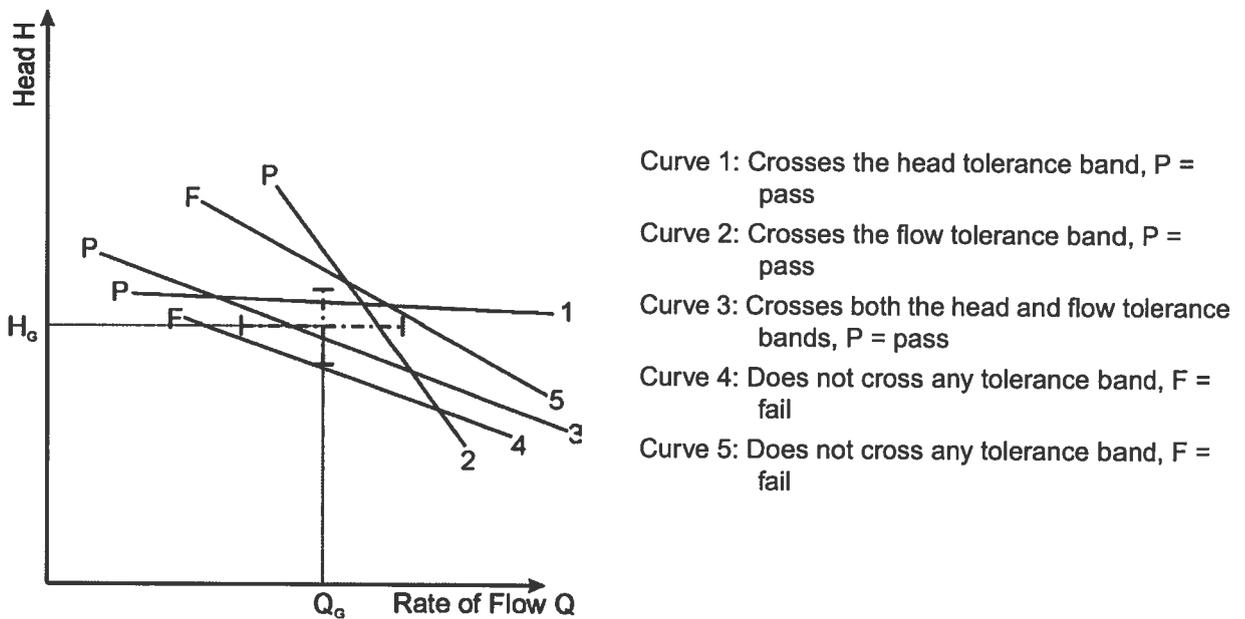
Figure 14.6.3.4.2b — Bilateral tolerance acceptance

Curve 1: Crosses the head tolerance band, P = pass

Curve 2: Crosses the flow tolerance band, P = pass

Curve 3: Crosses both the head and flow tolerance bands, P = pass

Curve 4: Does not cross any tolerance band, F = fail

Curve 5: Does not cross any tolerance band, F = fail

× Measuring points
1: H(Q) 2: η(Q) 3: P(Q)
∘ Guarantee point
• Constructed result
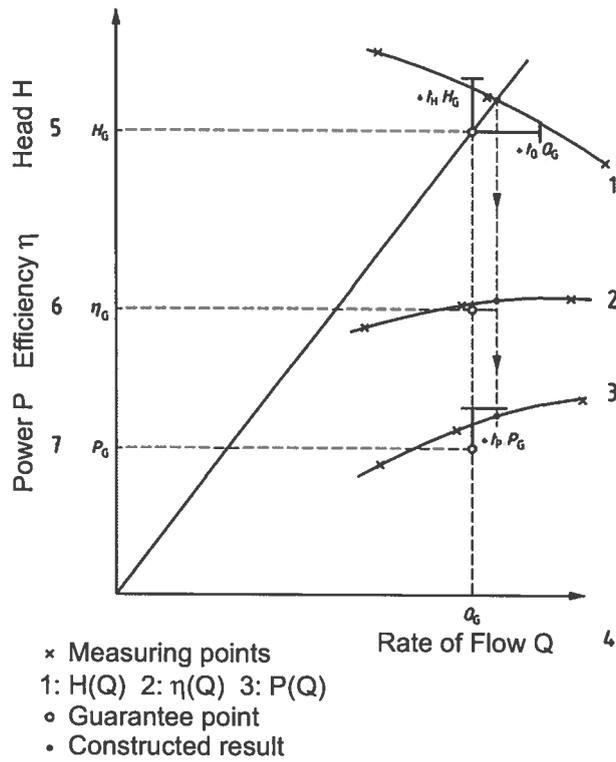
Figure 14.6.3.4.3a — Tolerance field for acceptance grades 1U and 2U

× Measuring points
1: H(Q) 2: η(Q) 3: P(Q)
∘ Guarantee point
• Constructed result
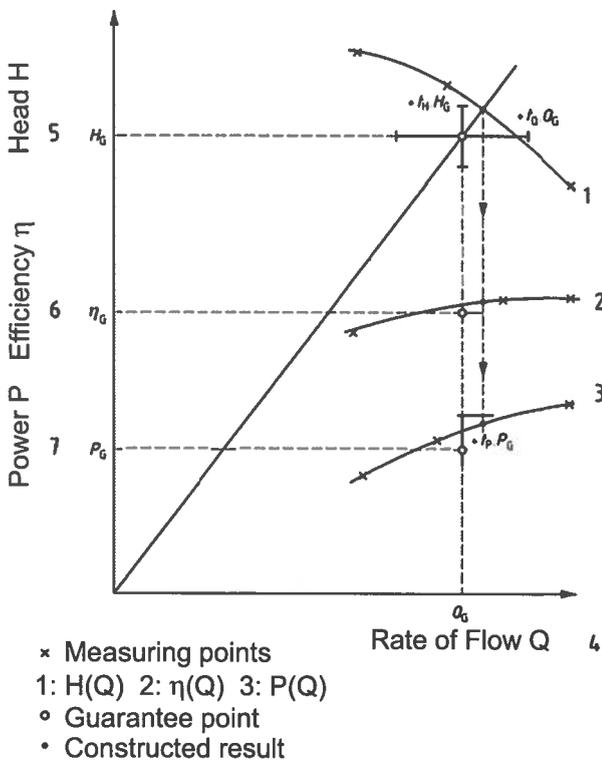
Figure 14.6.3.4.3b — Tolerance field for acceptance grade 1E

× Measuring points
1: H(Q) 2: η(Q) 3: P(Q)
∘ Guarantee point
• Constructed result

Figure 14.6.3.4.3c — Tolerance field for acceptance grades 1B, 2B, and 3B

14.6.4 Default test acceptance grades

For cases where a guarantee point is given, but no acceptance grade is specified, this standard reverts to a default test 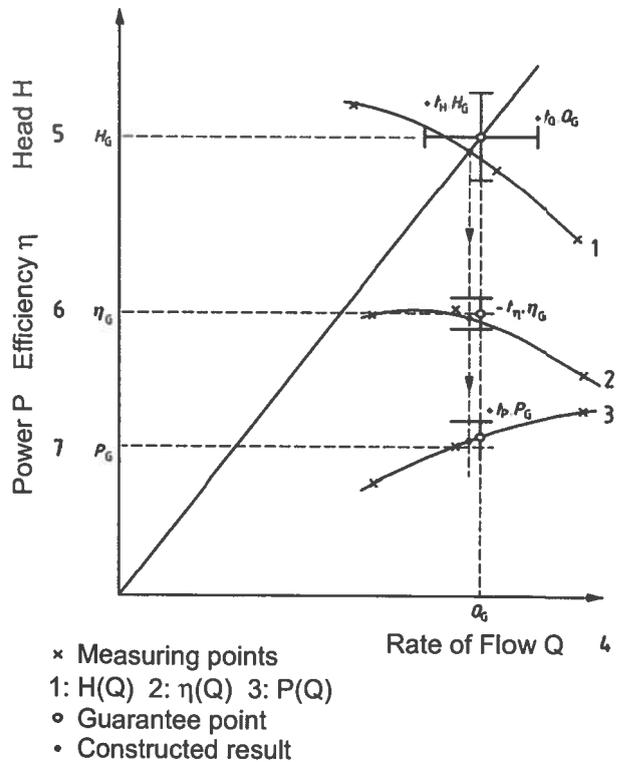acceptance grade, as specified in Table 14.6.4, whereby only flow and head will be guaranteed. Table 14.6.4 specifies the applicable acceptance grade for a pump based on the pump's rated shaft power and the purchaser's intended service for the pump. The purchaser is encouraged to specify its own preferred acceptance grade at the time that a guarantee point is agreed on. When this is done, it takes precedence over any classification provided by this table, and this section and table shall not be used.

Table 14.6.4 — Default acceptance grade based on purchaser's intended service

Application		Rated shaft power of pump	
		> 10 to 100 kW (13 to 134 hp)	> 100 kW (134 hp)
Municipal water and wastewater		2B	1B
Building trades and HVAC		2B	1B
Electric power industry		1B	1B
Oil and gas industry	API pumps	1B	1B
	Pipeline	1B	1B
	Water injection	Not applicable	1B
Chemical industry		2B	2B
Cooling tower		2B	2B
Pulp and paper		2B	2B
Slurry		3B	3B
General industry		3B	2B
Dewatering, drainage, and irrigation		3B	2B
Pumps not listed above		3B	2B

NOTE: This table only applies to situations where the purchaser and manufacturer have agreed to a guarantee point, but no test acceptance grade has been specified.

Other specified duty points, including their tolerances, shall be per separate agreement between the manufacturer and purchaser. If other specified duty points are agreed on, but no tolerance is given for these points, then the default acceptance grade for these points shall be grade 3B.

14.6.5 Test procedures

14.6.5.1 General

This standard is intended for tests conducted at pump test facilities, such as manufacturers' pump test facilities or laboratories. Special agreement is necessary for performance tests on-site providing all the requirements of this standard can be satisfied. It is, however, recognized that the conditions at most sites typically preclude full compliance with this standard. In these instances, site performance tests may still be acceptable providing the parties have agreed on how allowances are made for the added inaccuracies that will inevitably result from deviation from the specified requirements of this document.

14.6.5.2 Date of testing

For witness testing, the date of testing shall be mutually agreed on by the manufacturer/supplier and the purchaser.

14.6.5.3 Test procedure

In case of witness tests, the program and procedure to be followed in the test shall be submitted to the purchaser in ample time for consideration and agreement.

Test data other than the guaranteed, determined during the tests, shall have merely an indicative (informative) function.

14.6.5.4 Testing equipment

The test instrumentation used shall be documented and this information shall be made available to the customer upon request. Instruments shall be periodically calibrated. Guidance for a suitable period between calibrations of test instruments is given in Appendix J.

14.6.5.5 Records and report

A complete set of records, written or electronic, shall be kept on file for a minimum of five years.

In the case of witness tests, all test records shall be initialed by representatives of the parties witnessing the test, each of whom shall be provided with a copy of all records.

The test results shall be evaluated to the extent possible while the tests are in progress. In order that questionable measurements can be reevaluated, it is advisable that the installation and instrumentation remain intact until accurate data is obtained.

If required, the test results shall be summarized in a report. Further guidance regarding the contents of a test report and a suitable pump test sheet is given in Appendix H.

14.6.5.6 Test arrangements

The conditions necessary to ensure satisfactory measurement of the characteristics of operation are defined here, taking into account the accuracy required for tests of grades 1, 2, and 3.

NOTE 1: The performance of a pump in a given test arrangement, however accurately measured, may not be repeatable in another installation.

NOTE 2: Recommendations and general guidance about suitable pipe arrangements to ensure satisfactory measurements for flow and head are given in Appendix A and, if necessary, they can be used in conjunction with the ISO Standards on measurement of flow rates in closed conduits concerning the different methods (see Appendix M).

14.6.5.7 Test conditions

14.6.5.7.1 Test procedure

The duration of the test shall be sufficient to obtain repeatable results.

All measurements shall be made under steady state conditions (see Appendix A). If not otherwise specified, the tests have to be conducted under conditions where cavitation does not affect the performance of the pump.

A minimum of five test points shall be taken for all performance tests, regardless of acceptance grade, with one of the points being within −5% and 0%, and one being within 0% and +5% of the guarantee point flow rate. The other three points shall be spaced over the allowable operating range of the pump performance curve, with points taken near the maximum allowable head and flow regions.

NOTE: Other test procedures apply to NPSH tests. See Appendix F.

14.6.5.7.2   Speed of rotation during test

Unless otherwise agreed, tests may be carried out at a speed of rotation within the range of 50 to 120% of the test point(s) specified speed of rotation to establish rate of flow, pump total head, and power input. The efficiency may be affected if testing with different speeds. However, if the variation of speed is within 20% of the specified speed, then the efficiency change is considered negligible.

For NPSH tests, the speed of rotation should lie within the range of 80 to 120% of the specified speed of rotation, provided that the rate of flow lies within 50 and 120% of the rate of flow corresponding to the maximum efficiency at the test speed of rotation.

14.6.5.8   NPSH tests

14.6.5.8.1   Objective of NPSH tests

The objective of the NPSH test is to verify the pump's NPSHR for the guarantee. This test deals only with measurements relating to the hydraulic performance of the pump (variations of head, flow, and power) and not with other effects that can be caused by cavitation (e.g., noise, vibration, and erosion).

Cavitation effects may be detected as a drop in head or power at a given rate of flow. Unless otherwise specified, a 3% drop in head (the accepted industry practice) will be used to determine NPSHR and defined as NPSH3. In the case of multistage pumps, the head drop shall be referred to the head of the first stage, which should be measured if accessible. For very low head pumps, a head drop larger than 3% may be agreed on.

In most cases, cavitation tests will be conducted with deaerated clear water. Cavitation tests in water cannot accurately predict the behavior of the pump with liquids other than clear water.

NOTE: Air content may have a significant effect on measured NPSHR values and should be considered.

14.6.5.8.2   NPSH test types

14.6.5.8.2.1   Type I test: Determination of NPSH3 for multiple flow rates

In this test NPSHA is reduced progressively until the drop of the total head at constant flow rate reaches 3%. This value of NPSHA is the NPSH3 (see Table 14.6.5.8.2.1). A minimum of four different suitably spaced flow rates shall be evaluated within the allowable operation region.

14.6.5.8.2.2   Type II test: Determination of NPSH3 for a single flow rate

In this test NPSHA is reduced progressively until the drop of the total head of 3% at specified constant flow rate can be determined. This value of NPSHA is NPSH3 (see Table 14.6.5.8.2.1).

14.6.5.8.2.3   Type III test: Verification of limited influence of cavitation on the performance at specified NPSHA

Verification is made at the specified NPSHA to show that the total head at rated flow of the pump is not affected by cavitation more than 3%.

14.6.5.8.2.4 Type IV test: Verification of guaranteed characteristics at specified NPSHA

The pump meets the requirements if the guaranteed pump total head and power are obtained under the specified rate of flow and under the specified NPSHA.

14.6.5.8.3 Tolerance factor for NPSHR

No positive tolerances are allowed for the guaranteed NPSHR value.

14.6.6 Analysis

14.6.6.1 Translation of the test results to the guarantee conditions

The quantities required to verify the characteristics guaranteed by the manufacturer/supplier are typically measured under conditions more or less different from those on which the guarantee is based.

To determine whether or not the guarantee would have been fulfilled if the tests had been conducted under the guarantee conditions, it is necessary to translate the quantities measured under different conditions to those guarantee conditions.

14.6.6.1.1 Translation of the test results into data based on the specified speed of rotation (or frequency) and density

All test data obtained at the speed of rotation n in deviation from the specified speed of rotation n_{sp} shall be translated to the basis of the specified speed of rotation n_{sp}.

If the deviation from the test speed of rotation n to the specified speed of rotation n_{sp} does not exceed the permissible variations stated in Section 14.6.5.7.2, the measured data on the rate of flow Q, the pump total head H, the power input P, and the pump efficiency η can be converted by means of the equations:

$$Q_T = Q \frac{n_{sp}}{n}$$

$$H_T = H \left(\frac{n_{sp}}{n}\right)^2$$

$$P_T = P \left(\frac{n_{sp}}{n}\right)^3 \cdot \frac{\rho_{sp}}{\rho}$$

$$\eta_T = \eta$$

and the results obtained for the NPSHR can be converted by means of the equation

$$(NPSHR)_T = (NPSHR) \left(\frac{n_{sp}}{n}\right)^x$$

As a first approximation for the NPSH, the value $x = 2$ may be used if the specified conditions given in Section 14.6.5.7.2 for the speed of rotation and the rate of flow have been fulfilled and if the physical state of the liquid at the impeller inlet is such that no gas separation can affect the operation of the pump. If the pump operates near its cavitation limits, or if the deviation of the test speed from the specified speed exceeds the specifications given in Section 14.6.5.7.2, the phenomena may be influenced by, for example, thermodynamic effects, the variation of the surface tension, or the differences in dissolved or occluded air content. Values of exponent x between 1.3 and 2 have been observed and an agreement between the parties is mandatory to establish the conversion formula to be used.

Table 14.6.5.8.2.1 — Methods of determining NPSH3

Type of installation	Open sump	Open sump	Open sump	Open sump	Closed loop	Closed loop	Closed sump or loop
Independent variable	Inlet throttle valve	Outlet throttle valve	Water level	Inlet throttle valve	Pressure in the tank	Temperature (vapor pressure)	Temperature (vapor pressure)
Constant	Outlet throttle valve	Inlet throttle valve	Inlet and outlet throttle valves	Water level	Rate of flow	Rate of flow	Inlet and outlet throttle valves
Quantities, the variation of which are dependent on control	Total head, rate of flow, NPSHA, water level	Total head, rate of flow, NPSHA, water level	Total head, rate of flow, NPSHA	NPSHA, total head, outlet throttle valve (for constant rate of flow)	Total head, NPSHA, outlet throttle valve (for constant rate of flow, when total head begins to drop)	NPSHA, head, outlet throttle valve (for constant rate of flow, when head begins to drop)	NPSHA; total head and rate of flow when a certain level of cavitation is reached
Head characteristic curve versus rate of flow and NPSH							
NPSH characteristic curve versus rate of flow							

In the case of combined motor pump units or when the guarantees are with respect to an agreed frequency and voltage instead of an agreed speed of rotation, the rate of flow, pump total head, power input, and efficiency data are subject to the above mentioned translation rules, provided that n_{sp} is replaced by the frequency f_{sp} and n by the frequency f. Such translation, however, shall be restricted to cases where the selected frequency during the test varies by no more than 1%. If the voltage used in the test is no more than 5% above or below the voltage on which the guaranteed characteristics are based, the other operational data require no change.

If the above mentioned deviations, i.e., ± 1% for frequency and ± 5% for voltage, are exceeded, it will be necessary for the purchaser and the manufacturer/supplier to arrive at an agreement.

14.6.6.1.2 Test made with NPSHA different from that guaranteed

Pump performance at a higher NPSHA than guaranteed cannot be accepted, after correction for speed of rotation within the permitted ranges in Section 14.6.5.7.2, to indicate performance at lower NPSHA.

On the other hand, pump performance at a lower NPSHA than guaranteed can be accepted, after correction for speed of rotation within the permitted ranges given in Section 14.6.5.7.2, to indicate performance at a higher NPSHA provided that the absence of cavitation has been checked in accordance with Sections 14.6.5.8.2.1, 14.6.5.8.2.2, or 14.6.5.8.2.3.

14.6.6.1.3 Performance curve

Curves of best fit to the measured points will represent the performance of the pump. Separate curves shall be made for head versus flow rate, power versus flow rate, and efficiency versus flow rate. These curves shall be deemed to determine the tested pump's performance and shall be used to evaluate the test results per Section 14.6.5.4 of this standard.

14.6.6.2 Obtaining specified characteristics

14.6.6.2.1 Reduction of impeller diameter

A reduction of the impeller diameter is usually carried out when it appears from the tests that the characteristics of the pump are higher than the specified characteristics.

If the difference between the specified values and the measured values is small, it is possible to avoid a new series of tests by applying proportionality rules that allow the evaluation of the new characteristics.

The application of this method and the practical conditions for reducing the impeller diameter shall be the subject of a mutual agreement.

14.6.6.2.2 Requirement for retesting after reducing impeller diameters

If it is necessary to dismantle a pump after the performance test for the sole purpose of trimming the impeller to meet the acceptance level, and if the type number K is ≤ 1.5 (specific speed in US customary units, $N_s \leq 4100$), then no retest will be required unless the reduction in diameter exceeds 5% of the tested diameter.

Appendix A

Test arrangements (normative)

A.1   General

The best flow measuring conditions are obtained when, in the measuring sections, the flow has

- An axially symmetrical velocity distribution

- A uniform static pressure distribution

- Freedom from swirl induced by the installation

It is possible to prevent a nonuniform velocity distribution or swirl by avoiding any bend or a combination of bends, any expansion, or any discontinuity in the transverse profile in the vicinity (less than four pipe diameters) of the measuring section.

Typically, the effect of the inlet flow conditions increases with the type number K of the pump (Table 14.6.2.1, item 3.1.37). When K is greater than 1.2 (specific speed in US customary units, $N_s < 3300$), it is recommended to simulate the site conditions.

> NOTE: (Specially valid for grade 1 tests.) It is recommended that for standard test arrangements leading from reservoirs with a free surface or from large stilling vessels in a closed circuit, the inlet straight length L shall be determined by the expression $L/D = K + 5$, where D is the pipe diameter.

This expression is also valid for an arrangement that includes, at a distance L upstream, a simple right-angle bend that is not fitted with guide vanes. Under these conditions, flow straighteners are not necessary in the pipe between the bend and the pump. However, in a closed circuit where there is neither a reservoir nor a stilling vessel immediately upstream of the pump, it is necessary to ensure that the flow into the pump is free from swirl induced by the installation and has a normal symmetrical velocity distribution.

Significant swirl can be avoided by

- Careful design of the test circuit upstream of the measuring section

- Judicious use of a flow straightener

- Suitable arrangement of the pressure tappings to minimize their influence on the measurement

A throttle valve is allowed in the suction pipe for NPSH testing only when other methods of NPSH testing cannot be used.

A.2   Measurement principles

The pump total head is calculated in accordance with its definition given in row 3.1.31 of Table 14.6.2.1. Expressed as a height of pumped liquid column, it represents the energy transmitted by the pump per unit weight of liquid.

The various quantities specified in the definition of head in row 3.1.31 of Table 14.6.2.1 should as a rule be determined in the inlet section S_1 and the outlet section S_2 of the pump (or of the pump set and fittings that are the subject of the tests). Practically, for convenience and measurement accuracy, the measurements are generally carried

out in cross sections S_1' and S_2' some way upstream from S_1 and downstream from S_2 (Figure A.1). Thus, account shall be taken of the friction losses in the pipe, i.e., h_{J1} between S_1' and S_1 and h_{J2} between S_2 and S_2' (and eventually of the local head losses), and the pump total head is given by

$$H = h_{2'} - h_{1'} + h_{J1} + h_{J2}$$

where $h_{1'}$ and $h_{2'}$ are the total head at S_1' and S_2'.

A.3 Various measurement methods

Depending on the installation conditions of the pump and on the layout of the circuit, the pump total head may be determined either by measuring separately the inlet and outlet total heads, or by measuring the differential pressure between inlet and outlet and adding the difference in velocity head, if any (see Figure A.1).

Total heads may also be deduced either from pressure measurements in conduits or from water level measurements in basins.

A.3.1 Pump tested on a standardized installation

A.3.1.1 Inlet measuring section

When a pump is tested in a standard test arrangement, the inlet measuring section shall normally be located at a distance of two pipe diameters upstream from the pump inlet flange, when the length of the inlet pipe allows it. Should this length not be available (for instance in the case of a short bellmouth), in the absence of a prior agreement, the available straight length should be divided so as to take the best possible advantage of the local conditions upstream and downstream of the measuring section (for example, in the ratio 2 upstream to 1 downstream).

The inlet measuring section should be located in a straight pipe section of the same diameter and coaxial with the pump inlet flange so that the flow conditions are as close as possible to those recommended in A.1. If a bend is present a short distance upstream of the measuring section and if only one or two pressure tappings are in use (grade 2 and 3 tests), these should be perpendicular to the plane of the bend.

For grade 2 tests, if the ratio of the inlet velocity head to the pump total head is very low (less than 0.5%) and if the knowledge of the inlet total head itself is not very important (such is not the case for NPSH tests), it may be sufficient that the pressure tapping (see A.3.1.3) be located on the inlet flange itself and not at two diameters upstream.

The inlet total head is derived from the measured gauge head, from the height of the measuring point above the reference plane, and from the velocity head calculated as if a uniform velocity distribution prevailed in the suction pipe. For pumps without inlet piping (for example, submersible pumps), see ANSI/HI 11.6 (Section 11.6.4.3.8.6).

Errors in the measurement of pump inlet head can occur at partial flow due to preswirl. These errors can be detected and should be corrected per examples as follows:

a) If the pump draws from a free surface reservoir where the water level and the pressure acting on it are constant, the head loss between the reservoir and the inlet measuring section, in the absence of preswirl, follows a square law with rate of flow. The value of the inlet total head should follow the same law. When the effects of preswirl lead to a departure from this relationship at low rates of flow, the measured inlet total head should be corrected to take this difference into account (see Figure A.4).

b) If the pump does not draw from a reservoir with a constant level and head, another measuring section shall be selected sufficiently far upstream where the preswirl is known to be absent and it is then possible to reason about the head losses between the two sections (but not directly about the inlet total head) in the same way as above (see Figure A.3).

The assumption is that the fluid in the gauge line is the same fluid that is being measured.

$$H = h_2 - h_1$$

$$h_1 = z_{1'} + \frac{P_{m1}}{\rho g} + z_{m1'} + \frac{U_{1'}^2}{2g} + h_{J1}$$

$$h_2 = z_{2'} + \frac{P_{m2}}{\rho g} + z_{m2'} + \frac{U_{2'}^2}{2g} + h_{J2}$$

$$H = z_{2'} - z_{1'} + \frac{P_{m2'} - P_{m1'}}{\rho g} + z_{m2'} - z_{m1'} + \frac{U_{2'}^2 - U_{1'}^2}{2g} + h_{J2} + h_{J1}$$

A.3.1.2 Outlet measuring section

The outlet measuring section should be arranged in a straight pipe section coaxial with the pump outlet flange and of the same diameter. When only one or two pressure tappings are used (grade 2 and 3 tests), the pressure tappings should be perpendicular to the plane of the volute or of any bend existing in the pump casing (see Figure A.5).
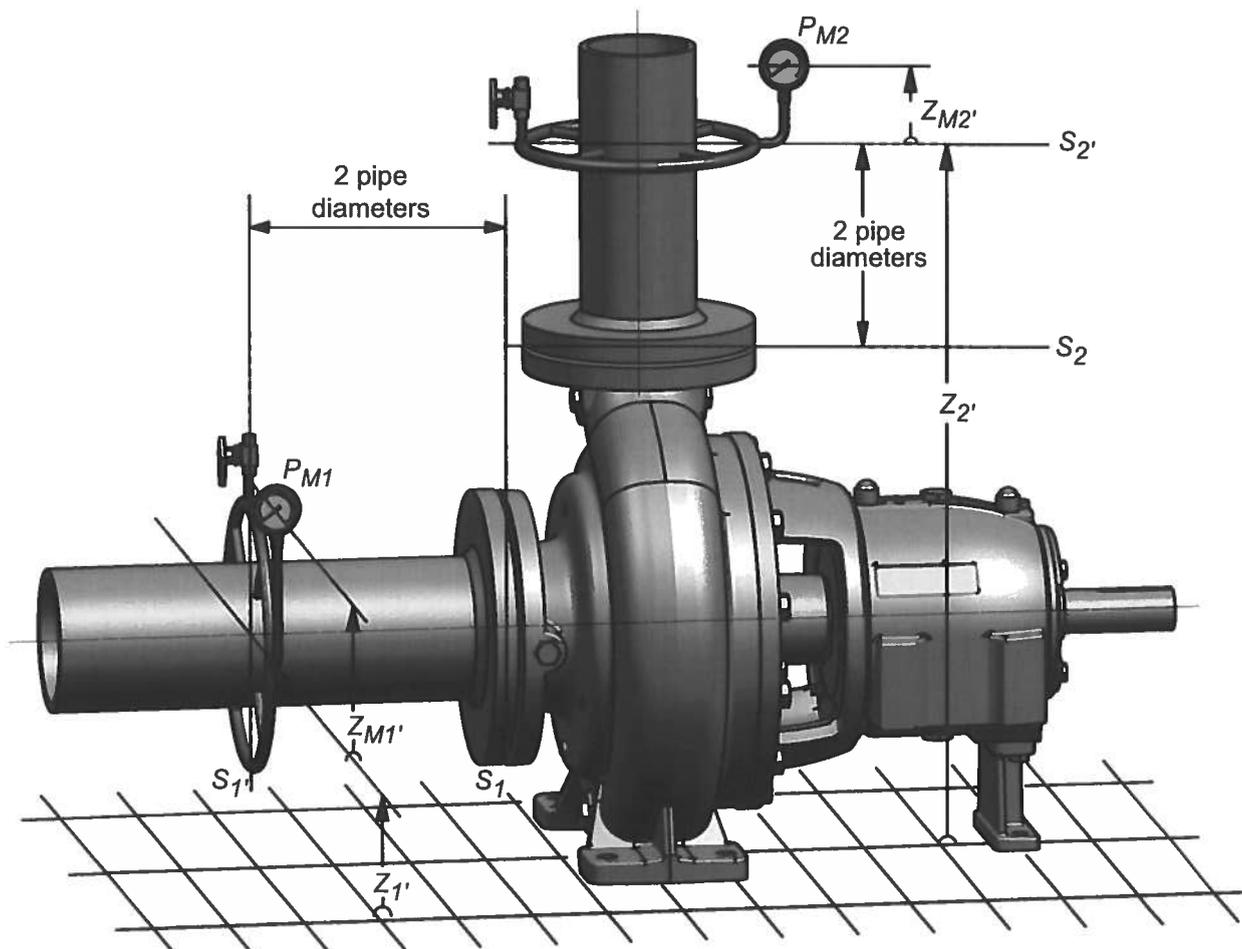
Key: 1 Line of total head (total energy)

NOTE: In this case for a horizontal shaft, $z_1 = z_D = z'_1$

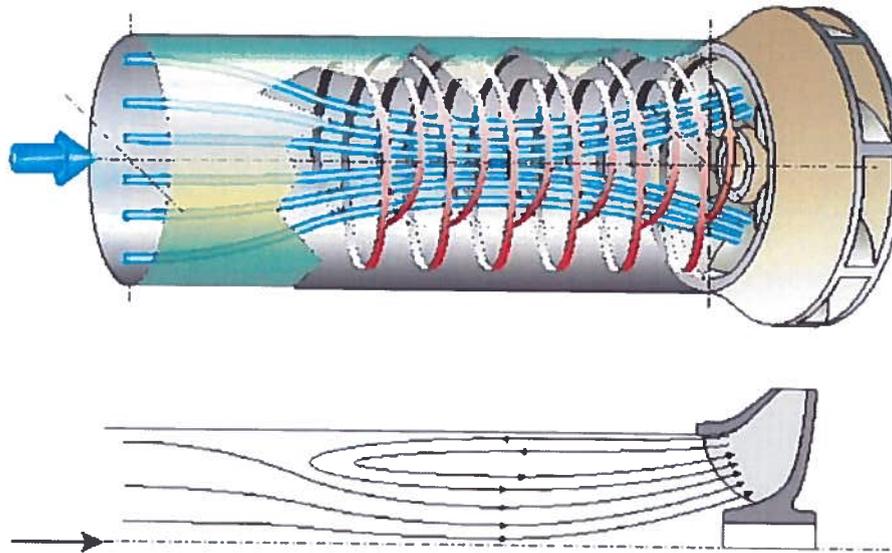**Figure A.1 — Determination of the pump total head (isometric illustration)**

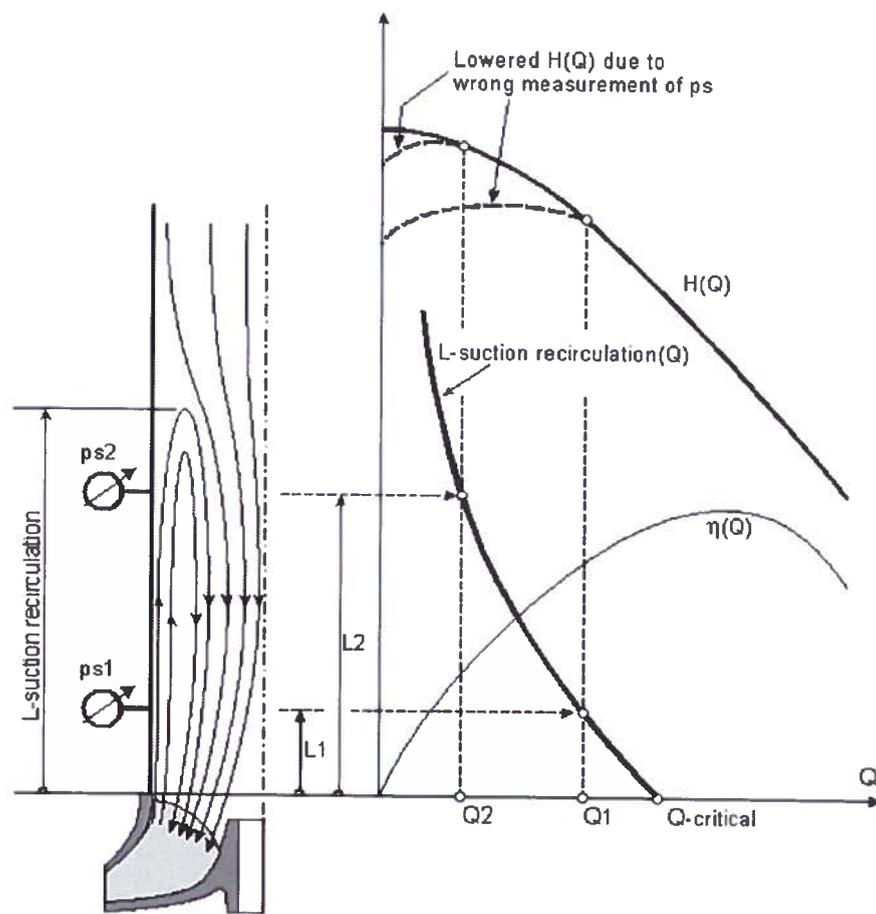**Figure A.2 — Flow at suction at part load**

Figure A.3 — Error in measurement of *H(Q)* depending on distance of suction pressure gauge from impeller

The outlet total head is derived from the measured gauge head, from the height of the measuring point above the reference plane, and from the velocity head calculated as if a uniform velocity distribution prevailed in the discharge pipe. The determination of the total head may be influenced by a swirl of the flow induced by the pump or by an irregular velocity or pressure distribution; the pressure tapping can then be located at a greater distance downstream. The head losses between the outlet flange and the measuring section shall be taken into account.

A.3.1.3 Pressure tappings

For grade 1 tests, four static pressure tappings are to be provided symmetrically disposed around the circumference of each measuring section (see Figure A.7).

For grade 2 and 3 tests, it is normally sufficient to provide one static pressure tapping at each measuring section, but when flow can be affected by a swirl or an asymmetry, two or more may be necessary (see Figure A.8).
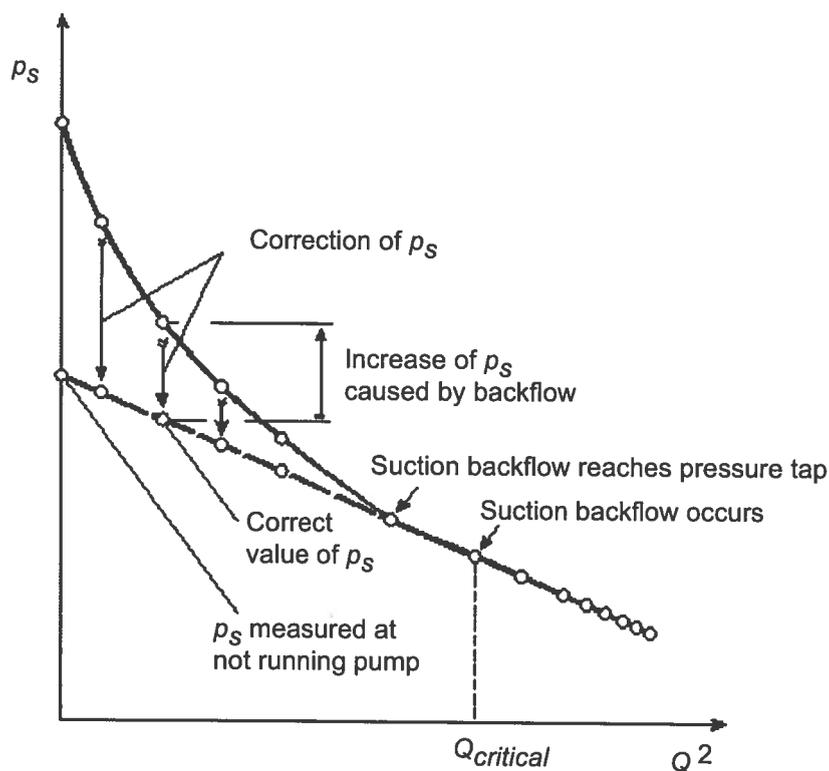

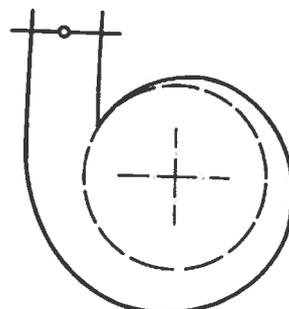**Figure A.4 — Correction of suction pressure for suction recirculation**

Figure A.5 — Pressure tapping perpendicular to the plane of the volute or to the plane of a bend, respectively

Except in the particular case where their position is determined by the arrangement of the circuit, the pressure tapping(s) should not be located at or near the highest nor the lowest point of the cross section.

Static pressure tappings shall comply with the requirements shown in Figure A.6 and shall be free from burrs and irregularities and flush with, and normal to, the inner wall of the pipe.

The diameter of the pressure tappings shall be between 3 and 6 mm (0.11 and 0.24 in) or equal to 1/10 of the pipe diameter, whichever is the smaller. The length of a pressure tapping hole shall not be less than two and a half times its diameter.

The bore of the pipe containing the tappings shall be clean, smooth, and resistant to chemical reaction with the liquid being pumped. Any coating, such as paint, applied to the bore shall be intact. If the pipe is welded longitudinally, then the tapping hole shall be displaced as far as possible from the weld.
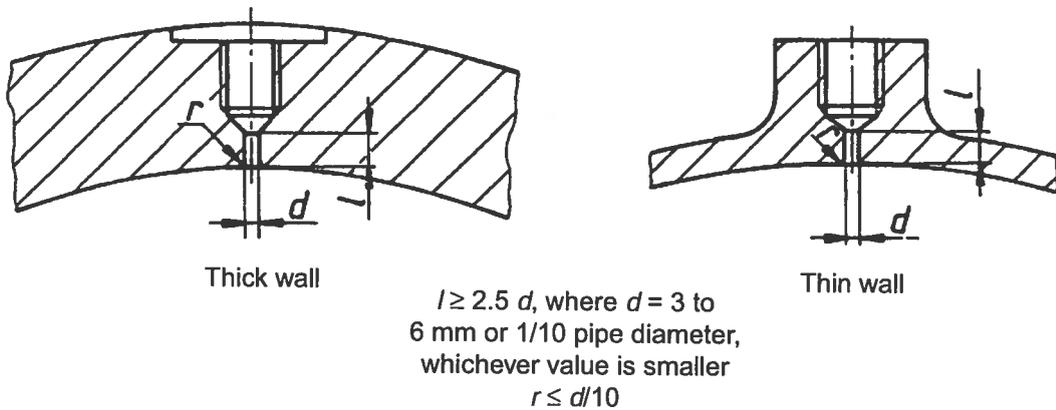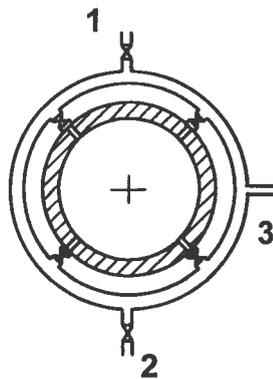
Thick wall

Thin wall

$l \geq 2.5 \, d$, where $d = 3$ to 6 mm or 1/10 pipe diameter, whichever value is smaller
$r \leq d/10$

Figure A.6 — Requirements for static pressure tappings

NOTE: 1 is a vent, 2 is a drain, and 3 is a connection to the pressure-measuring instrument.

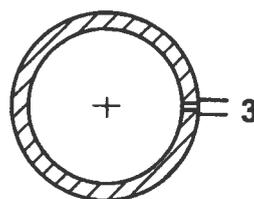**Figure A.7 — Four pressure tappings connected by a ring manifold (grade 1)**

Figure A.8 — One pressure tapping (general for grade 2 and 3)

When several pressure tappings are used, the pressure tappings shall be connected through shut-off cocks to a ring manifold of cross-sectional area not smaller than the sum of the cross-sectional areas of the tappings, so that the pressure from any tapping may be measured if required. Before making observations, the pressure with each individual tapping successively open shall be taken at the normal test condition of the pump. If one of the readings shows a difference of more than 0.5% of the total head with respect to the arithmetical mean of the four measurements, or if it shows a deviation of more than one times the velocity head in the measuring section, then the cause of this spread shall be ascertained and the measuring conditions rectified before the test proper is started.

When the same pressure tappings are used for NPSH measurement, this deviation shall not exceed 1% of the NPSH value or one times the inlet velocity head.

Pipes connecting pressure tappings to possible damping devices and to instruments shall be at least equal in bore to the bore of the pressure tappings. The system shall be free from leaks.

Any high point in the line of the connecting pipes shall be provided with a purging valve to avoid trapping of air bubbles during measurements.

Whenever possible, it is recommended that transparent tubing be used to determine if air is present in the tubing.

A.3.1.3.1   Correction for height difference

Correction of the pressure reading p_M for height difference between the middle of the measuring section and the reference plane of the pressure-measuring instrument shall be made by the following formula:

$$p = p_M + z_M$$

The assumption is that the fluid in the gauge line is the same fluid being measured.

A.4   Simulated test arrangements

When, for the reasons given above, it is agreed to test a pump under simulated site conditions, it is important that at the inlet of the simulated circuit the flow should as far as possible be free from significant swirl induced by the installation and have a symmetrical velocity distribution. All necessary provisions shall be made to ensure these conditions are achieved.

If necessary, for grade 1 tests, the velocity distribution of the flow into the simulated circuit shall be determined by careful Pitot tube traverses, in order to establish that the required flow characteristics exist. If not, then the required characteristics can be obtained by the installation of suitable means, such as a flow straightener adapted for the fault of the flow to be corrected (swirl or asymmetry). Specifications of the most widely used types of flow straighteners can be found in ISO 7194. Care shall be taken to ensure that the conditions of test will not be affected by the head losses associated with some straightening devices.

A.5   Pumps tested with fittings

If specified in the contract, standard tests can be carried out on a combination of a pump and

a)   associated fittings at the final site installation, or

b)   an exact reproduction thereof, or

c)   fittings introduced for testing purposes and taken as forming part of the pump itself.

The flow at the inlet and outlet of the whole combination shall comply with the requirements specified in A.1.

If the tests are on the combination of the pump and the whole or part of its upstream and downstream connecting fittings, these being considered an integral part of the pump, the provision of A.1 apply to the inlet and outlet flanges of the fittings instead of the inlet and outlet flanges of the pump. This procedure debits against the pump all head losses caused by the fittings.

A.6 Pumping installation under submerged conditions

Where a pump, or a combination of a pump and its fittings, is tested or installed in conditions where the standard pipe connection as described in A.3.1 cannot be made owing to inaccessibility or submergence, measurements shall be taken in accordance with the following requirements.

The inlet total head is equal to the height above the reference plane of the free surface level of the liquid from which the pump draws, plus the head equivalent to the gauge pressure prevailing above this surface.

According to the circumstances, the outlet total head can be determined either by a pressure measurement in the discharge pipe (see A.3.1.2) or, if the pump discharges into a free surface basin, by a level measurement in this basin. In this case, and provided that the liquid is really at rest near the level measuring point, the outlet head is equal to the height above the reference plane of the free surface level of the liquid in which the pump discharges plus the head equivalent to the gauge pressure prevailing above this surface.

The definition of head is sometimes misunderstood for vertically suspended pumps compared to other rotodynamic pumps, which can also lead to misunderstanding on power or efficiency (refer to Table 14.6.2.1, row 3.1.38.1 and 3.1.42.1). Manufacturers' performance curves and acceptance are based on bowl head rather than pump total head. This is due to the fact that losses within the pump are not known until the pump configuration is selected and finalized. These losses must be calculated by qualified personnel.

The bowl assembly total head (H_{ba}) is the gauge head measured at the gauge connection located on the column pipe downstream from the bowl assembly, plus the velocity head (h_{vd}) at the point of the gauge connection, plus any losses between top of bowl assembly to tap location, plus datum from gauge ($z_{M2'} + z_{2'}$), plus liquid level (z_L). Refer to Figure A.9. Friction losses of suction piping, can (barrel), and strainers must also be added if significant.

Unless otherwise agreed on, acceptance of test result of a vertically suspended pump is based on bowl performance.

Pump total head is the sum of the head as measured at the centerline of the pump discharge head flange or thereafter plus the distance from the centerline of the discharge head to the water surface. It includes the velocity head correction at the discharge gauge pressure tap.

This procedure debits against the pump all head losses arising between the measuring sections.

If necessary, the friction head losses between the measuring sections and the contractual limits of the pump may have to be determined by calculations. The local head losses due to the singularities of the circuit and to various fittings (suction filter, nonreturn valve, delivery elbow, valve, expanders, etc.) shall as far as possible be specified when drafting the contract, by the party that provides these fittings. If this appears impossible, then the purchaser and the manufacturer/supplier shall agree the loss value to be adopted before the acceptance tests.

As deep-well pumps (Figure A.9) are normally not tested with their whole vertical pipes, unless the acceptance test is carried out on-site, the friction head losses in the missing parts shall be evaluated and specified to the purchaser by the manufacturer/supplier. If it appears necessary to verify the specified characteristics by an on-site test, then this shall be specified in the contract.

For tests of pumps of this kind, the guarantees apply to the fittings only when specified in the contract.

$$h = h_2 - h_1$$

For p_{m1} pressure tap location above liquid level and equal to atmospheric pressure:

$$p_{M1} = p_{atm} = 0$$

$$h_1 = z_L$$

$$h_2 = z_{2'} + \frac{p_{M2}}{\rho g} + z_{M2'} + \frac{U_2^2}{2g} + h_{j2}$$

$$h = z_{2'} - z_L + \frac{p_{M2}}{\rho g} + z_{M2'} + \frac{U_2^2}{2g} + h_{j2}$$

For p_{m1} pressure tap location above liquid level and not equal to atmospheric pressure:

$$h_1 = z_L + \frac{p_{M1}}{\rho g}$$

$$h_2 = z_{2'} + \frac{p_{M2}}{\rho g} + z_{M2'} + \frac{U_2^2}{2g} + h_{j2}$$

$$H = z_{2'} - z_L + \frac{p_{M2} - p_{M1}}{\rho g} + z_{M2'} + \frac{U_2^2}{2g} + h_{j2}$$

For p_{M1} pressure tap location below the pump intake and U_1 is not 0, see Figure A.10.

NOTE: Borehole and deep-well pumps cannot usually be tested with their complete lengths of delivery main and, consequently, the loss of head in the portions omitted, and the power absorbed by any shafting therein, cannot be taken into account. The thrust bearing will be more lightly loaded during the test than it would be in the final installation.

A.7 Self-priming pumps

In principle the priming ability of self-priming pumps shall always be verified at the contractual static suction head with the attached inlet piping equivalent to that in the final installation. When the test cannot be carried out in the described manner, the test arrangement to be used should be specified in the contract.
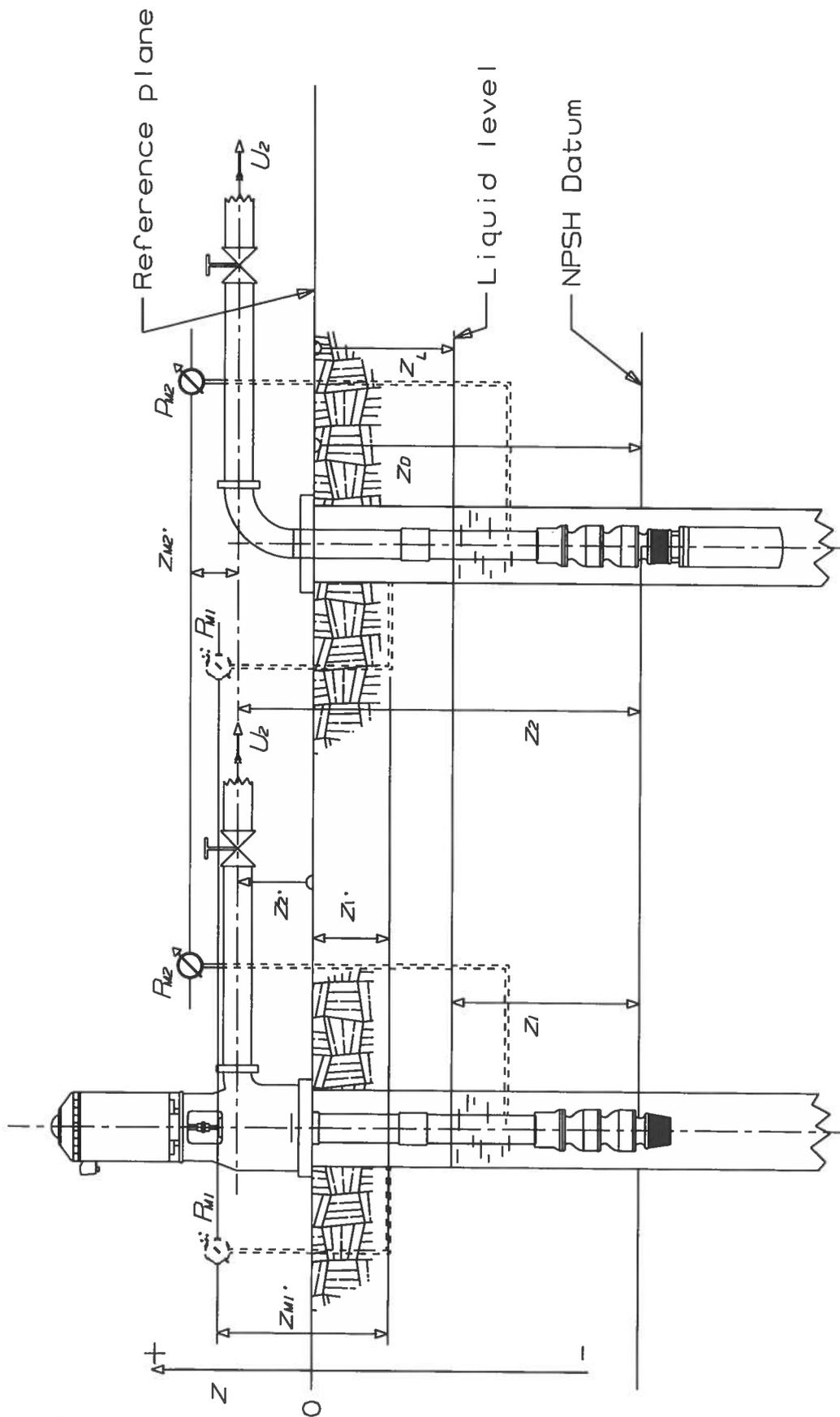
A.8 Friction losses at inlet and outlet

The guarantees under Section 14.6.3.4 refer to the pump inlet and outlet flanges, and the pressure-measuring points are in general at a distance from these flanges. It may therefore be necessary to add to the measured pump total head the head losses due to friction (h_{J1} and h_{J2}) between the measuring points and the pump flanges.

Such a correction should be applied only if

$$h_{J1} + h_{J2} \geq 0.005H \text{ for grade 2 and 3 or}$$

$$h_{J1} + h_{J2} \geq 0.002H \text{ for grade 1.}$$

Figure A.9 — Measurement example of pump total head *H* for submerged pumps

NOTE: P_{b} is typically atmospheric pressure or "O"

If the pipe between the measuring points and the flanges is unobstructed, straight, of constant circular cross section, and of the length L, then:

$$h_J = \lambda \frac{L}{D} \frac{U^2}{2g}$$

The value of λ may be determined from a Moody chart or derived from

$$\frac{1}{\sqrt{\lambda}} = -2\log_{10}\left[\frac{2.51}{Re\sqrt{\lambda}} + \frac{k}{3.7D}\right]$$

where relative roughness is

$$\frac{k}{D} = \frac{\text{pipe diameter uniform roughness}}{\text{pipe diameter}} \text{ (pure number)}$$

If the pipe is other than unobstructed, straight, and of constant circular cross section, then the correction to be applied shall be the subject of special agreement in the contract.
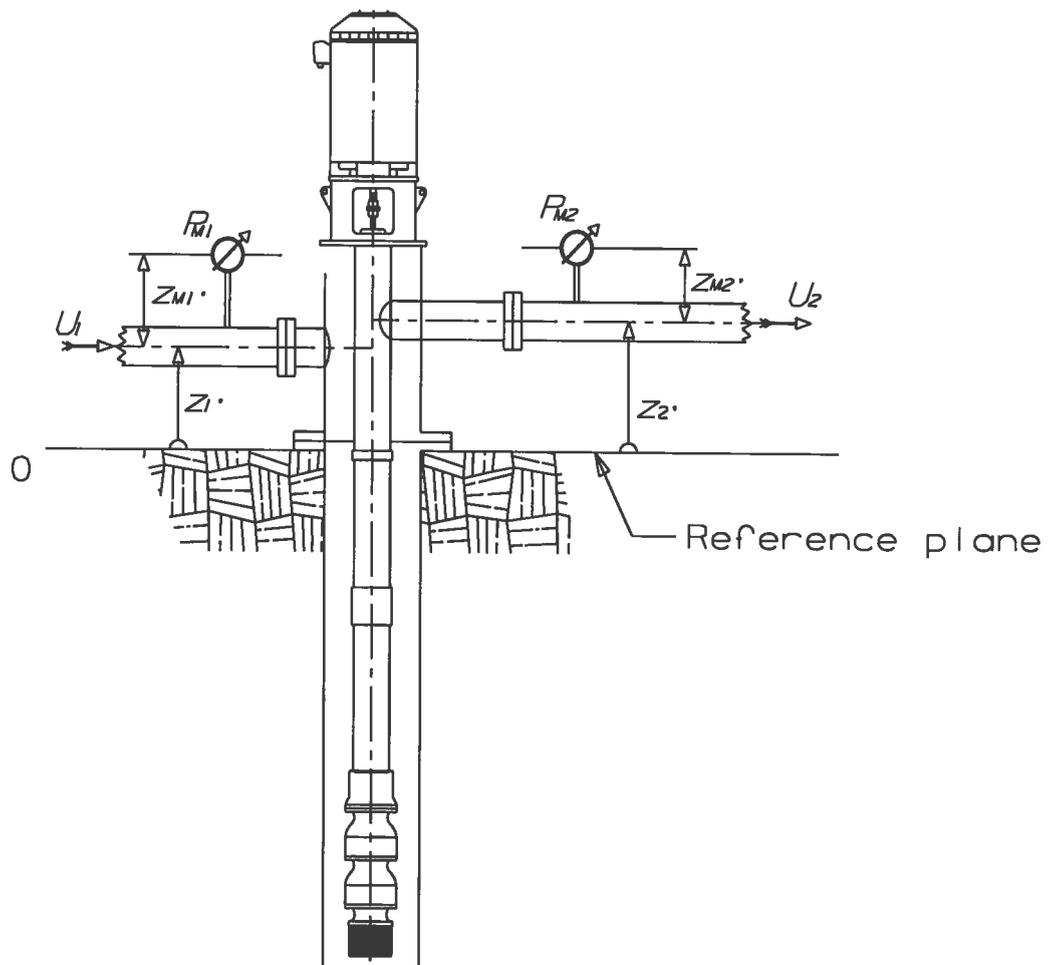
Figure A.10 — Measurement example of pump total head H for submerged pump with closed suction

Appendix B

Hydrostatic pressure testing (normative)

B.1 Scope

This appendix covers a factory hydrostatic pressure test procedure that, when specified, applies to pressure-containing parts of all types of rotodynamic pumps within the scope of this standard, including any auxiliary equipment making up a pump unit.

Requirements are included for applying a hydrostatic pressure test to separate zones within a pump that are subject to different pressures.

B.2 Definitions

For purposes of this document, the following definitions apply:

- **Pressure-containing parts** means any part or assembly of parts that is normally subjected to a pressure differential

- **Rated pressure** is the maximum pressure that would occur in that part when the pump is operated at rated conditions for the given application of the pump

- **Containment of liquid** means only prevention of its escape through the external surfaces of the pumps or pump components, normally to atmosphere

- **Item(s) to be tested** means any part, component, subassembly, pump, or pump unit that is to be the subject of hydrostatic pressure test

    NOTE: A mechanical seal assembled into a pump or into a separate subassembly, together with an end plate to connect the stationary elements of the seal to the stationary parts of the pump, will not be considered an item to be tested but may be subjected to the test pressure.

B.3 General

All pressure-containing items shall be hydrostatically pressure tested. The purpose of this test is to demonstrate the absence of leakage through pressure-containing walls of any item under test and from the joints formed by an assembly of items under test, by imposing a defined pressure in excess of the rated pressure for which the item is supplied.

Pumps with double volutes, multiple stages, and otherwise with internal separating walls and to be tested in segments shall have a hydrostatic pressure applied to each segment related to the operating conditions in the segment when the pump is working at its rated discharge pressure.

Pressure-containing chambers that function independently shall be tested separately without pressure being applied to any adjacent chamber.

B.4 Timing of the test

The manufacturer shall allow sufficient time during manufacture for the test and examination of all items within the scope of this standard. The test may be carried out on an individual item or a subassembly as a group of items. Normally, the hydrostatic pressure test is carried out:

- after completion of machining

- following nondestructive testing, or after special leak tests below a gauge pressure of 50 kPa (7.25 psi)

B.5 Preparation for testing

Items to be subjected to test shall be free from grease, oil, and other contaminants, and any cleaner used shall be compatible with the materials of manufacture of the pump, its auxiliaries, and its intended use.

The items to be tested are to be assembled for the test and all openings are to be sealed by appropriate means, which may include blind flanges, plugs, closures, and tension rings.

The item to be tested and the gauge lines are to be vented and filled completely with the test liquid. Provisions shall be made to vent all the air at the high points on the item.

Wherever feasible the mating parts and fasteners used in any assembly shall be those to be used in the delivered pump, otherwise reasons for deviating from this requirement shall be recorded.

B.6 Test liquid

The hydrostatic pressure test is normally to be carried out using clean water at ambient temperature, with the addition of corrosion preventatives, wetting agents, and organic growth inhibitors where necessary. If the properties of the material impose a limit on the test temperature, which affects the test procedure, then this is to be noted on the test record.

An alternative liquid, such as oil having maximum viscosity of 32 cSt (150 SSU), contractually agreed on, shall be used where clean water is not appropriate for the application. The liquid used and the reason for its selection shall be recorded.

B.7 Test pressure

The hydrostatic test pressure relates to the rated pressure at ambient temperature of the items to be tested.

The hydrostatic test pressure shall be calculated from the formula:

$$p_{test} = K_1 \times K_2 \times p_{rated}$$

Where:

p_{test} =   the hydrostatic test pressure

p_{rated} =   the rated pressure at operating temperature

K_1 =   a factor whose value shall be determined by the pump specification, but shall not be less than 1.3, except for thermoset parts

K_2 =   a factor whose value shall be determined by the material strength, as follows

= $\dfrac{\text{allowable stress at ambient temperature}}{\text{allowable stress at operating temperature}}$

Due to the irreversible damage that can occur to the reinforcement of thermoset parts that are put under excessive pressure, hydrostatic test pressure shall be 1.1 times the rated pressure for those parts. The manufacturer shall verify through test records that adequate sampling was done to prove that the parts can sustain full hydrostatic test pressure calculated using the above formula. When a full hydrostatic test pressure on thermoset parts is requested, all parties should agree to the consequences of possible irreversible damage.

B.8 Test procedure

At the start of the hydrostatic pressure test, the external walls of the item to be tested shall be dry.

The arrangements used to seal the chambers to be pressurized shall be inspected for proper bolting and enclosure before applying the test pressure. The hydrostatic pressure shall be steadily increased in a controlled manner until the test pressure is achieved. The test pressure shall be held essentially constant for the duration of the test period. Throughout this procedure continuous observation shall be maintained to detect any leakage.

The test pressure shall be maintained for a sufficient time to allow complete inspection of the item being tested. Table B.1 gives the minimum test times based on the tested pump's pressure rating. For pumps with a shaft power of less than 10 kW (13 hp) and a pressure rating of less than 1000 kPa (145 psi), the minimum test time shall be 1 minute.

Larger pumps, parts, and assemblies will by necessity require longer test times; this shall be determined by the pump specification or as otherwise set out in Table B.1 below.

Table B.1 — Requirements for longer test periods

Items to be tested Pumps and pressure-containing parts	Test duration, in minutes
Rated above 2500 kPa (360 psi)	30
Rated above 1000 kPa (145 psi) and not above 2500 kPa (360 psi)	10
Rated up to 1000 kPa (145 psi)	5

If the item to be tested is a completely assembled pump or pump unit, checks shall be made to ensure that excessive stresses are not created on the shaft seals and the results of the checks recorded.

If necessary, the tested item shall be completely drained of test liquid and thoroughly washed and dried.

B.9 Acceptance criteria

The integrity of the item under test is to be regarded as satisfactory if during the test period there are no visible signs of leakage.

Leakage through shaft packing, temporary gaskets, or internal test partitions required for segmental hydrostatic pressure testing is acceptable.

B.10 Repairs

If the acceptance criteria stated in Appendix B.9 are not met, the cause shall be identified and the manufacturer shall either safely dispose of the item tested or apply suitable corrections. After correction, the hydrostatic pressure test shall be repeated. The methods of correction and any changes to the design shall be recorded.

B.11 Test records

The manufacturer shall keep a record of the test and its results for a minimum of five years. The record shall contain:

a) Identification of the item tested.

b) Details of the test, including the liquid used, the test pressure, and the test result.

c) Any corrective actions applied, including changes to the design.

d) Other information required by this document to be recorded.

B.12 Test certificate

If required by the purchaser, a test certificate may be made available. The test certificate shall certify that the hydrostatic pressure test was carried out in accordance with this document, and that the test item met the acceptance criteria.

B.13 Test report

If required by the purchaser, the manufacturer/supplier may supply a report of the hydrostatic pressure test, which shall indicate at least the following information:

- Identification of the pump

- Type and characterizing data (e.g., dimensions) of the item tested

- List of any modifications made to the design

- Applicable test standard

- Type of test liquid

- Test pressure

- Test duration

- Result of the test

- Date when the test was carried out

- Signature of the inspector or test controller

Each item subject to a test report shall be marked uniquely for identification.

In the case of a witnessed test, the acceptance of an item or its release for further manufacturing steps shall be confirmed by the signature of the purchaser's representative.

Appendix C

Purpose of test tolerances (informative)

C.1 Explanation of test tolerances - variations

Variations in hydraulic performance, as measured in a test, may occur for several reasons.

Common reasons for performance variations include manufacturing tolerances, instrument fluctuations, instrument accuracy, and inherent fluctuations in the pumped media in the vicinity of the pump suction and discharge. The magnitude of variation will vary directly with the degree of precision applied both to manufacturing processes and to test equipment and procedures.

Variation in test results, as indicated above, also includes an uncertainty related to instrument measurements. In this standard, the reported results are applied directly to the acceptance criteria. Uncertainty of the measurement is reported separately as a characteristic of the instrumentation and is included in the test tolerance, which is explained in Section 14.6.3.3 of the standard. The application of a specific test tolerance to any guarantee point therefore infers a level of precision that must be achieved to provide repeatable results.

Various manufacturing processes often dictate the level of precision that is possible to obtain in the manufacture of a certain part. The selection of the appropriate tolerance bandwidth therefore must take into account the overall requirements in terms of product performance and repeatability against cost and manufacturing lead time.

C.2 Manufacturing variations

A large percentage of pump components are produced as cast parts that must be subsequently machined, cleaned, dressed, and assembled into stationary and rotating assemblies prior to pump test. Normal manufacturing variations occur that have an impact on the continuity of hydraulic geometry and therefore impact performance.

C.2.1 Casting dimensions

Casting dimensional variations depend on the molding process. Low-volume, large components are frequently molded in sand using manual processes. As with most manual processes, there is an inherent large variation in the process, both dimensionally and in the resulting surface finish. Machine molding is used on higher production volume components and results in less variation in dimensions and finishes. For a higher level of precision, some components may be cast using an investment process that results in the smallest variation in both dimensions and finishes.

C.2.2 Casting surface finish

The variation of hydraulic surface finish of the casing or impeller is different for the various types of cores used to make the part. Wax cores exhibit very little variation in finish and typically do not cause hydraulic performance variations. Sand cores, particularly those that are hand-molded, can have significant variation in surface finish. Variation in molten metal temperature can also result in variations in surface finish in a single foundry run. The surface finish in these types of cores can affect head and efficiency.

Minor variations in the contour of the leading edge of impeller vanes can affect NPSHR significantly. Because the leading edges are often thin, these can be easily deformed.

Casting shrink rates are affected by chemical composition variations within alloy specifications, pouring temperature, and cooling rates. Resulting dimensional variations affect head and efficiency.

C.2.3   Casting cleaning

Casting cleaning is the process in which a raw casting is manually or automatically cleaned and dressed. Casting burrs, surface imperfections, mold split lines, etc. are often ground off and smoothened out. Hydraulic passageways are often dressed and matched by hand. These processes result in variations of the local geometry of the part. An increased level of precision can be obtained through the introduction of more sophisticated metal removal processes and metrology.

C.2.4   Machining dimensions

Assemblies are made with close clearances between adjacent parts. Variations in machined component dimensions can have an impact on controlling leakage between different pressure regions within the pump assembly. The resulting variation in leakage can change the pump performance.

C.2.5   Machining finishes

Normal machining finish variations do not affect total developed head significantly, but can affect efficiency and power. For example, variations of surface finish on a packed sleeve can affect the friction between packing and sleeve. Impeller shroud surface finish quality has an impact on disk friction. (Smoother shroud surfaces create less friction). The effect of increasing shroud surface roughness is a reduction in head and efficiency (more noticeably on pumps with lower type numbers [specific speeds]).

Although the addition of the effects of each variation can result in large performance variations, usually many variations counteract each other, resulting in a lesser overall effect. Manufacturing tolerances follow the normal distribution law, thus a manufacturer can calculate the likelihood of meeting a certain tolerance requirement. Experience has shown that these typical manufacturing process variations result in performance variations in the range of ± 2 to 5%. A higher level of precision may reduce this variation but will demand the use of selective manufacturing processes, additional controls, and consequent impact on cost and lead time.

C.3   Effect of accessories on mechanical losses (power)

It must be noted that the mechanical losses occur within bearing units and shaft sealing (dynamic, mechanical, packing). These are friction losses and have a direct effect on the measured power absorbed during test. Magnitude varies with bearing and seal design, load, setting, and speed of rotation. Any variation of these factors must be accounted for between tests.

C.4   Selection of pump test acceptance grades and corresponding tolerance bands

There are a wide range of parameters to be considered during manufacture and test, any variation of which may have influential impact on the accuracy and repeatability of test-stand measurements. The magnitude of these effects on pump performance often depends on pump design and type numbers (specific speed). The selection of pump test acceptance grades and corresponding tolerance bands should therefore reflect a measure of the level of engineering analysis and manufacturing effort required for a specific application in order to fulfil the overall needs of the pump owner. Noting that, if no acceptance grade is specified, then the default values per Section 14.6.4 and Table 14.6.4 of the standard will be applied.

Appendix D

Recommended tests (informative)

D.1 General

ANSI/HI 14.6 mandates an acceptance test based on pump flow and head delivered. It also details several optional tests that a pump purchaser may want to have performed. This appendix gives a baseline recommendation for optional tests and it attempts to shed light on the consequences of specifying tighter test tolerances as well as optional tests.

It is recommended that the costs of acceptance tests and special tests be clearly stated in the contract. It should be understood that specifying tighter acceptance tolerances generally leads to higher testing costs and longer lead times. When NPSH testing is specified, test costs will increase because the tested pump will have to undergo a different and more time-consuming test program, often performed at a different test rig, requiring additional set-up and tear-down time.

D.1.1 Bandwidth of manufacturing tolerances

To reduce the tolerance bandwidth, tighter manufacturing tolerances are required, which add cost and may increase delivery time.

Sand-cast molds are the least expensive to make, but typically have the widest tolerance. Investment casting will have the best finish and the most repeatable dimensions. The molding equipment cost for an investment casting may be two to four times that of a sand mold. It takes a high production volume to justify the cost of this casting equipment. To have a smaller bandwidth when sand casting may require many hours in hand labor to obtain the same exact results from one pump to another.

Machining parts to closer tolerances can increase the cost of the labor by 50% as well as the time to make it to the required tolerances.

Reducing the tolerances increases costs because of the additional care required during production and a likely scrap rate increase. Both effects will increase delivery time.

Sometimes the impeller can be hand worked to obtain as-quoted results; again this increases the labor and delivery time.

Sometimes the impeller can be axially adjusted to obtain better impeller-casing alignment. This will obtain more head, but may increase assembly cost and delivery time.

If it is desired to reduce the head of the pump, even though it is within the tolerance band, the impeller diameter can be reduced. This also adds to the delivery time.

Machining the outside of the impeller shrouds and polishing the hydraulic passage of the casings or diffusers can improve performance by reducing internal fluid friction losses, but adds significantly to the cost and delivery.

D.1.2 Bandwidth of measuring tolerances

Reduction of the tolerance bandwidth for measuring tolerances will add cost and may increase delivery time. This is a result of using more sophisticated test equipment and extension of the actual test period to get a higher stability of the operation point.

D.2 Recommended test specification matrix

The recommended test specification matrix shown below is intended to be used as a guideline for users as to which tests should be specified based on pump power and intended service. For normally manufactured pumps, users may consider a certificate of compliance in place of actual testing.

Table D.1 — Matrix of recommended tests

	Recommended Test Specification Matrix					
	Service	Test Type	Performance	Hydrostatic	NPSH	Mechanical
Standard Pumps	Up to 50 kW (67 hp)		NO	NO	NO	NO
	Greater than 50 kW (67 hp)		YES	NO	NO	NO
Additional Considerations	New/Unique Design		YES	YES	NO	YES
	Critical Service or Application		YES	YES	NO	YES
	NPSH Critical		YES	NO	YES	NO

Definitions:

Standard pumps - Pumps that have previously been manufactured and have a history of consistently demonstrated compliance with HI test standards.

New/unique design - Pumps of a new size, or with new design features.

Critical service or application - As defined by customer.

Appendix E

Mechanical test (informative)

E.1 Mechanical test objective

A mechanical test demonstrates the satisfactory mechanical operation of a pump at the rated conditions, including vibration levels; lack of leakage from shaft seals (except for packed stuffing box), gaskets, and lubricated areas; and free-running operation of rotating parts. When specified, bearing temperature stabilization will be recorded.

These tests do not apply to submersible close-coupled pumps, both diffuser and volute style, as described in ANSI/ HI 1.1-1.2 *Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions.*

References to shaft seal do not apply to sealless pumps.

E.2 Mechanical test setup

The test setup should conform to the requirements of Appendix A of the standard where applicable, and the test liquid should be clear water. In addition, instrumentation should be added to measure the following:

a) Vibration at the pump bearing housing in two directions perpendicular to the shaft and in the axial direction.

b) Temperature of both bearings or bearing housings.

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

d) Oil temperature, when oil sump is used.

E.3 Mechanical test operating conditions

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

a) Shaft speed – as required to meet rated conditions as specified in the customer order. Testing at other rotational speeds is allowed if agreed to by manufacturer and purchaser.

b) Rate of flow – the rated flow for which the pump is specified. Testing at other flow rates is allowed if agreed to by manufacturer and purchaser.

c) Suction pressure – as available from the test facility sufficient to prevent cavitation.

d) Liquid temperature – at ambient condition.

e) Air temperature – ambient.

E.4 Mechanical test instrumentation

E.4.1 Vibration

Vibration instruments can be either handheld or rigidly attached to the pump. For pumps with speeds above 600 rpm, the measurement instrumentation should be capable of measuring the RMS vibration velocity. For pumps with speeds of 600 rpm and below, the measurement instrumentation should be capable of measuring RMS velocity and RMS peak-to-peak displacement. Refer to ANSI/HI 9.6.4 *Rotodynamic Pumps for Vibration Measurements and Allowable Values* for vibration test requirements, including instrumentation.

E.4.2 Temperature

Temperature instruments can be any recognized temperature sensor, such as pyrometers, thermometers, thermocouples, and the like. They should be capable of measuring the metal temperature on the outside of the housing of both bearings, and may be handheld or rigidly attached to the bearing housing. The top center over the bearing is usually the location of the highest temperature. Where temperature sensors are built into the pump, they should be used instead of sensors on the bearing housing. If built-in, they must be at a location where temperature is of interest.

E.5 Mechanical test procedure

The pump speed and rate of flow should be set per specification. The pump should be operated for a minimum of 10 minutes, and the following observations should be made and recorded:

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

b) Vibration level in accordance with ANSI/HI 9.6.4 *Rotodynamic Pumps for Vibration Measurements and Allowable Values*.

c) Bearing temperatures at both inboard and outboard bearings. When specified, the pump should be operated until the bearing temperature stabilizes. See ANSI/HI 1.4 *Rotodynamic (Centrifugal) Pumps for Manuals Describing Installation, Operation, and Maintenance* for the temperature stabilization procedure.

d) Rubbing of rotating parts may be detected by indications of structureborne noise (in some instances, this may be audible) and input power fluctuations. Observe 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.

E.6 Mechanical test acceptance criteria

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

a) Vibration levels do not exceed the specified allowable limits.

b) Temperature of both bearings' housing surfaces do not exceed the pump manufacturer's standard for the product as established prior to test.

c) Mechanical seals may have an initial small leakage, but should have no visible leakage when running at test operating conditions for a minimum of 10 minutes. There are seal designs that may exhibit a prescribed level of leakage during test and this can be confirmed with the seal manufacturer. When shut down, there should be no visible leakage from seals for 5 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 typically should have no more than 12 drops per minute leakage for a 25-mm (1-in) shaft up to 3500 rpm. For larger shafts or higher test speeds and pressures, allowable leakage will be increased proportionately with shaft diameter speed and pressure or as agreed to by the purchaser.

   There should be no visible leakage through pressure-containment parts, gaskets, seal recirculation piping, bearing housing, etc.

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

E.7 Mechanical test records

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

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

b) Vibration levels on both bearings in two directions perpendicular to the shaft. In addition, a single vibration reading that is paralled to the rotational axis of the shaft.

c) Bearing temperatures.

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.

Appendix F

NPSH test arrangements (informative)

F.1 General

The test described in Section 14.6.5.8 of the standard can be conducted by any of the methods indicated in Table 14.6.5.8.2.1 and in any of the installations described in the following clauses.

It is possible to vary two control parameters and thus keep the rate of flow constant during a test, but this is usually more difficult.

F.2 Characteristics of the circuit

The circuit should be such that when cavitation appears in the pump, it should not occur elsewhere to such an extent that it affects the stability or the satisfactory operation of the installation or the measurement of the pump performance.

It should be ensured that cavitation and the bubbles and degassing produced by cavitation in the pump do not affect the functioning of instrumentation, particularly the flow measuring device.

The measuring conditions on the cavitation test rig should conform to the conditions specified in Appendix A of this standard.

Special regulating valves at inlet and outlet may be needed to avoid cavitation that can influence the test results.

Cavitation in the flow through a throttle valve can sometimes be prevented by using two or more throttle devices connected in series or by arranging for the throttle valve to discharge directly into a closed vessel or a large-diameter tank interposed between the throttle and the pump inlet. Baffles and means of extracting air from such a vessel may be needed, especially when the NPSH is low.

When a throttle valve is partially closed it is necessary to make sure that the pipe is full of liquid, and pressure and velocity distributions at the inlet measuring section are uniform. This may be achieved by use of a suitable flow-straightening device and/or long straight pipe of at least 12 D lengths at the pump inlet.

F.3 Characteristics of the test liquid

The liquid should be clean and clear and should not contain solid matter. Free gas should be removed to the extent possible before test.

Deaeration of water used for a cavitation test is necessary only if the pump is to be used in practice with deaerated water.

Conversely, to avoid degassing in any part of the pump, the water of the circuit should not be supersaturated.

The general flow conditions stipulated in Appendix A, especially at the inlet of the pump, should be fulfilled.

F.4 Allowable air content during NPSH testing

NOTE: All testing is assumed to be done in cold water.

F.4.1 Testing in open loops

Assumptions:

*   The atmospheric pressure above the water level is 100 kPa (14.5 psi)

*   The water is saturated with air (worst case, assumption is on the safe side)

The standard could limit the air content to 1% by volume of total flow at the pump suction nozzle. An air content of 1% by volume is assumed to have no significant impact on performance and NPSH. Under these assumptions, the pump can be tested down to about NPSHA = 7 m (23 ft), as demonstrated by Figure F.1. Testing with much lower NPSHA would require degassing in a closed loop for accurate measurements. However, when testing with NPSHA lower than 7 m (23 ft), the testing is on the safe side, i.e., if the pump meets the guaranteed NPSH3 with air content higher than 1%, operation is safe with respect to suction capability. The pump vendor may thus choose – without detriment to the client – to test for simplicity and cost reasons with NPSHA lower than 7 m (23 ft), as long as the guarantees are fully met.

F.4.2 Testing in closed loops with suction pressure below atmospheric

In contrast to open loops, air cannot escape from the circuit when testing in a closed loop. Testing with water that has not been properly degassed may negatively impact the test result. Reducing the content of dissolved air to one forth of saturation (or less) ensures that free air bubbles are largely absent from the water. The water must be partially degassed by applying a vacuum in the tank while circulating water with sufficiently high velocity through the loop in order to sweep trapped air and allow it to be removed. As an example, saturated water at 20 °C (68 °F) dissolves about 9.5 ppm of oxygen. Depending on the level of the vacuum achieved in the tank, an oxygen content of less than 2.5 ppm should be attained during degassing. Addition of fresh water into the test tank will require a period of degassing. The system should be run in vacuum long enough to ensure proper degassing of the water. The duration of degassing may be based upon experience or checked with an oxygen meter or other methods.

F.5 Determination of the vapor pressure

The vapor pressure of the test liquid entering the pump should be determined with sufficient accuracy to comply with Appendix F.3. When the vapor pressure is derived from standard data and the measurement of the temperature of the liquid entering the pump, the necessary accuracy of temperature measurement should be demonstrated.
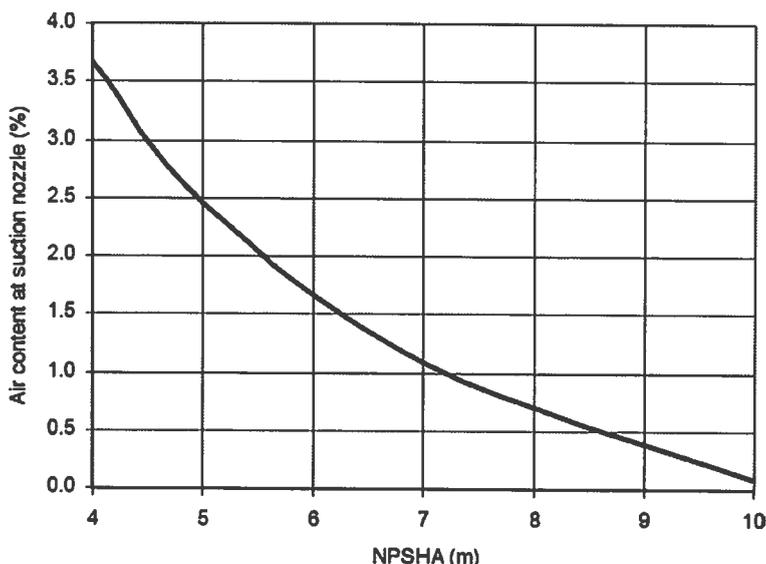
Figure F.1 — Air separating at the pump suction nozzle as a function of NPSHA if the pump draws cold water from a tank with air-saturated water. Air pressure above tank equal to atmospheric pressure.

The source of standard data to be used should be agreed on between manufacturer/supplier and purchaser.

The active element of a temperature-measuring probe should not be less than one eighth of the inlet pipe diameter from the wall of the inlet pipe. If the immersion of the temperature-measuring element in the inlet flow is less than that required by the instrument manufacturer, then a calibration at that immersion depth will be required.

Care should be taken to ensure that temperature-measuring probes inserted into the pump inlet pipe do not influence the measurements of inlet pressure.

F.6 Example of test arrangements

Examples of different test arrangements are described below and in Table 14.6.5.8.2.1.

F.6.1 Closed-loop arrangement

The pump is installed in a closed pipe loop as shown in Figure F.2 in which, by altering the pressure, level, or temperature, the NPSHA is varied without influencing the pump head or rate of flow until cavitation occurs in the pump.

Arrangements for cooling or heating the liquid in the loop may be needed in order to maintain the required temperature, and a gas separation tank may also be required.

A liquid recirculation loop may be necessary to avoid unacceptable temperature difference in the test tank.

The tank should be of sufficient size and so designed as to prevent the entrainment of gas in the pump inlet flow. Additionally stilling screens may be needed in the tank if the average velocity exceeds 0.25 m/s (0.82 ft/s).

F.6.2 Open sump with level control

The pump draws liquid through an unobstructed suction pipe from a sump in which the level of the free liquid surface may be adjusted (see Figure F.3).

F.6.3 Open sump with throttle valve

The pressure of the liquid entering the pump is adjusted by means of a throttle valve installed in the inlet pipe at the lowest practicable level (see Figure F.4).

Figure F.2 — Variation of NPSHA in a closed loop by head and/or temperature controlled
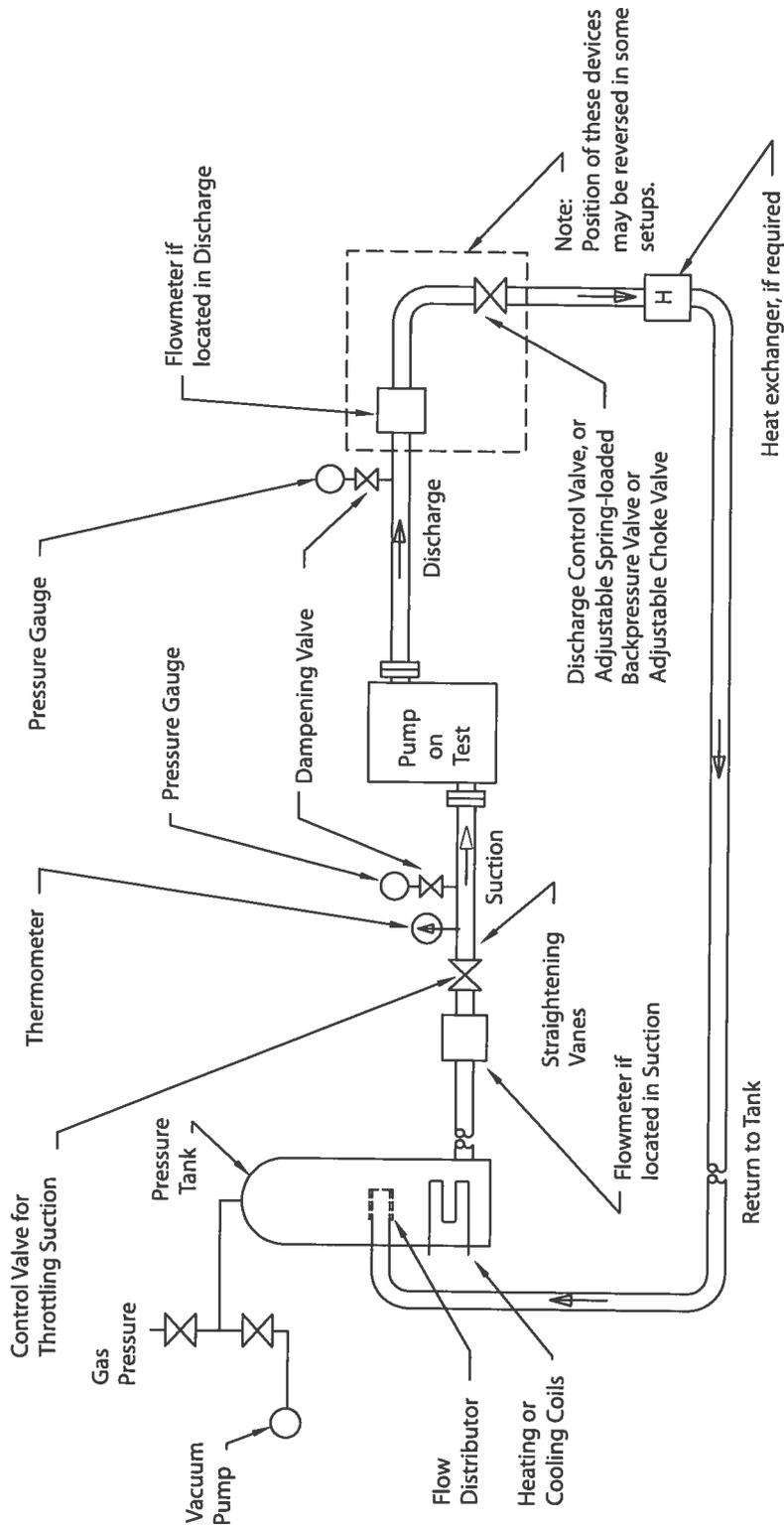
NOTE: Cooling by means of a coil may be replaced by injection of cool water above the free liquid surface and extraction of heated water.
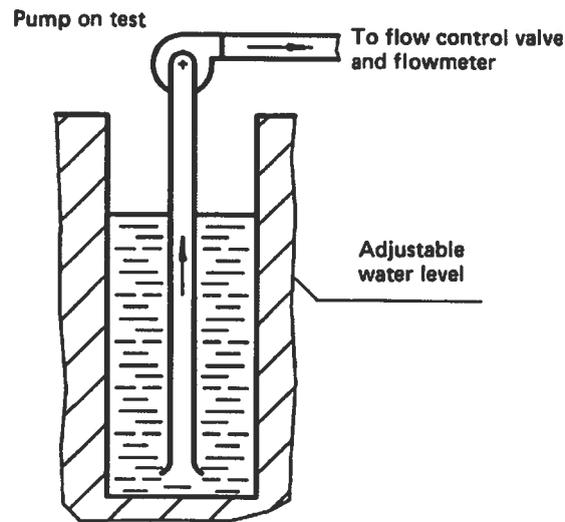
Figure F.3 — Variation of NPSHA by control of liquid level at pump inlet sump
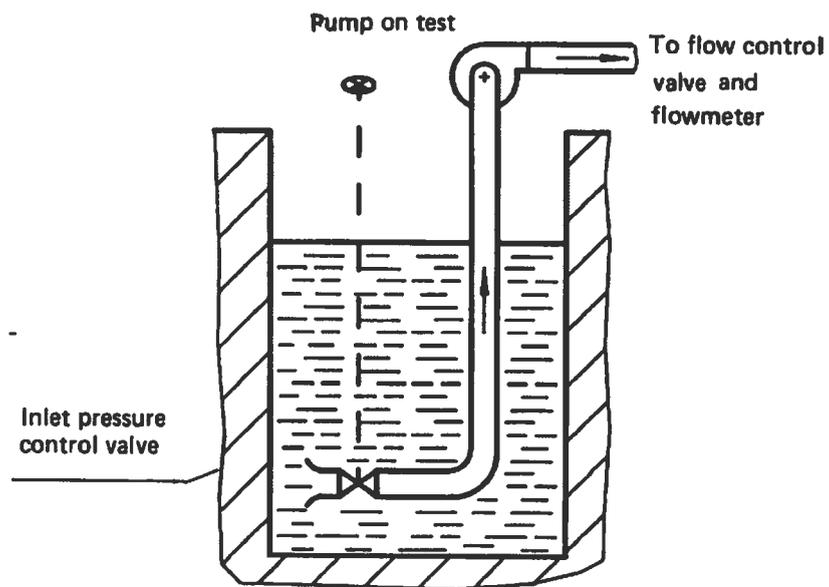
Figure F.4 — Variation of NPSHA by means of an inlet throttle valve

Appendix G

Tests performed on the entire equipment set - string test (informative)

Generating a pump curve requires the measurement of head, capacity, and power. From this information the efficiency of the pump can be calculated. The efficiency shown on the pump curve has always been related to the shaft input power. The published efficiency is the hydraulic power produced by the pump divided by the mechanical input power to the pump shaft. Thus the efficiency published is only that of the pump, not of any other component. From a testing standpoint, the most accurate way to obtain the power data is by direct measurement of the shaft torque and rpm. This is typically done using a torque transducer and a tachometer. These values are then used when calculating the power input to the pump.

A less accurate method, but one that may be specified, is to do a "string" test using the complete assembly with the motor, pump, and drive (e.g., gearbox, belt drive, etc.). The accuracy of this test will be lower than when the pump is tested by itself. In this instance, the power measured is the input power to the motor. The input power to the pump shaft is then calculated by taking into account the published motor and drive efficiencies. Since these efficiencies are not known precisely, this method of calculating pump input power is less accurate than when the shaft torque and rpm are directly measured.

When a VFD is used as a part of the string, it becomes very difficult to obtain an accurate value of input power to the pump shaft. A wattmeter cannot accurately measure the power from the VFD to the motor because of the non-sinusoidal wave form generated by the VFD. A wattmeter can measure the input power to the VFD. But when the input power to the VFD is measured, the efficiency of the VFD must be known to calculate VFD output power to the motor. This information may be available, but it adds yet another degree of error since the motor efficiency will change due to the nonsinusoidal wave form of the output power from the VFD. (Although many VFDs provide a measurement of output power, the value of this measurement is only approximate and is generally not accurate enough for acceptance testing. This reading also does not take into account the reduction in motor efficiency when operated on VFD power.)

The need for string testing with a VFD can come from two requirements. The first is when the customer wishes to use its VFD on the string test. The second is when a string test is required and the customer wishes to have curves produced at a number of different speeds. In both instances the suggested procedure is to conduct one test without using a VFD by running the motor directly across the line. This will allow a complete head-capacity-efficiency curve to be produced at nominal speed. The VFD can then be connected to the motor, and head-capacity curves can be produced at the required speeds without any power data being measured.

Table G.1 gives the influencing factors needed to calculate pump efficiency for different configurations. The configurations are shown from the highest to the lowest measuring accuracy.

It is not possible to obtain pump efficiency during a string test of an engine-driven pump. In this situation, the pump should be tested separately to obtain accurate shaft power measurements.

The pump manufacturer's curves often only provide the end user the required power at the pump input shaft. Furthermore this information is generally provided with the pump being sealed by packing. From an energy consumption standpoint, these data do not provide the user with the true cost to operate the pump.

In fact, it is far more useful to provide wire-to-water efficiency and power consumption curves, but this is rarely requested. Wire-to-water performance can be measured with all of the configurations given in Table G.1 simply by placing a wattmeter at the input to the motor or VFD. These data will allow the end user to know the true power

consumption of the pump system and to evaluate the true operating cost of various seal, drive, motor, and VFD options.

Table G.1 — Influencing factors for calculating pump efficiency for different configurations

Configuration	Drive	Power Measurement	RPM Measurement	Influencing Factors	Pump Efficiency Accuracy
Pump only	Mechanical	Torque transducer	Tachometer	None	Highest
Pump and motor, direct connected	Line power	Wattmeter	Tachometer	(1) Motor efficiency	
Pump and motor, belt or gear driven	Line power	Wattmeter	Tachometer	(1) Motor efficiency (2) Transmission efficiency	
Pump and submersible motor	Line power	Wattmeter	From motor or vibration data	(1) Motor efficiency (2) Seal power consumption (3) Cooling system power consumption	
Pump and motor, direct connected	Motor + VFD	Wattmeter input to VFD	Tachometer	(1) Motor efficiency (2) VFD efficiency (3) Motor efficiency correction for VFD power	
Pump and motor, belt or gear driven	Motor + VFD	Wattmeter input to VFD	Tachometer	(1) Motor efficiency (2) Mechanical drive efficiency (3) VFD efficiency (4) Motor efficiency correction for VFD power	
Pump and submersible motor	Motor + VFD	Wattmeter input to VFD	From motor or vibration data	(1) Motor efficiency (2) Seal power consumption (3) Cooling system power consumption (4) VFD efficiency (5) Motor efficiency correction for VFD power	Lowest

Appendix H

Reporting of test results (informative)

H.1 Performance test report

The applicable acceptance criteria should be clearly indicated on the head/flow curve plot in the form of a vertical line for head limits (at the flow rate guarantee point), and a horizontal line for flow limits (at the head guarantee point), see Figures 14.6.3.4.2a, 14.6.3.4.2b, 14.6.3.4.3a, 14.6.3.4.3b, and 14.6.3.4.3c for examples using various acceptance grades. The ends of the vertical line should represent the upper and lower limits of head; and the ends of the horizontal line should represent the upper and lower limits of flow rate. The lines should intersect at the guarantee point. The head/flow curve must touch or cross at least one of the lines to pass the head/flow acceptance criteria. Measured test points should be shown on the curve.

The following data should be included in pump test report, as applicable:

- Pump data

    - Model number

    - Impeller type

    - Number of impellers

    - Impeller diameter

    - Nominal speed

    - Inlet and outlet nozzle sizes

    - Serial number

- Test condition data

    - Test code and acceptance grade

    - NPSH available

    - Test liquid temperature

- Test data (uncorrected and corrected data)

    - Flow

    - Head

    - Power

- Test motor data (as applicable for factory or job motor)

- Test performed by/witnessed by

MANUFACTURER		PUMP SERIAL NUMBER:		TEST CURVE NUMBER:	CUSTOMER / ORDER INFORMATION:
SPEED n (rpm)	POWER P (kW)	IMPELLER (IDENTIFICATION)	DIAMETER D (m)	PUMP MODEL NUMBER:	
TEST DATE:		TESTER:		ACCEPTANCE CLASS: 3B	

DATA POINT	RATE OF FLOW Q (m³/s)	HEAD				PUMP POWER INPUT P (kW)	PUMP EFFICIENCY η %
		GAUGE p (kPa)	VELOCITY h_v (m)	ELEVATION z (m)	PUMP TOTAL H (m)		
1	0.0000	871.4	0.00	0.70	89.58	51.2	00.0
2	0.0389	778.3	0.24	0.70	80.32	66.4	46.1
3	0.0518	741.0	0.43	0.70	76.71	70.0	55.6
4	0.0680	695.0	0.70	0.70	72.29	75.3	63.9
5	0.0905	642.9	1.25	0.70	67.52	83.0	72.1
6	0.1087	601.5	1.83	0.70	63.88	89.0	76.4
7	0.1345	547.9	2.65	0.70	59.23	96.7	80.7
8	0.1455	501.2	3.26	0.70	55.08	97.4	80.6
9	0.1694	408.5	4.42	0.70	46.79	98.0	79.2

GUARANTEE POINT		CERTIFICATION: TEST WAS PERFORMED WITH CLEAR WATER AT 21 °C AMBIENT TEMPERATURE. HEAD, FLOW, AND POWER DATA ACQUIRED BY ELECTRONIC METERING EQUIPMENT. ACCURACY OF TEST EQUIPMENT IS VERIFIED BY PERIODIC CALIBRATIONS.
FLOW	0.1400 m³/s	
HEAD	55.6 m	
EFFICIENCY	%	
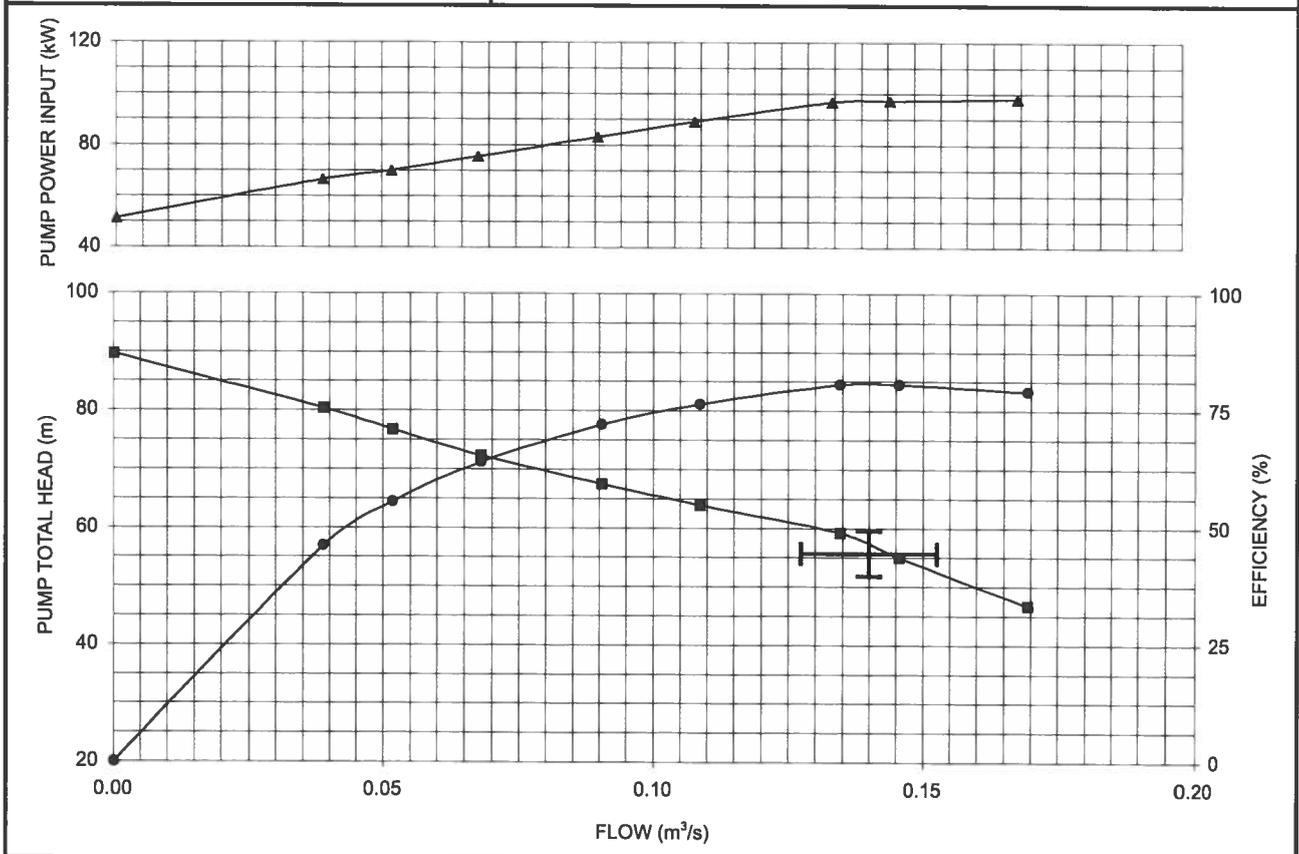
Figure H.1 — Sample pump test curve

H.1.1   NPSH test report requirements

The results of NPSH3 testing may be displayed on the pump performance curve or reported on a separate NPSH test curve.

H.1.2   Pump test sheet

The pump test sheet illustrated in this appendix (Figure H.2) is given for guidance for presenting pump test results and to assist in their interpretation. It does not purport to include all the information required from a pump test, and modifications may be necessary depending on the type of pump and its application, and is subject to agreement between manufacturer and specifier.

PUMP TEST SHEET

Purchaser		Sheet Number		Nature of test	

Pump	Type	Manufacturer's order number	Order number	Inlet diameter
				Outlet diameter
				Impeller diameter

Guaranteed values	Rate of flow (Q_G)	Speed of rotation (n_{sp})	Power input (P_G)
	Total head (H_G)	Efficiency (η_G)	Net positive suction head (NPSH)

Pumped liquid	Temperature (Θ)	Vapor pressure (p_v)	Kinematic viscosity (ν)
	Density (ρ)		Degree of acidity (pH)

Motor	Manufacturer	Test certificate	Number of phases	Voltage
	Type	Power	Speed of rotation	Current

	Rate of flow	Inlet head	Outlet head	NPSH	Torque	Power	Speed of rotation	Gear

Measuring method	Method used
	Constant

Test conditions	Ambient temperature	Barometric pressure	Head correction to reference plane	Inlet
	Temperature of test liquid			Outlet

Result of measurement

		Units	1	2	3	4	5	6	7	8	9	10
Rate of flow	Speed of rotation											
	Time interval											
	Reading											
	Measured flow											
Head	Outlet head reading											
	Inlet head reading											
	Outlet head											
	Inlet head											
	$\Delta v^2/2g$											
	Difference of measuring position											
	Pump total head											
	$\Delta v_1^2/2g$											
	NPSH											
Power	Pump power output P_u											
	Voltage											
	Current											
	Wattmeter reading 1											
	Wattmeter reading 2											
	Total of wattmeter readings											
	Motor power input											
	Motor efficiency											
	Torque reading											
	Gear efficiency											
	Motor power output											
	Pump power input											
	Overall efficiency											
	Pump efficiency											
Values referred to specified speed of rotation	Volume rate of flow											
	Total head											
	Power											
	NPSH											

NOTES			
Date	Test Technician	Representatives of the purchaser	of the manufacturer

Figure H.2 — Example test sheet

Appendix I

Measurement equipment (informative)

I.1   Head measuring apparatus

I.1.1   Spring pressure gauges

This type of gauge uses the mechanical deflection of a loop of tube, plain or spiral (Bourdon dial gauge), or a membrane to indicate pressure.

If this type of apparatus is used to measure the pressure at inlet or outlet, then it is recommended that:

a)   Each apparatus is used within its optimum measuring range (above 40% of its full scale).

b)   The interval between two consecutive scale graduations is between 1.5 and 3 mm (0.06 and 0.12 in).

c)   Such divisions correspond to a maximum of 5% of the pump total head.

The calibration of this measuring apparatus shall be checked regularly.

Figure I.1 shows an arrangement for determining the reference plane of spring pressure gauges.

I.1.2   Electronic pressure transducers

There is a large diversity of pressure transducers, absolute or differential, based on the variation of various mechanical and/or electrical properties. They may be used provided the required accuracy, repeatability, and reliability are achieved; the transducer is used within its optimum measuring range; and the transducer together with its electronic equipment are calibrated regularly by comparison with a pressure device of higher accuracy and reliability.

I.2   Measurement of rotating speed

The speed of rotation can be measured by (1) counting revolutions for a measured interval of time, (2) a direct indicating tachometer, (3) a tachometric dynamo or alternator, (4) an optical or magnetic counter, or (5) a stroboscope.

In the case of a pump driven by an AC motor, the speed of rotation can also be deduced from observations of the grid frequency and motor slip data, either supplied by the motor manufacturer or directly measured (for example, using an induction coil). The speed of rotation is then given by the following formula: $n = \frac{2}{i}\left(f - \frac{i}{\Delta t}\right)$

1 - Reference plane of the manometer
2 - Open to atmosphere

Figure I.1 — Arrangement for determination of reference plane of spring pressure gauges

When the speed of rotation cannot be directly measured on a rotating element due to the design of the pump (for example, for submersible pumps), it can be measured by counting a vibration frequency detected by an accelerometer reading displacement or vibration velocity in a direction radial to the pump shaft.

I.3 Measurement of flow rate

Any flow measuring system may be used for measurement of pump flow rate provided that:

a) The entire flow passing through the pump also passes through the instrument section.

b) It can be demonstrated that the measuring instrument meets the requirements of Table 14.6.3.3.3 of the standard.

The piping upstream of the flow measuring device should be straight, having the same diameter as the device and a length of at least 10 pipe diameters. The piping downstream of the flow measuring device has the same requirements, except that it can have a length of down to five pipe diameters. The lengths are measured from flange to flange.

I.3.1 Measurement by weighing

The weighing method, which gives only the value of the average rate of flow during the time taken to fill the weighing tank, may be considered the most accurate method of flow rate measurement. This procedure is mainly used for calibration of other flow measuring devices.

ISO 4185 and ASME MFC-9M indicate all necessary information for the measurement of the liquid rate of flow by the weighing method.

I.3.2 Volumetric method

The volumetric method approaches the accuracy of the weighing method and similarly only supplies the value of the average rate of flow during the time it takes to fill the gauged rate of flow.

ISO 8316 indicates all necessary information for the measurement of the liquid rate of flow by the volumetric method.

I.3.3 Differential pressure devices

The construction, installation, and use of orifice plates, nozzles, and venturi tubes are the subject of ISO 5167-1 and ASME MFC-3M, while ISO 2186 and ASME MFC-8M give specifications on connecting piping for the manometer.

Attention should particularly be drawn to the minimum straight lengths to be adhered to upstream of the differential pressure device. These are specified in ISO 5167-1 and ASME MFC-3M for various configurations of piping. If it is necessary to place the differential pressure device downstream of the pump (which is not covered in the tables referred to), then the pump may be considered for the purpose of this standard to create a disturbance in the flow equivalent to a single 90-degree bend either in the same plane as the pump volute, the last stage of a multistage pump, or the outlet branch of the pump.

It should also be noted that the diameter of the pipe and the Reynolds number should fall within the ranges specified in ISO 5167-1 and ASME MFC-3M for each type of device. It should be ensured that the flow measuring apparatus is not influenced by cavitation or degassing that can occur, for example, at a control valve. The presence of air can usually be detected by operating the air vents on the measuring device.

It should be possible to check the differential pressure measurement apparatus by comparison with other measuring apparatus. If all the requirements of the relevant standards are met, then the discharge coefficients given in the standards can be used without calibration.

I.3.4 Thin-plate weirs

Particular attention is to be drawn to the great sensitivity of these devices to the upstream flow conditions and thus to the necessity to comply with the prescriptions for the approach channel.

For the application of this standard, the smallest scale division of all instruments used for the measurement of the head over the weir shall not be more than that corresponding to 1.5% of the rate of flow to be measured.

The specifications for the construction, installation, and utilization of rectangular or triangular thin-plate weirs are given in ISO 1438-1, and ISO 4373 indicates prescription for the level measuring device.

I.3.5 Velocity area methods

These methods are the subject of ISO 3354, ISO 3966, and ASME MFC-12M, which deal with discharge measurements in closed conduits by means of current meters and Pitot static tubes. These standards give all the necessary specifications concerning conditions of application, choice, and operation of the apparatus; measurement of local velocities; and calculation of the rate of flow by integration of the velocity distribution.

Except in very long pipe installations, it is preferable that the measuring section should be placed upstream of the pump to avoid too much turbulence or swirling flow.

The complication of these methods does not justify their use for grade 2 and 3 tests, but they are sometimes the only ones that can be applied when testing pumps with large rates of flow for grade 1 tests.

I.3.6 Electromagnetic method

Electromagnetic flowmeters do not require very long upstream straight pipe lengths (a length of five times the pipe diameter is in most cases sufficient) and achieve a very good accuracy.

Requirements for electromagnetic velocity meters should be in accordance with ISO 9104 and ASME MFC-16.

I.3.7 Ultrasonic method

Ultrasonic flowmeters are very sensitive to the velocity distribution and should be calibrated in their actual conditions of operation.

Requirements for ultrasonic velocity meters should be in accordance with ISO/TR 12765 (ISO 6416) and ASME MFC-5M.

I.3.8 Turbine meters

Turbine meters do not require very long upstream straight pipe lengths (a length of five times the pipe diameter is in most cases sufficient) and achieve a very good accuracy.

Requirements for turbine meters should be in accordance with ISO 9951 and ASME MFC-4M (for gases).

I.3.9 Tracer and other methods

These methods, applied to the measurement of the flow rate in the pipes, are the subject of ISO 2975 and ASME MFC-13M, the different parts of which cover both the dilution method (constant rate injection) and transit time method, each method using either radioactive or chemical tracers.

Some apparatus, such as vortex or variable area flowmeters, may be used provided they are calibrated beforehand by means of one of the primary methods described in this appendix. When installed permanently on a test facility, the possibility of a periodic check of their calibration shall be taken into account.

The calibration should bear on the whole of the flowmeter and the associated measuring system. The calibration should normally be carried out under the actual operating conditions (head, temperature, water quality, etc.) prevailing during the tests, and attention should be given to the fact that the flowmeter is not affected by cavitation during the tests.

As for the velocity area methods, the tracer methods are justified only for grade 1 tests. They should only be used by specialized staff, and it should be noted that the use of radioactive tracers is subject to certain constraints.

I.4 Measurement of pump power input

I.4.1 General

Pump power input may be determined by dynamometers, torque meters, calibrated motors, wattmeters, or other devices that can be demonstrated to meet the requirements of Section 14.6.3.3.3 of the standard.

Where the power input to an electric motor coupled to an intermediate gear, or the speed of rotation and torque measured by a torque meter between gear and motor are used as a means for determining the pump power input, the method for determining the losses due to the reduction gear should be stated in the contract.

If necessary, see ISO 5198 for more information on the methods described in the following.

I.4.2 Torque measurement

Torque should be measured by a suitable dynamometer or a torque meter capable of complying with the requirement of Section 14.6.3.3.3 of the standard.

Measurement of torque and speed of rotation should, within practical limits, be simultaneous.

I.4.3 Electric power measurements

Where the electrical power input to an electric motor coupled directly to the pump is used as a means of determining the pump power input, the motor should be operated only at conditions where the efficiency is known with sufficient accuracy. Motor efficiency should be determined in accordance with the recommendations of IEC 60034-2 and is to be stated by the motor manufacturer.

This efficiency does not take into account motor cable losses.

The electric power input to the AC motor should be measured by either the two-wattmeter or the three-wattmeter method. This allows using either single-phase wattmeters or a wattmeter measuring two or three phases simultaneously or integrating watt-hour-meters.

In the case of a DC motor, either a wattmeter, or an ampere meter and a voltmeter may be used.

The type and grade of accuracy of the indicating instruments for measuring electrical power should be in accordance with IEC 60051.

I.5 Special cases

I.5.1 Pumps with inaccessible ends

In the case of combined motor-pump units (for example, a submersible pump or monobloc pump, or separate pump and motor with overall efficiency guarantee), the power of the unit should be measured at the motor terminals if accessible. When a submersible pump is involved, the measurement should be effected at the incoming end of the cables; cable losses should be taken into account and specified in the contract. The efficiency given should be that of the combined unit proper, excluding the cable and the starter losses.

I.5.2 Deep-well pumps

In this case, the power absorbed by the thrust bearing and the vertical shafting and bearings should be taken into account. Because deep-well pumps in general are not tested with the entire stand pipe attached, unless the acceptance test is performed on-site, the thrust and vertical shaft bearing losses should be estimated and stated by the manufacturer/supplier.

I.5.3 Motor pump units with common axial bearing (other than close-coupled pumps)

In this case, if the power and the efficiency of the motor and those of the pump should be determined separately, then the influence of the axial thrust, and possibly of the weight of the pump rotor on the losses in the thrust bearing, should be taken into account.

I.6 Determination of motor pump unit overall efficiency

To determine the overall (wire-to-water) efficiency of a motor pump unit (i.e., pump and driver coupled together and treated as an integral unit), the driver input power and the pump output power are measured with the driver working under conditions specified in the contract. The ratio between output power and input power yields the overall pump unit efficiency.

Appendix J

Suitable time periods for calibration of test instruments (informative)

J.1 Recalibration interval

The frequency of instrument recalibration depends on usage and the design of the equipment. Table J.1 is based on experience with general usage of instruments. If historical data exist to support a longer recalibration interval, then this should be acceptable to all parties. If an instrument is physically abused or overloaded, then it should be recalibrated before being used.

Table J.1 — Instrument recalibration intervals

EQUIPMENT	PERIOD (Years)	EQUIPMENT	PERIOD (Years)
Rate of Flow		**Input Power**	
Weigh tank	1	Dynamometer	0.5
Volumetric tank	10	Torque meter	1
Venturi, nozzle, orifice, weir	NR[a]	Calibrated motor	NR[b]
Turbine	1	Wattmeter	1
Electromagnetic	1[c]	Gears	10
Ultrasonic	0.5	**Head**	
Current meter	2	Bourdon tube	0.33
Pump Speed		Dead weight	NR
Tachometer	3	Manometer	NR
Electronic (gear teeth)	NR[c]	Transducers	0.33
Frequency Responsive Devices		**Temperature**	
Magnetic	10	Electric	2
Optical	10	Mercury	5
Stroboscopes	5		

[a] Not required unless a change of a critical dimension is suspected.

[b] Not required unless damage is suspected.

[c] Secondary (electronic processor). The primary section should be recalibrated every five years, unless electrical or mechanical failure.

Appendix K

Special test methods (informative)

K.1   General

For certain pump acceptance testing situations there are specialized test methods that may be more practical to use than those described earlier. These and other possible methods are typically highly specialized and require experience and intimate knowledge of the respective methods and processes in order to obtain accurate and repeatable test results. Two examples of special test methods are given below.

- Testing of a smaller-scale pump model. This method requires that a geometrically similar model pump is constructed such that the entire internal pump geometry is linearly scaled down, including surface roughness and gaps. The pump test results, including efficiency, can be scaled up to accurately represent the full-scale prototype pump. The model should be constructed as large as possible to achieve the best accuracy. See Appendix K.2 below for a detailed description of this test method. Japanese Industrial Standard JIS B 8327:2002 also describes this test method.

- Pump efficiency testing may be performed by precisely measuring the difference in pumped media temperature between the suction and discharge of the pump. This is commonly referred to as the *thermodynamic test method*. See ISO Standard 5198 for a detailed description of this method.

K.2   Model tests for pump acceptance

Acceptance of hydraulic performance for very large pumps may sometimes be advantageous to determine by using a geometrically similar, reduced-scale model of the prototype pump. Even if it might be feasible to test the prototype unit in the factory, a model may be tested with greater accuracy and thoroughness.

The relationships between the model and the prototype are governed by the rules of hydraulic similarity. This implies geometric, kinematic, and dynamic similarity.

Geometric similarity refers to the similarity of dimensions and shapes and requires that the ratio of any two dimensions in the model be the same as the corresponding ratio in the prototype. In many instances, geometric similarity cannot be fully achieved, for example, variation in surface roughness height is not scaled in relation to model scale.

Kinematic similarity relates to similarity of motion. Essentially, velocities and accelerations at equivalent points in the model and prototype should be in a constant ratio.

Dynamic similarity requires that forces at equivalent points in the model and prototype should be in a constant ratio. For flow of a homogeneous liquid in a rotodynamic pump equivalence of pressure, viscous and inertial effects should be achieved.

Given that the model pump is constructed to be geometrically similar with the prototype, achieving dynamic similarity will imply that kinematic similarity is also achieved.

Testing models in advance of final design and installation of a large pump not only provides advance assurance of performance, but also makes design alterations possible at a lower cost and faster in time for incorporation in the prototype pump.

Not all installations lend themselves to a practical model testing. The pumping of water carrying considerable quantities of solids or other foreign material is not readily reproduced in model operation. This standard, therefore, is limited to scale models pumping clear water, free from abnormal quantities of air or solids.

The effects of free-surface liquid disturbances in open channel sumps, interference between neighboring pump units, and hydraulic conditions caused by poor approach flow quality are not covered by a model pump acceptance test. See ANSI/HI 9.8 *Pump Intake Design* for sump model tests addressing these types of problems.

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-size pumps. If NPSH data 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.

K.2.1   Pump model test

By adopting a fixed linear scale model for prototype pump, accurate performances may be obtained. The model pump shall have an impeller diameter of not less than 0.30 m (11.8 in) in diameter. Typical model scale ratios are found in the interval between 2 and 6. The pump manufacturer shall select the model-to-prototype scale. All dimensions of the model hydraulic passages must be in accordance with the selected model-to-prototype ratio. Efforts should be made to approach this scale relationship for surface finishes and gaps, although it may not be possible to completely meet this requirement. This will mainly affect the pump's efficiency. Formulas below give guidance on how to handle such situations. Subscript "m" refers to model and "p" to prototype.

If corresponding diameters of model and prototype are D_m and D_p, respectively, then the model speed n_m and model rate of flow Q_m, under the test head H_m, must agree with the relationships:

$$\frac{n_m}{n_p} = \frac{D_p}{D_m} \times \left(\frac{H_m}{H_p}\right)^{0.5}, \text{ and } \frac{Q_m}{Q_p} = \left(\frac{D_m}{D_p}\right)^2 \times \left(\frac{H_m}{H_p}\right)^{0.5}$$

One simplification of the above is to require that the model pump be run at heads equal to the prototype heads. This results in equal model and prototype velocities. For this special case, the formula can be simplified as shown below.

$$\frac{n_m}{n_p} = \left(\frac{D_p}{D_m}\right), \text{ and } \frac{Q_m}{Q_p} = \left(\frac{D_m}{D_p}\right)^2$$

Example (Metric): A single-stage pump designed to deliver 5.556 m³/s against a head of 120 m at 450 rpm and have an impeller diameter of 2.00 m. This pump is too large for a factory test and, in place of such test on the prototype pump, a model test is to be tested at a reduced head of 100 m. The model impeller is to be 0.50 m in diameter.

Determine speed and rate of flow for the above model test.

Apply the above relationships:

$$\frac{n_m}{n_p} = \left(\frac{D_p}{D_m}\right) \times \left(\frac{H_m}{H_p}\right)^{0.5}$$

or

$$n_m = n_p \times \left(\frac{D_p}{D_m}\right) \times \left(\frac{H_m}{H_p}\right)^{0.5} = \left[450 \times \left(\frac{2.0}{0.5}\right) \times \left(\frac{100}{120}\right)^{0.5}\right] = 1643 \text{ rpm}$$

$$\frac{Q_m}{Q_p} = \left(\frac{D_m}{D_p}\right) \times \left(\frac{H_m}{H_p}\right)^{0.5}$$

or

$$Q_m = Q_p \times \left(\frac{D_m}{D_p}\right)^2 \times \left(\frac{H_m}{H_p}\right)^{0.5} = 5.556 \times \left(\frac{0.50}{2.00}\right)^2 \times \left(\frac{100}{120}\right)^{0.50} = 0.317 \text{ m}^3/\text{s}$$

The model test should therefore be run at a speed of 1643 rpm delivering 0.317 m³/s against a head of 100 m.

To check these results, it will be noted that the prototype pump type number is:

$$K_p = \frac{2\pi n_p Q_p^{0.5}}{(gH_p)^{0.75}} = \frac{6.28 \times 450 \times 5.556^{0.5}}{60 \times (9.81 \times 120)^{0.75}} = 0.55$$

And the model pump type number will be:

$$K_m = \frac{2\pi n_m Q_m^{0.5}}{(gH_m)^{0.75}} = \frac{6.28 \times 1643 \times 0.371^{0.5}}{60 \times (9.81 \times 100)^{0.75}} = 0.55$$

Therefore the type numbers (specific speeds) are the same as required.

K.2.2 Efficiency scaleup

The efficiency of the model will not, in general, be 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. As mentioned before, 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 still be lower than that of the larger unit due to operating at a lower Reynolds number. Further, it is generally not practical to model running clearances or bearing sizes. When such is the case, the model efficiency will be reduced.

The Reynolds number can be calculated with the following formula:

$$Re = n \times \frac{D^2}{\nu}$$

Where:

n = speed of rotation, in rps

D = impeller diameter, in m

ν = kinematic viscosity, in m²/s

With prior agreement between the manufacturer and the user, the prototype efficiency may be scaled up from the tested model values obtained.

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. The calculation can be applied conveniently according to the formula in use for rotodynamic pumps; namely

$$\frac{1-\eta_m}{1-\eta_p} = \left(\frac{Re_p}{Re_m}\right)^x$$

For the special case where the model is run at prototype velocities, the following simplified formula may be used:

$$\frac{1-\eta_m}{1-\eta_p} = \left(\frac{D_p}{D_m}\right)^x$$

The values for the exponent (x) have been found to vary between 0.05 and 0.26, depending on relative surface roughness of model and prototype and other factors. When no supporting data are available for selection of the exponent, a conservative value of 0.10 shall be used. See K.2.4 for the affinity formulas to be used when applying an efficiency scaleup.

K.2.3   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.

It should be pointed out, however, that the reduced-size model, coupled with an increase in head corresponding to an increase in speed tends to minimize the change in Reynolds number; that is, the product of flow velocity and linear dimensions tends to be equal for the model and the prototype. This effect tends to restore dynamic similarity in model and prototype and to equalize efficiencies and other performance factors. With increased head, however, the suction specific speed value must still be the same for the model and in the prototype, 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 increase in suction lift.

K.2.4   Model test scaling formulas

When testing a pump where the model efficiency is to be scaled up to prototype values, the model best efficiency point (BEP) shall be scaled up to determine the prototype value. The ratio of prototype to model efficiency at the BEP shall be used to correct the efficiency at all other points away from the BEP.

When the pump type number K is less than 0.76 (specific speed N_s is less than 2000 in US customary units), the disk friction can be assumed to be the dominant factor in determining pump efficiency, and increases in prototype efficiency shall be accounted for by a reduction in prototype power. This is shown in the following affinity formulas:

$$\frac{1-\eta_m}{1-\eta_p} = \left(\frac{Re_p}{Re_m}\right)^x \text{, therefore:}$$

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$$\eta_p = 1 - \frac{1 - \eta_m}{\left(\frac{Re_p}{Re_m}\right)^x} = 1 - (1 - \eta_m) \times \left(\frac{Re_m}{Re_p}\right)^x$$

$$Q_p = Q_m \times \left(\frac{D_p}{D_m}\right)^3 \times \frac{n_p}{n_m}$$

$$H_p = H_m \times \left(\frac{D_p}{D_m}\right)^2 \times \left(\frac{n_p}{n_m}\right)^2 \times \frac{\eta_p}{\eta_m}$$

$$P_p = P_m \times \left(\frac{D_p}{D_m}\right)^5 \times \left(\frac{n_p}{n_m}\right)^3$$

$$S_p = S_m$$

Example (US customary units): A single-stage pump designed to deliver 60,000 gpm against a head of 180 ft at 505 rpm and have an impeller diameter of 54.0 in [$n_s = (60,000)^{0.5} \times 505/(180^{0.75}) = 2517$]. The prototype driver will be 3500 hp. The guaranteed minimum efficiency at the rated point is 88%. The expected NPSHR at the rated point is 30 ft, while the minimum NPSHA is 35 ft. This pump is too large for the manufacturer's test facility. A model test at prototype heads will be undertaken to confirm the prototype performance. The model impeller is to be 18.0 in. in diameter.

Determine model rated points:

Since the model is to be run at prototype heads:

$$n_m = n_p \times \frac{D_p}{D_m} = 505 \times \frac{54.0}{18.0} = 1515 \text{ rpm}$$

$$Q_m = \frac{Q_p}{\left(\frac{D_p}{D_m}\right)^3} \times \frac{n_m}{n_p} = 60,000 \times \frac{\left(\frac{1515}{505}\right)}{\left(\frac{54.0}{18.0}\right)^3} = 6667 \text{ gpm}$$

$$H_m = H_p = 180 \text{ ft}$$

$NPSHA_m = NPSHA_p = 35.0$

The model test is completed and demonstrates the following results:

H_m = 178 ft at 6667 gpm at 1515 rpm

P_m = 343.7 bhp at 6667 gpm at 1515 rpm

η_m = 87.2% at 6667 gpm at 1515 rpm

$NPSH3$ = 29.5 ft at 6667 gpm at 1515 rpm

Q_m = 7000 at BEP at 1515 rpm

H_m = 172 ft at 7000 gpm (BEP) at 1515 rpm

η_m = 87.8% at 7000 gpm (BEP) at 1515 rpm

Calculate the prototype efficiency at the BEP:

$$\eta_p = 1 - (1 - \eta_m) \times \left(\frac{Re_m}{Re_p}\right)^x$$

Since the model was run at a head equal to the prototype, and since no prior data are available to determine the exponent, this equation becomes:

$$\eta_p = 1 - (1 - \eta_m) \times \left(\frac{D_m}{D_p}\right)^x = 1 - (1 - 0.878) \times \left(\frac{18.0}{54.0}\right)^{0.10} = 0.8907 = 89.1\%$$

Calculate the prototype performance at the rated point:

$$\eta_p = 87.2 \times \frac{89.1}{87.8} = 87.2 \times 1.0148 = 88.5\%$$

$$Q_p = Q_m \times \left(\frac{D_p}{D_m}\right)^3 \times \left(\frac{n_p}{n_m}\right)^2 = 6667 \times \left(\frac{54.0}{18.0}\right)^3 \times \frac{505}{1515} = 60{,}003 \text{ gpm}$$

$$H_p = H_m \times \left(\frac{D_p}{D_m}\right)^2 \times \left(\frac{n_p}{n_m}\right)^2 \times \frac{n_p}{n_m} = 178 \times \left(\frac{54.0}{18.0}\right)^2 \times \left(\frac{505}{1515}\right)^2 \times \frac{89.1}{87.8} = 180.6 \text{ ft}$$

$$P_p = P_m \times \left(\frac{D_p}{D_m}\right)^5 \times \left(\frac{n_p}{n_m}\right)^3 = 343.7 \times \left(\frac{54.0}{18.0}\right)^5 \times \left(\frac{505}{1515}\right)^3 = 3093 \text{ bhp}$$

$NPSHR_p = NPSHR_m = NPSH3_m = 29.5$ ft

Appendix L

Unit conversions (informative)

This appendix gives factors for conversion units used within this document. The conversion factor is the number by which the value expressed in various units should be multiplied to find the corresponding value in metric units.

Table L.1 — Conversion factors

Quantity	Symbol of metric unit	Various units		Conversion factors
		Name	Symbol	
(Volume) rate of flow	m^3/s	liter per second	L/s	10^{-3}
		cubic meter per hour	m^3/h	1/3600
		liter per hour	L/h	1/3,600,000
		liter per minute	L/min	1/60,000
		imperial gallon per minute	gal (UK)/min	75.77×10^{-6}
		cubic foot per second	ft^3/s	28.3168×10^{-3}
		gallon (US) per minute	gal (US)/min	63.09×10^{-6}
		barrel (US) per hour (petroleum)	barrel (US)/h	44.16×10^{-6}
Mass rate of flow	kg/s	metric ton per second	t/s	10^3
		metric ton per hour	t/h	10/36
		kilogram per hour	kg/h	1/3600
		pound per second	lb/s	0.45359237
Pressure	Pa	kilopond per square centimeter	kp/cm^2	98,066.5
		kilogram-force per square centimeter	kgf/cm^2	98,066.5
		bar	bar	10^5
		hectopieze	hpz	10^5
		torr	torr	133.322
		conventional millimeter of mercury	mm Hg	133.322
		conventional millimeter of water	$mm\ H_2O$	9.80665
		poundal per square foot	pdl/ft^2	1.48816
		standard atmosphere	atm	101,325
		pound-force per square inch	lbf/in^2 (psi)	6894.76
Density	kg/m^3	kilogram per cubic decimeter	kg/dm^3	10^3
		gram per cubic centimeter	g/cm^3	10^3
		pound per cubic foot	lb/ft^3	16.0185
Power	W	kilowatt	kW	10^3
		kilopond meter per second	$kp \cdot m/s$	9.80665
		I.T. kilocalorie per hour	$kcal_{IT}/h$	1.163
		cheval vapor	ch	735.5
		horsepower	hp	745.7
		British thermal unit per hour	Btu/h	0.293071
		kilogram-force meter per second	$kgf \cdot m/s$	9.80665

Table L.1 — Conversion factors (*continued*)

Quantity	Symbol of metric unit	Various units		Conversion factors
		Name	Symbol	
Viscosity (dynamic viscosity)	Pa•s	poise dyne second per square centimeter gram per second centimeter kilopond second per square meter Poundal second per square foot	P $dyn \cdot s/cm^2$ $g/s \cdot cm$ $kp \cdot s/m^2$ $pdl \cdot s/ft^2$	10^{-1} 10^{-1} 10^{-1} 9.80665 1.48816
Kinematic Viscosity	m^2/s	stokes square foot per second	$St = cm^2/s$ ft^2/s	10^{-4} 92.903×10^{-3}
Pump type number	K^a	Specific speed - based on cubic meter per second and meter - assumed dimensionless	n_s	52.93
		Specific speed - based on gallon per minute and foot - assumed dimensionless	N_s	2733.72

[a] Pump type number K is not a metric unit but is derived from consistent metric units, see Table 14.6.2.1, row 3.1.37.

Appendix M

References (informative)

The following referenced documents are presented to help the user in considering factors beyond this standard; they do not form a part of this standard.

ASME – American Society of Mechanical Engineers

Fluid meters, Their Theory and Application

ASME MFC-3M, *Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi*

ASME MFC-4M, *Measurement of Gas Flow by Turbine Meters*

ASME MFC-5M, *Measurement of Liquid Flow in Closed Conduits Using Transit-Time Ultrasonic Flowmeters*

ASME MFC-8M, *Fluid Flow in Closed Conduits — Connections for Pressure Signal Transmissions between Primary and Secondary Devices*

ASME MFC-9M, *Measurement of Liquid Flow in Closed Conduits by Weighing Method*

ASME MFC-12M, *Measurement of Fluid Flow in Closed Conduits Using Multiport Averaging Pitot Primary Elements*

ASME MFC-13M, *Measurement of Fluid Flow in Closed Conduits: Tracer Methods*

ASME MFC-16, *Measurement of Liquid Flow in Closed Conduits with Electromagnetic Flowmeters*

ANSI/ASME PTC 18-1992, *Hydraulic Turbines*

American Society for Mechanical Engineers
Three Park Avenue
New York, NY 10016-5990

www.asme.org

IEEE - Institute of Electrical and Electronics Engineers

IEEE Standard 552, *Application of Power Measuring Equipment*

ANSI/IEEE 112, *Test Procedures for Polyphase Induction Motors & Generators*

ANSI/IEEE 113, *Test Procedures for Direct Current Machines*

ANSI/IEEE 115, *Test Procedures for Synchronous Machines*

Institute of Electrical and Electronics Engineers
3 Park Ave., 17th Floor
New York, NY 10016-5997

www.ieee.org

IEC – International Electrotechnical Commission

IEC Publication 34-2, *Recommendations for rotating electrical machinery (excluding machines for traction vehicles) — Part 2: Determination of efficiency of rotating electrical machinery*

IEC Publication 41, *International code for the field acceptance tests of hydraulic turbines*

IEC Publication 51, *Recommendations for direct acting electrical measuring instruments and their accessories*

IEC Publication 497, *International code for model acceptance tests of storage pumps*

International Electrotechnical Commission
3, rue de Varembé
P.O. Box 131
CH - 1211 Geneva 20, Switzerland

www.iec.ch

ISO - International Organization for Standardization

ISO 31, *General principles concerning quantities, units and symbols*

ISO 1438-1, *Water flow measurement in open channels using weirs and venturi flumes — Part 1: Thin-plate weirs*

ISO 2186, *Fluid flow in closed conduits — Connections for pressure signal transmissions between primary and secondary elements*

ISO 2372, *Mechanical vibrations of machines with operating speeds from 10 to 200 rev/s — Basis for specifying evaluation standards*

ISO 2975-1, *Measurement of water flow in closed conduits — Tracer methods — Part 1: General*

ISO 2975-2, *Measurement of water flow in closed conduits — Tracer methods — Part 2: Constant rate injection method using non-radioactive tracers*

ISO 2975-3, *Measurement of water flow in closed conduits — Tracer methods — Part 3: Constant rate injection method using radioactive tracers*

ISO 2975-6, *Measurement of water flow in closed conduits — Tracer methods — Part 6: Transit time method using non-radioactive tracers*

ISO 2975-7, *Measurement of water flow in closed conduits — Tracer methods — Part 7: Transit time method using radioactive tracers*

ISO 3354, *Measurement of clean water flow in closed conduits — Velocity-area method using current-meters in full conduits and under regular flow conditions*

ISO 3740, *Acoustics — Determination of sound power levels of noise sources — Guidelines for the use of basic standards and for the preparation of noise test codes*

ISO 3744, *Acoustics — Determination of sound power levels of noise sources — Engineering methods for freefield conditions over a reflecting plane*

ISO 3745, *Acoustics — Determination of sound power levels of noise sources — Precision methods for anechoic and semi-anechoic rooms*

ISO 3746, *Acoustics — Determination of sound power levels of noise sources — Survey method*

ISO 3945, *Mechanical vibration of large rotating machines with speed range from 10 to 200 rev/s — Measurement and evaluation of vibration severity in situ*

ISO 3966, *Measurement of fluid flow in closed conduits — Velocity area method using Pitot static tubes*

ISO 4185, *Measurement of liquid flow in closed conduits — Weighing method*

ISO 4373, *Measurement of liquid flow in the channels — Water level measuring devices*

ISO 5167-1, *Measurement of fluid flow by means of pressure differential devices — Part 1: Orifice plates, nozzles and Venturi tubes inserted in circular cross section conduits running full*

ISO 5198, *Centrifugal, mixed flow and axial pumps — Code for hydraulic performance tests — Precision class*

ISO 6081, *Acoustics — Noise emitted by machinery and equipment — Guidelines for the preparation of test codes of engineering class requiring noise measurements at the operator's or bystander's position*

ISO 7194, *Measurement of fluid flow in closed conduits — Velocity-area methods of flow measurement in swirling or asymmetric flow conditions in circular ducts by means of current-meters or Pitot-static tubes*

ISO 8316, *Measurement of liquid flow in closed conduits — Method by collection of the liquid in a volumetric tank*

ISO 9104, *Methods of evaluating the performance of electromagnetic flowmeters for liquids in closed conduits*

Appendix N

Index

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

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

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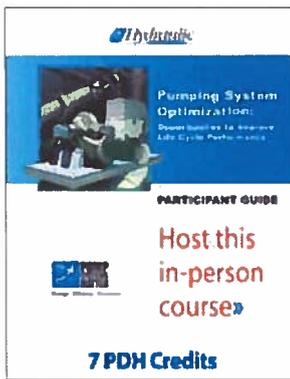Individual Standards
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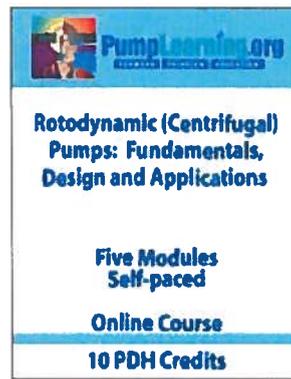
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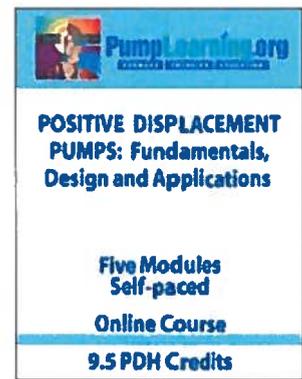
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