ANSI/HI 9.6.4-2009

American National Standard for

Rotodynamic Pumps

for Vibration Measurements and Allowable Values

ANSI/HI 9.6.4-2009

Hydraulic Institute

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

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Sponsor

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Approved May 13, 2009

American National Standards Institute, Inc.

American National Standard

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Foreword (Not part of Standard)

Purpose and aim of the Hydraulic Institute

The purpose and aims of the Institute are to promote the continued growth of pump knowledge for the interest of pump users and manufacturers and to 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 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 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.

Major revisions

This standard, ANSI/HI 9.6.4-2009 *Rotodynamic Pumps for Vibration Measurements and Allowable Values*, was last published as ANSI/HI 9.6.4-2000. This updated edition is essentially a complete rewrite as compared to the previously published standard. In order to assist users of the standard in recognizing the changes, major revisions to the standard are provided as follows.

Tutorial information regarding dynamic analysis has been removed. This topic will be covered in a new document that is planned to be published as new guideline, ANSI/HI 9.6.8 *Dynamics of Pumping Machinery*. The new document was in the development stage at the time of this writing and a precise publication date is not known.

The scope of application of the standard has been clarified to identify pump types that are excluded from the standard.

Metric units are now primary, with US customary units secondary. For speeds above 600 rpm, the measurement quantity to be used for measuring the vibration of nonrotating parts of rotodynamic pumps is velocity in millimeters per second RMS or inches per second RMS.

Additional requirements have been added for operational speeds 600 rpm and lower. For speeds 600 rpm and below, measurement of the overall peak-to-peak displacement is required in addition to overall RMS velocity.

The Hydraulic Institute standard pump type nomenclature provided in ANSI/HI 1.1-1.2 *Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions* and ANSI/HI 2.1-2.2 *Rotodynamic (Vertical) Pumps for Nomenclature and Definitions* has been incorporated, with a new system of icons introduced to assist the user in recognizing the pump types and identifying the measurement locations.

Although this standard requires that the vibration test location is in the field by default, levels are provided for optional factory tests for most pump types.

The allowable vibration requirements have been updated to reflect acceptable performance both for field and factory tests, based on hundreds of data points obtained in surveys conducted since the publication of the previous edition. The revised allowable values have been compared with those presented by other standards organizations, including ISO and API, and can be considered as commensurate.

Values are presented for preferred operating region (POR) and allowable operating region (AOR).

To assist the user of the standard, a suggested vibration test report form is provided in an appendix.

Recommended default initial field alarm and trip settings are provided in an appendix.

A vibration troubleshooting chart is provided in an appendix.

Information on transducer types, mounting, and sensitivity is provided in two appendices.

Units of measurement

Metric units of measurement are used, and corresponding US customary units appear in brackets. Charts, graphs, and sample calculations are also shown in both metric and US customary units. Since 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 to be applied be stated in reference to this standard. If no such statement is provided, metric units shall govern.

Consensus

Consensus for this standard was achieved by use of the canvass method. The following organizations, recognized as having interest in rotodynamic pumps for vibration measurements and allowable values, were contacted prior to the approval of this revision of the standard. Inclusion in the list does not necessarily imply that the organization concurred with the submittal of the proposed standard to ANSI.

Anspach Consulting
Baldor Electric Company
Bechtel Power Corporation
Black & Veatch
Budris Consulting
Buse, Fred – Consultant
Flowserve Pump Division
Grundfos Pumps Corporation
Healy Engineering
ITT - Industrial Process

ITT - Residential & Commercial Water
Malcolm Pirnie
National Pump Company, LLC
Patterson Pump Company
Peerless Pump Company
Pentair Water
Powell Kugler, Inc.
Sulzer Pumps (US) Inc.
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 – Jack Claxton, Patterson Pump Company
Vice-Chair – Michael Cropper, Sulzer Pumps (US) Inc.

Committee Members	**Company**
Stefan Abelin	ITT - Water & Wastewater
Graeme Addie	GIW Industries, Inc.
Thomas Angle	Weir Specialty Pumps
John Anspach	John Anspach Consulting
Julian Atchia	SJE-Rhombus®
Allen Behring	A.G. Behring Company
James Bonifas	Emerson Motors/US Motors
Charles Cappellino	ITT - Industrial Process
James Healy	Healy Engineering
Thomas Hendrey	Whitley Burchett & Associates
Greg Highfill	A.R. Wilfley & Sons, Inc.
Michael Hiscock	South Florida Water Management District
Allen Hobratschk	National Pump Company, LLC
Al Iseppon	Pentair Water
Garr Jones	Brown and Caldwell
Dimitar Kalchev	Peerless Pump Company
Mika Kaplan	King County Wastewater Treatment Division
Yuri Khazanov	InCheck Technologies Inc.
Zan Kugler	Powell Kugler, Inc.
William Marscher	Mechanical Solutions, Inc.
Tony Radcliff	Grundfos Pumps Corporation
James Roberts	ITT - Residential & Commercial Water
Mark Rosebraugh	Yeomans Chicago Corporation
Aleksander Roudnev	Weir Minerals North America
Arnold Sdano	Fairbanks Morse Pump Corporation
Ernest Sturtz	CDM
Roger Turley	Flowserve Pump Division
Fred Walker	Weir Floway, Inc.
Brett Zerba	TACO, Inc.

Alternate	**Company**
Charles Allaben	CDM
Edward Allis	Peerless Pump Company
Lech Bobowski	Peerless Pump Company
Randal Ferman	Flowserve Pump Division
Mark Jaminet	ITT - Water & Wastewater
John Kelly	Yeomans Chicago Corporation
Gary Leander	Weir Minerals North America
Patrick Moyer	ITT - Residential & Commercial Water

Michael Mueller	Flowserve Pump Division
Richard O'Donnell	ITT - Industrial Process
Sami Sarrouh	Brown and Caldwell
Robert Visintainer	GIW Industries, Inc.

Experts

Lyn Greenhill	Dyna Tech
Mark Corbo	No Bull Engineering

Additional Contributors

William A. Beekman
Fred Buse

9.6.4 Rotodynamic Pumps for Vibration Measurements and Allowable Values

9.6.4.1 Introduction and scope

9.6.4.1.1 Introduction

This standard pertains to evaluation of vibration when the vibration measurements are made on nonrotating parts (bearing housings) of rotodynamic pumps. It provides specific maximum allowable vibration values measured on bearing housings of rotodynamic pumps in field and factory test environments.

For certain vertical pump types, the vibration probes shall be located near the top of the motor support (refer to Figure 9.6.4.2.3.1, Measurement locations and directions). For vertical pump types, the reference to "bearing housing" refers to this location.

Vibration measurements can be useful for many purposes, such as acceptance tests, diagnostic or analytic investigations, and operational monitoring.

A general description of the principles to be applied for the measurement and assessment of vibration on rotodynamic pumps is given for vibration on nonrotating parts.

This standard is based on experiences from pump users and manufacturers as well as vibration measurements by many companies. Vibration data from both factory test and field test environments have been incorporated into the maximum allowable vibration values. Values are applicable when the pump is installed per Hydraulic Institute Standards or the manufacturer's specifications.

9.6.4.1.2 Scope

This standard applies to the evaluation of vibration on rotodynamic pump applications absorbing more than 2 kW (3 hp) and of the types as indicated in Figure 9.6.4.2.3.1. It pertains to evaluation of vibration when the vibration measurements are made on nonrotating parts (bearing housing vibration).

The general evaluation criteria are included for acceptance tests in field environments or at the manufacturer's test facility, as appropriate and as defined in the standard.

The following types of pumping equipment and associated equipment are **excluded** from this standard:

- Submersible pumps (refer to ANSI/HI 11.6)
- Submersible vertical turbine pumps
- Wet pit cantilever belt-driven pumps with overhung motors
- API 610 or ISO 13709 pumps (refer to API 610 or ISO 13709)
- Reciprocating engine-driven pumps
- Separately mounted drivers (i.e., vertical motors on separate floors)
- Right-angle gear drives
- Maritime applications

For these pumping equipment types, refer to other industry standards, if available, or the pump manufacturer.

Torsional vibration is not dealt with in this standard.

Information for assessing shaft vibration measured on rotating shafts is **excluded** from the scope of this document. The user is referred to the ISO 10816-7 standard for information on this topic.

This standard applies to tests conducted within the rated speed ±10%. Tests conducted at speeds exceeding these limitations (such as variable-speed pumps or 50-cycle pumps tested with 60-cycle power for factory tests) shall be mutually agreed upon by the user and manufacturer due to the possibility of objectionable resonance effects.

This standard is applicable to solids-handling pump types. A solids-handling pump is defined as a pump designed to ensure maximum freedom from clogging when handling liquids containing organic solids or stringy materials.

This standard is applicable to slurry pump types. A *slurry pump* is defined as a pump suitable for pumping a mixture of abrasive solids with specific gravity greater than 1 and concentration by volume greater than 2% in a liquid carrier, usually water. For more information on slurry pump types, refer to the latest edition of ANSI/HI 12.1–12.6 *Rotodynamic Slurry Pumps for Nomenclature, Definitions, Applications, and Operation*. If an ASME B73.1-type pump is used as a slurry pump, the acceptable vibration levels for slurry pumps as provided herein shall be used.

9.6.4.2 Bearing housing vibration measurement

9.6.4.2.1 Measurement units and procedure

For rotodynamic pumps operating at all speeds greater than 600 rpm, the measurement quantity to be used is overall velocity in millimeters per second root mean square (RMS) (inches per second RMS), measured on nonrotating parts in accordance with Section 9.6.4.2.3. Additionally, for rotodynamic pumps operating at speeds equal to or less than 600 rpm, the measurement readings of overall displacement shall be recorded in millimeters (inches) peak-to-peak, also measured on nonrotating parts in accordance with Section 9.6.4.2.3.

9.6.4.2.1.1 Speeds above 600 rpm

For speeds above 600 rpm, the sole measurement quantity to be used for measuring the vibration of nonrotating parts of rotodynamic pumps is velocity in millimeters per second RMS or inches per second RMS.

A velocity or acceleration probe is used to measure vibration at various frequencies, and the measurements are integrated in an electronic circuit as agreed upon by the user and the pump manufacturer to determine the overall RMS vibration in the appropriate units.

Overall RMS vibration is a measure of the total RMS vibration magnitude obtained using instruments that integrate the vibration within a fixed frequency range over a fixed period of time.

Measurement of vibration filtered to discrete frequencies is not applicable for acceptance testing of pumps according to this standard. Such methodologies, including complete frequency analysis, may be useful in diagnosing vibration problems, should they occur.

9.6.4.2.1.2 Speeds of 600 rpm and below

For speeds 600 rpm and below, measurement of the overall peak-to-peak displacement is required in addition to overall RMS velocity.

9.6.4.2.2 Vibration measurement instrumentation and transducers

9.6.4.2.2.1 General

For all speeds, a 6-dB per octave filter shall be used to filter out frequencies outside the measurement range to reduce the electronic noise. This filter is typically built into dedicated vibration data collection instruments. The filter may not be present when using software processing alone.

For speeds above 600 rpm, the measurement instrumentation shall be capable of measuring the RMS vibration velocity in a minimum frequency range of 5 Hz to 1000 Hz.

For pumps with speeds of 600 rpm and below, the measurement instrumentation shall be capable of measuring RMS vibration velocity and RMS peak-to-peak displacement in a minimum frequency range of 2 Hz to 1000 Hz.

The manufacturer and the purchaser should agree on the type of data collector and the data collector settings for frequency range, frequency resolution, filter settings, the number of readings to average, and any other instrument settings that may affect the measured vibration values.

9.6.4.2.2.2 Precautions

Personnel conducting vibration tests should take precautions to eliminate sources of errors in the measurements potentially caused by transducer mountings, transducer cable lengths, transducer orientation, magnetic fields, temperature variations, and sound fields.

The vibration transducers should be mounted in such a way as to not adversely affect the accuracy of the measurements. If magnetic vibration transducers are used, the surface of the measured equipment should be prepared in accordance with the transducer manufacturer's instructions at the point of contact to avoid measurement errors. Appropriate mounting methods are shown in Figure 9.6.4.2.2.2.

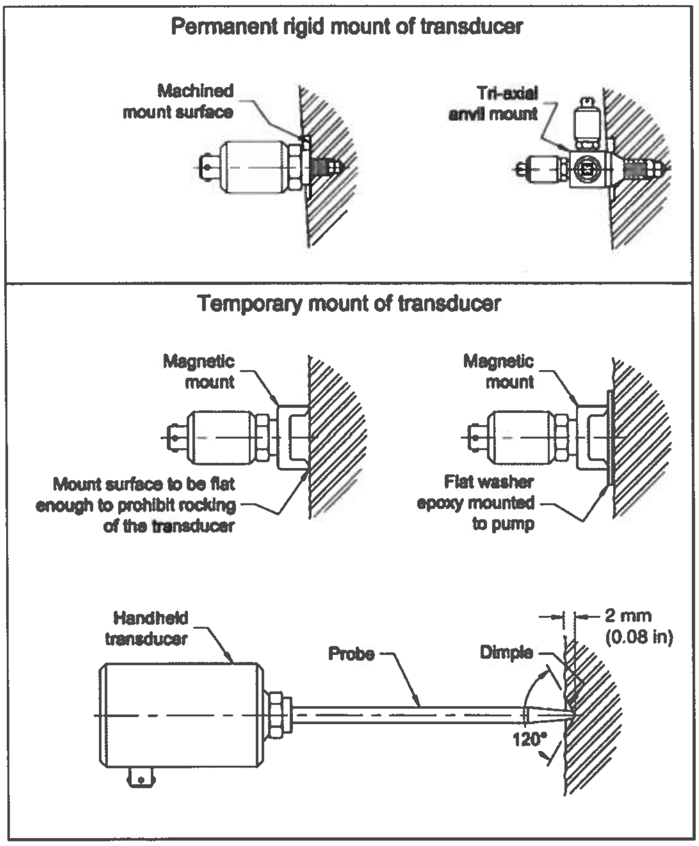


Figure 9.6.4.2.2.2 — Mounting methods for vibration transducers

The manufacturer and the purchaser should agree on the type of measurement probe to be used. In the absence of any specific recommendation from the vibration transducer manufacturer, the following should be applied:

For permanently mounted transducers, the machined mounting surface should be a minimum of 1.1 times the diameter of the transducer. Mounting surfaces should be flat to within 0.025 mm (0.001 in) with a surface roughness of less than 3.2 µm (125 microinches).

For temporary mounting of the transducer using a magnetic mount, the surface should be flat, preferably using a machined area specifically intended for the purpose, of a diameter at least equal to the diameter of the mounting surface of the probe. The paint should be removed at the point of contact to allow contact of the mount with bare metal.

For temporary mounting of the transducer using a dimple, the dimple should conform to the dimensions shown in Figure 9.6.4.2.2.2. The dimple may be formed by machining or casting. The paint should be removed at the point of contact to allow contact of the mount with bare metal. Surface roughness should be less than 3.2 µm (125 microinches).

9.6.4.2.3 Measurement locations and directions

9.6.4.2.3.1 Bearing housing measurements

In general, the vibration probes should be located approximately at the middle of the bearing housing of between-bearing pumps, near the outer bearing of end suction pumps, and near the top of the motor support for vertical wet pit pumps. Designated measurement locations for specific pump types within the scope of this standard are provided in Figure 9.6.4.2.3.1. Probes must not be located on the flexible panels, nameplates, or motor end covers.

NOTE: This ANSI/HI standard uses a method of pump identification similar to ISO 13709/API 610. It should be noted that the limits of acceptable vibration performance are different. This standard does not apply to ISO 13709 or API 610 applications. For ISO 13709 or API 610 applications, refer to those standards (see Section 9.6.4.1.2). For a more complete description of pump types, refer to ANSI/HI 1.1-1.2 *Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions* and ANSI/HI 2.1-2.2 *Rotodynamic (Vertical) Pumps for Nomenclature and Definitions*.

9.6.4.2.4 Vibration acceptance tests

Vibration acceptance tests are performed if required by the job specifications or contract. Details such as location, instrumentation, and procedures may involve variations as mutually agreed on by the parties involved. This standard provides details that shall apply if not superseded by other specification requirements.

The maximum vibration magnitude observed at the designated measurement locations (refer to Figure 9.6.4.2.3.1) is to be used in comparison to the maximum allowable vibration values indicated in the graph for the specific pump type, applicable test location (field [in situ] or factory), and pump input power at the test point considered. Note that the pump input power will be affected by the specific gravity of the fluid being pumped during the test, which will in turn determine what acceptance levels are to be used. Applicable instrumentation, installation requirements, and operating requirements for the test are provided herein.

9.6.4.2.4.1 Pump location for acceptance tests

By default, vibration acceptance tests shall be conducted in the field location. Factory vibration acceptance tests are performed if required by the job specifications or contract and with mutual agreement of the parties involved. For some pump types, factory vibration tests are regularly used for acceptance.

The acceptable vibration-level graphs indicate both field and factory vibration levels for most pump types. When factory levels are not provided for certain pump types, factory test acceptance levels for those pump types shall be mutually agreed on by the parties involved.

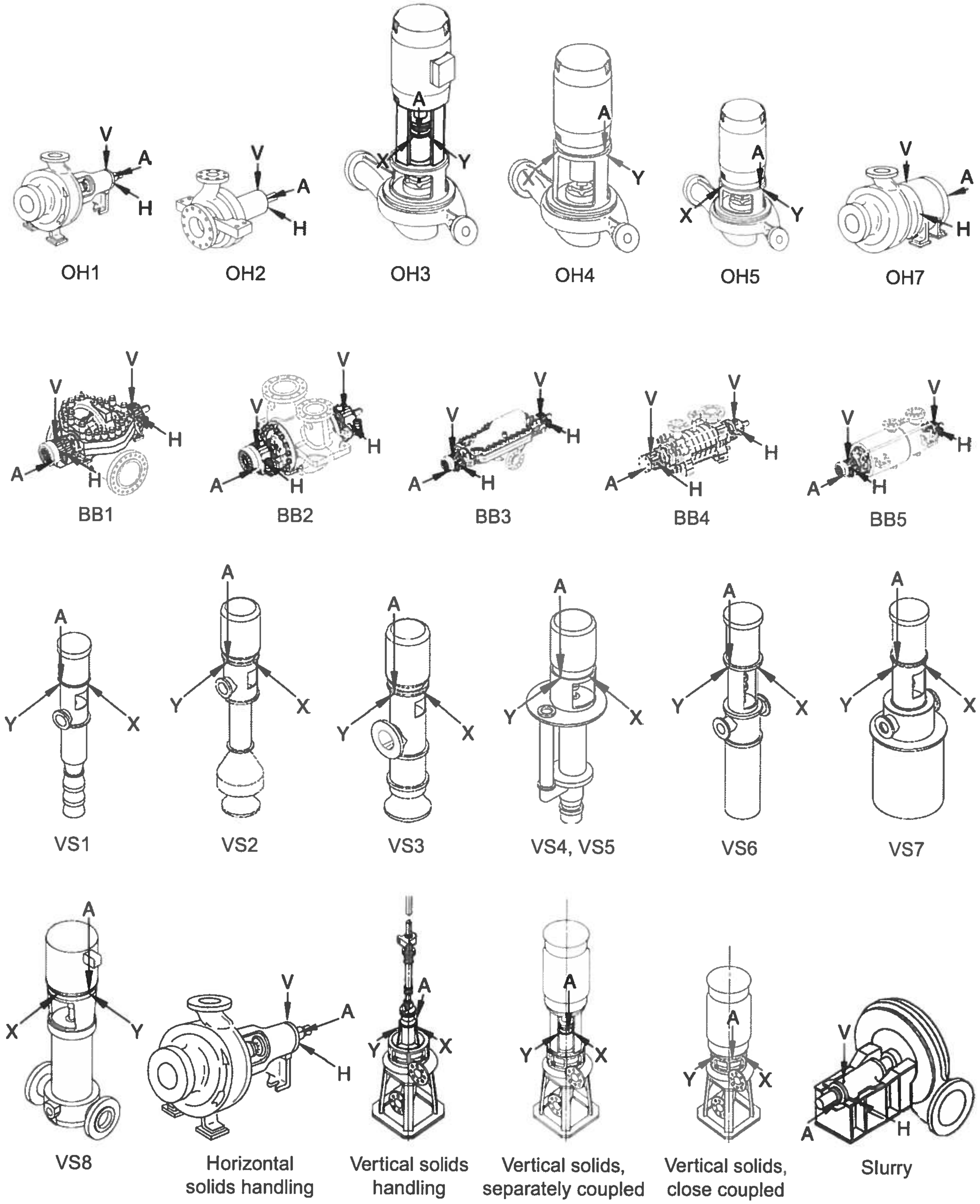


Figure 9.6.4.2.3.1 — Measurement locations and directions (For a more complete description of pump types, refer to ANSI/HI 1.1-1.2 *Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions* and ANSI/HI 2.1-2.2 *Rotodynamic (Vertical) Pumps for Nomenclature and Definitions.*)

9.6.4.2.4.2 Factory acceptance test

A factory test, if specified, is performed in the manufacturer's test facility or a facility mutually agreed on by the parties involved, using a temporary test setup that may not be equivalent to the permanent conditions in the field installation. The factory test may also possibly necessitate using a different fluid with different properties (i.e., specific gravity, viscosity) that can affect vibration and can also result in different acceptance levels because of the resulting different pump input power values. Higher vibration is to be expected as compared to the field installation conditions.

If the values at the factory test facility do not fulfill the requirements of this standard, the manufacturer shall investigate to clarify the reason for the deviation and resolve the matter as agreed by mutual consent.

9.6.4.2.4.3 Field acceptance test

The field acceptance test conditions apply to pumps that are installed on-site according to the latest standards of the Hydraulic Institute and the manufacturer's instructions.

9.6.4.2.4.4 Acceptance test installation and operating conditions

The following general conditions apply to the use of this standard (factory tests and field tests):

- Pump impellers shall be balanced (typically a single-plane spin balance) in accordance to ISO 1940 balance quality grade G6.3 (see Figures B.1 and B.2). For balance of slurry pump types, refer to the latest edition of ANSI/HI 12.1–12.6. When the ratio of the largest outside diameter of the component divided by the impeller width at the outside of the shroud(s) is less than 6 (see Figures B.3 and B.4), a two-plane balance may be required (refer to ISO 1940). Other grades may be used as agreed on by the user and manufacturer. Note: In the specific case of impellers, the width is measured at the periphery, including the thickness of any shrouds, but not the back vane.

- Operation shall be under steady state conditions at the rated speed ±10% (applies to field tests). This requirement may not always be practicably achievable for factory tests (i.e., 50-cycle pumps tested with 60-cycle power), therefore the operating speed for factory tests shall be mutually agreed on by the user and manufacturer.

- The coupling alignment shall be in accordance with the pump manufacturer's recommendations.

- The bearing housing vibration level recorded shall be the maximum of measurements taken in each plane as indicated by the illustrations in Figure 9.6.4.2.3.1.

- To the fullest extent practicable, no other equipment should be operated during the test in the vicinity of the pump being tested; otherwise, the effects of the other equipment on the vibration test should be considered.

The following conditions apply to the use of this standard with regard to field tests:

- Field measurements on pump bearing housings with antifriction or fluid film bearings shall be performed when the bearing housings have reached their normal steady state operating temperatures.

  NOTES:
  This requirement does not apply to sleeve bearings used on vertically suspended pumps, because such bearings are normally inaccessible for temperature monitoring.

  Additionally (unless otherwise agreed) this requirement does not apply during factory testing because the time for bearing temperature stabilization may be excessive.

- The field-tested pump shall be operated at the specified or rated operating conditions (flow rate, head, and speed). Acceptable vibration levels for specified operating conditions within the preferred operating region (POR) are provided in graphs herein. Values to use for specifed operating conditions outside the POR but within the allowable operating region (AOR) are indicated on the graphs for the specific pump type. For more information on operation within the pump POR and AOR, refer to the latest edition of ANSI/HI 9.6.3 *Rotodyanmic Pumps for Allowable Operating Region.*

- Pump shall be installed so that nozzle loads do not exceed the loads within ANSI/HI 9.6.2 *Rotodynamic (Centrifugal and Vertical) Pumps for Allowable Nozzle Loads* (if applicable) or the manufacturer's recommendations. When there is a difference between the values of ANSI/HI and the manufacturer, the values of the manufacturer shall be used unless otherwise agreed on by the parties involved (applies to field acceptance tests).

- Prior to a field test it shall be established that the vibration measured with the pump not running does not exceed 25% of the vibration level established for acceptable operation. In any situation where this limit is exceeded, all necessary steps shall be taken to find and eliminate the source of this vibration, otherwise limits for acceptable vibration must be mutually agreed on by the manufacturer and the user.

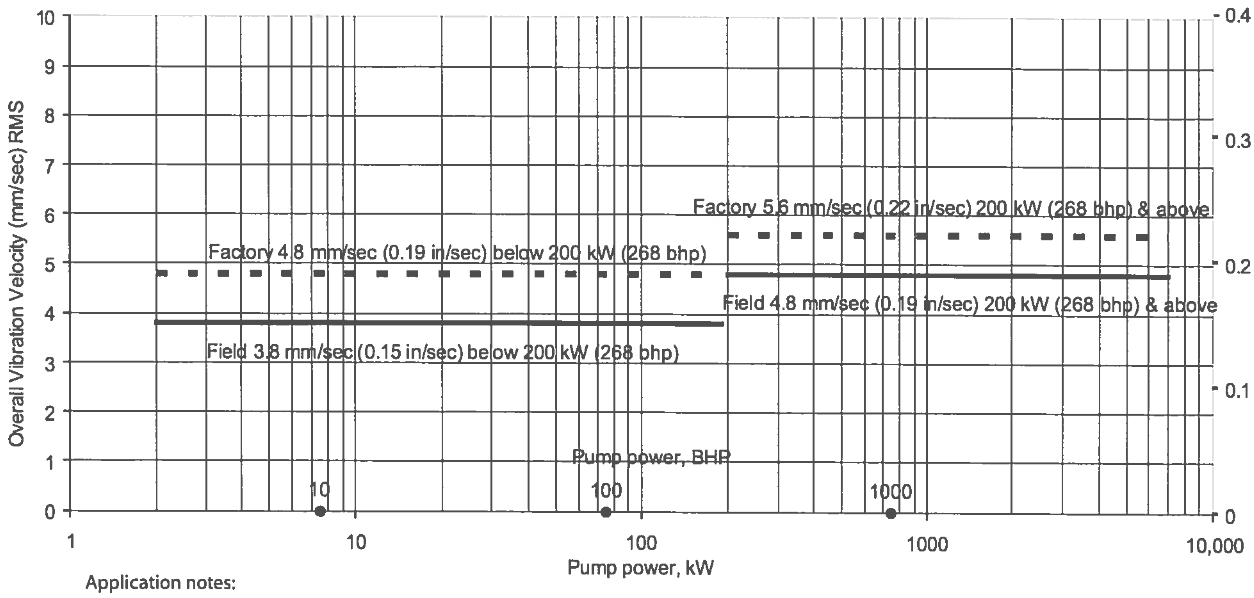
- The field test conditions shall represent actual in-service operation as closely as practicable. Site conditions such as temporary flow loops, enhanced inlet pressure, special test setups for hardware, and other abnormal operating configurations may have a significant effect on vibration and are to be avoided or otherwise agreed on by the manufacturer and the user.

- There shall be no suction-related adverse hydraulic phenomena (submerged vortices, free-surface vortices, preswirl, nonuniform velocity distributions at the impeller eye, entrained gas, cavitation, or flow field variations with time) present to an extent significant enough to adversely influence the vibration measurements. For best practice, refer to the latest edition of ANSI/HI 9.8 *Pump Intake Design* or ANSI/HI 9.6.6 *Rotodynamic Pumps for Pump Piping*, as applicable.

9.6.4.2.5 Allowable pump bearing housing vibration

9.6.4.2.5.1 Allowable pump bearing housing vibration (antifriction bearings), speeds above 600 rpm

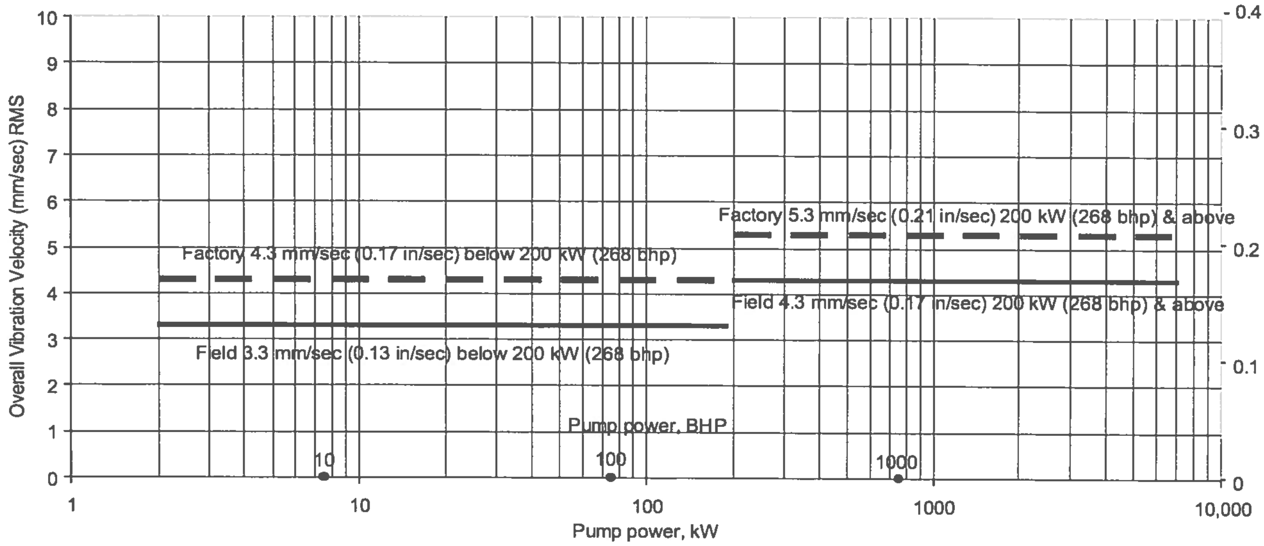
The vibration values shown in the following graphs are for RMS velocity readings. When using the figures, the abscissa refers to the power the pump is drawing at the time the vibration measurement is made.



Application notes:

1) New pumps, operation in the preferred operating region (POR), tested with clear liquids. (Maximum, any plane. For measurement locations indicated in Figure 9.6.4.2.3.1.)
2) For operation outside the POR but within the allowable operating region (AOR), increase the values shown by 30%.
3) Pump types refer to Figure 9.6.4.2.3.1.

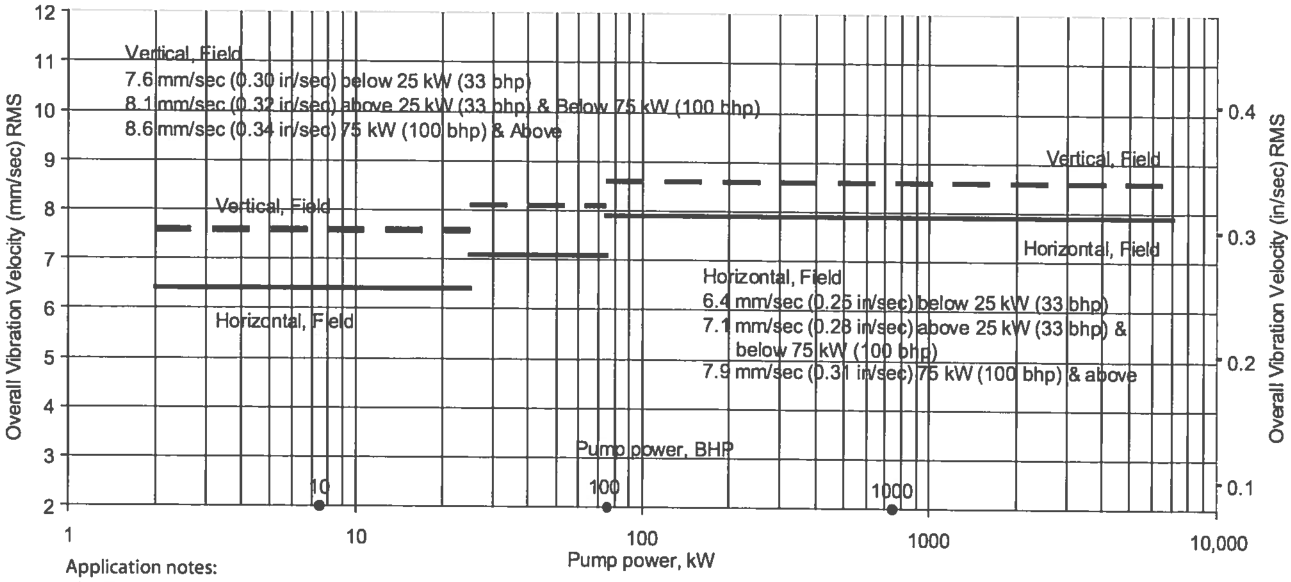
Figure 9.6.4.2.5.1a — Allowable pump vibration, pump types BB1, BB2, BB3, BB4, BB5, OH1, OH2, OH3, OH4, OH5, and OH7 (For a more complete description of pump types, refer to ANSI/HI 1.1-1.2 *Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions* and ANSI/HI 2.1-2.2 *Rotodynamic (Vertical) Pumps for Nomenclature and Definitions.*)



Application notes:

1) New pumps, operation in the preferred operating region (POR), tested with clear liquids. (Maximum, any plane. For measurement locations indicated in Figure 9.6.4.2.3.1.)
2) For operation outside the POR but within the allowable operating region (AOR), increase the values shown by 30%.
3) For solids-handling versions of pump types VS4, VS5, and VS8, increase the AOR and POR values (as applicable) by 50% (tests are to be done with clear liquids).
4) Pump types refer to Figure 9.6.4.2.3.1.

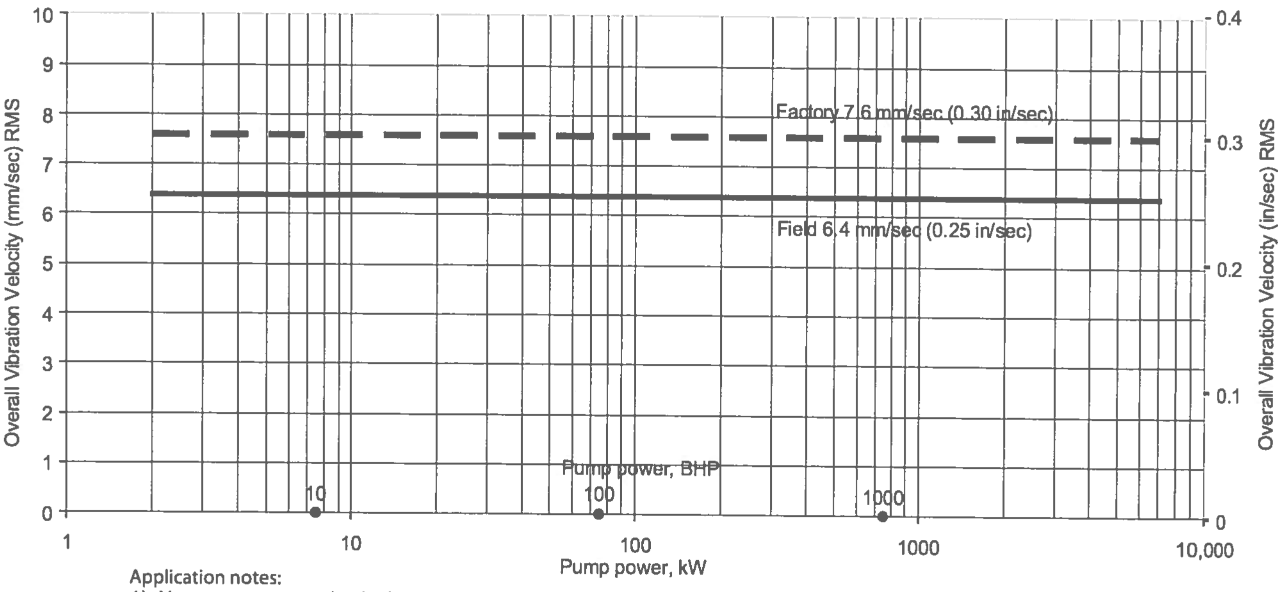
Figure 9.6.4.2.5.1b — Allowable pump vibration, pump types VS1, VS2, VS3, VS4, VS5, VS6, VS7, and VS8 (For a more complete description of pump types, refer to ANSI/HI 1.1-1.2 *Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions* and ANSI/HI 2.1-2.2 *Rotodynamic (Vertical) Pumps for Nomenclature and Definitions.*)



Application notes:

1) Field tests (factory tests not applicable), new pumps, operation in the preferred operating region (POR) tested with clear liquids. (Maximum, any plane. For measurement locations indicated in Figure 9.6.4.2.3.1.)
2) For operation outside the POR but within the allowable operating region (AOR), increase the values shown by 30%.
3) Pump types refer to Figure 9.6.4.2.3.1.

Figure 9.6.4.2.5.1c — Allowable pump vibration, solids-handling pump types



Application notes:

1) New pumps, operation in the preferred operating region (POR), tested with clear liquids. (Maximum, any plane. For measurement locations indicated in Figure 9.6.4.2.3.1.)
2) For operation outside the POR but within the allowable operating region (AOR), increase the values shown by 30%.
3) Pump types refer to Figure 9.6.4.2.3.1.

Figure 9.6.4.2.5.1d — Allowable pump vibration, slurry pump types

9.6.4.2.5.2 Additional acceptance criteria, allowable pump bearing housing vibration, speeds of 600 rpm and below.

Table 9.6.4.2.5.2a — Additional acceptance criteria for pumps operating at 600 rpm and below[1]

Operating Conditions & Application Data	Field	Factory
Operation within the POR	80 μm (3.0 mils), peak to peak	100 μm (4.0 mils), peak to peak
Operation within the AOR	100 μm (4.0 mils), peak to peak	125 μm (5.0 mils), peak to peak

[1] Adjust these values using the applicable additive values in Table 9.6.4.2.5.2b.

Table 9.6.4.2.5.2b — Additive values to Table 9.6.4.2.5.2a

Application Data	Field	Factory
Additive value per meter of height when the measurement location per Figure 9.6.4.2.3.1 is greater than 1.5 m (5 ft) above foundation. (Adjust for fractions of a meter.)	50 μm (2.0 mils), peak to peak	
Additive value for slurry pumps	100 μm (4.0 mils), peak to peak	
Additive value for solids-handling wet pit pumps	50 μm (2.0 mils), peak to peak	
Additive value for solids-handling pumps	50 μm (2.0 mils), peak to peak	Not applicable

Appendix A

Test Report

This appendix is not part of the standard, and is presented for informative purposes only.

A suggested test report form is provided on the following page.

Vibration Test Report

Subject Pump	
Manufacturer	
Pump Type Designation (Ref. Fig. 9.6.4.2.3.1)	
Model	
Size & Type	
Serial Number	
No. of Stages	
Driver Size & Description	
Speed (Rated)	
Flow Rate (Rated)	
Total Head (Rated)	
Power (Rated)	
Pumped Liquid	
Specific Gravity	
Viscosity	
Temperature	

Test Conditions	
Test Location	
Test Speed	
Flow Rate	
Discharge Head	
Suction Head	
Pumped Liquid	
Liquid Specific Gravity	
Viscosity	
Liquid Temperature	
Inboard Bearing Housing Temperature	
Outboard Bearing Housing Temperature	
Ambient Temperature	
Power	

Instrumentation			
Description	Model	Serial	Date Calibrated

Data					
Pump Shaft Orientation	Measurement Location Description (Ref. Figure 9.6.4.2.3.1)	Direction (Ref. Figure 9.6.4.2.3.1)	Measured Values (Units =___)	Maximum Allowable	Pass/Fail
Vertical		X			
		Y			
		A			
Horizontal		H			
		V			
		A			
		H			
		V			

Tested By: ______________________________ Date: ______________________

Reported By: ____________________________ Date: ______________________

Appendix B

Factors Affecting Vibration

This appendix is not part of the standard, and is presented for informative purposes only.

B.1 Residual mechanical unbalance of rotating parts

High levels of residual unbalance in rotating parts can generate high unbalance forces resulting in excessive bearing and shaft loading and inducing high levels of vibration. Balancing methods and residual unbalance limits for impellers are described in this section.

NOTE: Other rotating parts may be subject to similar limits. However, drive system components such as motors and couplings that are addressed by other standards may need other considerations.

Pump impellers are typically balanced in accordance with ISO 1940 balance quality grade G6.3 or better (see Figures B.1 and B.2). These figures indicate the center of gravity displacement or residual unbalance that is acceptable for balance grade 6.3. (See ISO 1940 for values relating to other balance grades.)

For balance of slurry pump type impellers, refer to the latest edition of ANSI/HI 12.1-12.6 *Rotodynamic (Centrifugal) Slurry Pumps for Nomenclature, Definitions, Applications, and Operation.*

Depending on component geometry, it may be satisfactory to perform a single-plane spin balance. Components are typically single-plane balanced if the ratio of diameter to width *D/b* is 6.0 or greater (see Figures B.3 and B.4). Two-plane (or dynamic) balancing is typically performed otherwise.

Figures B.1 and B.2 are used by entering the graph at the maximum expected service speed, such as 3000 rpm, and reading the acceptable residual unbalance as 0.021 kg–mm/kg (0.85 oz–in/oz). Multiply this number by the impeller weight in kg (oz) and the result is the allowable unbalance of the impeller in kg–mm (oz–in).

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

Balancing machines are capable of measuring unbalance independent of its speed. When the value for allowable unbalance is determined from Figure B.1 and B.2, it is not necessary to operate the balancing machine at the same speed as the pump speed.

Balance machines shall be calibrated as recommended by their manufacturer. When specified, calibration shall be done just prior to balancing.

Pertinent aspects:

- The practice of component balancing is appropriate for a large proportion of rotodynamic pump types that can be proven to meet the specified vibration performance criteria while using clearance fits between the rotating component parts and the shaft. Residual unbalance grades are then determined to meet vibration performance acceptance levels while also considering the mass eccentricity effects caused by clearance fits and the resulting component runout. Clearance fits are preferred and used whenever possible to facilitate pump assembly and disassembly.

- There are, however, many pump types and applications where it is necessary to use shrink fits and perform a supplementary two-plane (dynamic) balance on the complete rotating assembly to meet the specified vibration performance criteria. In these instances, the manufacturer and the purchaser should agree on the appropriate residual unbalance grade.

– Balance machines have a weight rating pertaining to the maximum weight of a part to be balanced on the machine. Balance-machine sensitivity is a function of the weight of the part to the weight rating of the machine. (Above 100%, one must check with the manufacturer of the balancing machine.) As an example, a 45-kg (100-lb) rated machine may provide adequate sensitivity and accuracy for a 4.5-kg (10-lb) part, but a 9-kg (20-lb) rated machine would be much more suited for the task; and a 1.5-kg (3-lb) part may not balance at all on the 45-kg (100-lb) machine to the quality grade required.

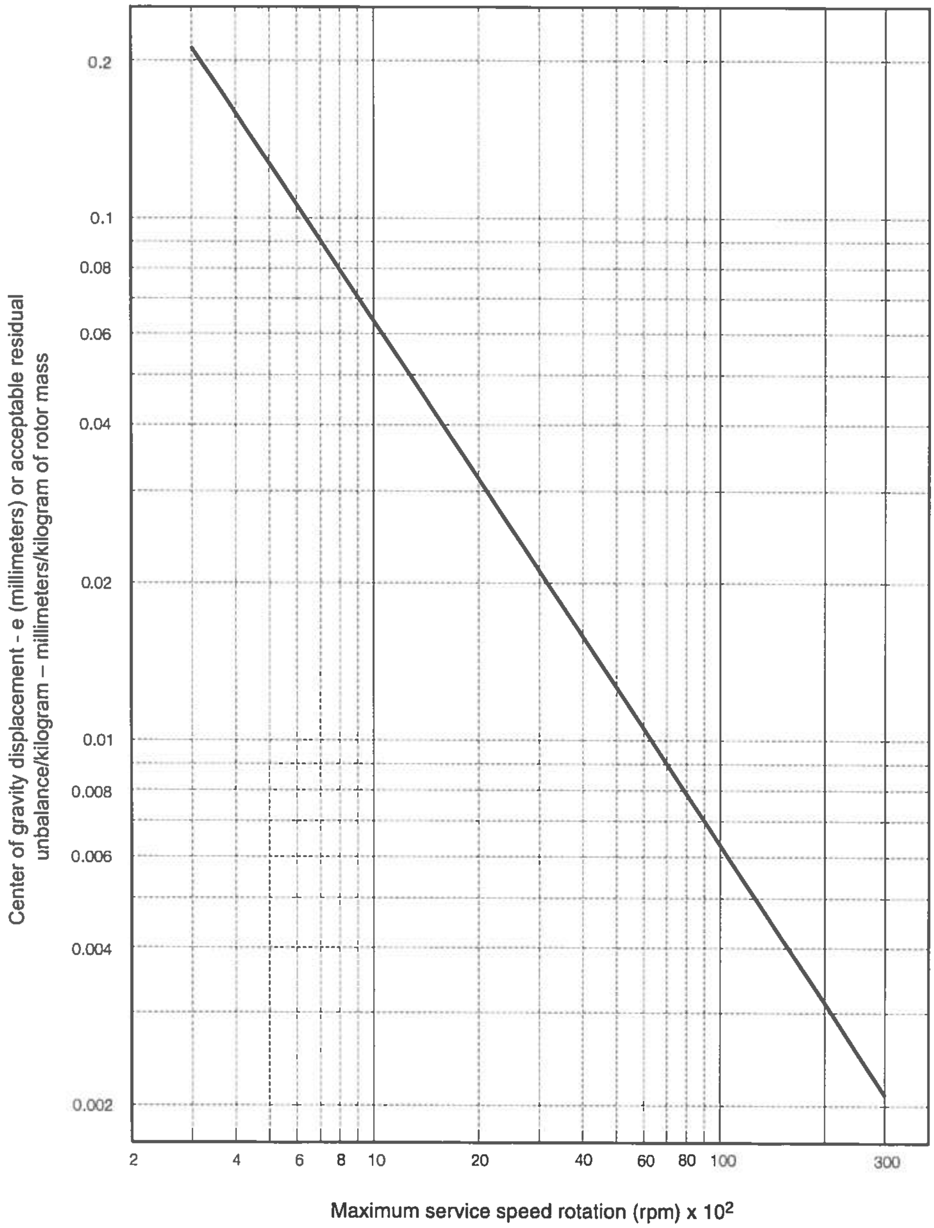


Figure B.1 — Center of gravity displacement or acceptable residual unbalance (balance grade G6.3) – metric units (Ref. ISO 1940)

Following are guidelines for the quality of balance procedure, equipment, tooling, and impeller geometry, giving both users and manufacturers a common ground for discussing these issues that have been learned through experience.

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

   The balancer drive may be checked by periodically rotating the drive splines 180 degrees after a part has been balanced and checking the residual unbalance. It should be within 10% of the original unbalance of the part being balanced. Runout in the balancing arbor should be checked when assembled in the balancer. It should be no more than 0.025 mm (0.001 in) total indicator movement.

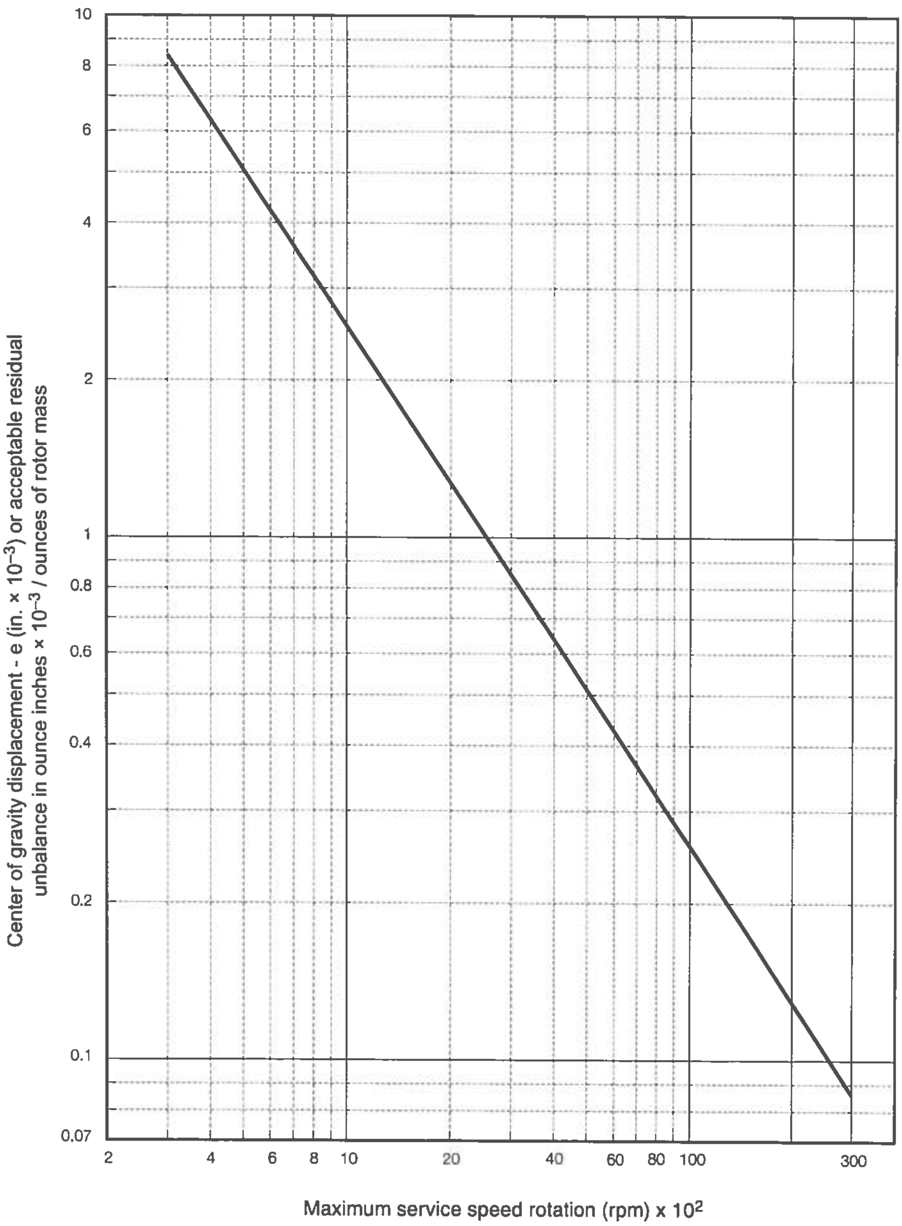


Figure B.2 — Center of gravity displacement or acceptable residual unbalance (balance grade G6.3) – US customary units (Ref. ISO 1940)

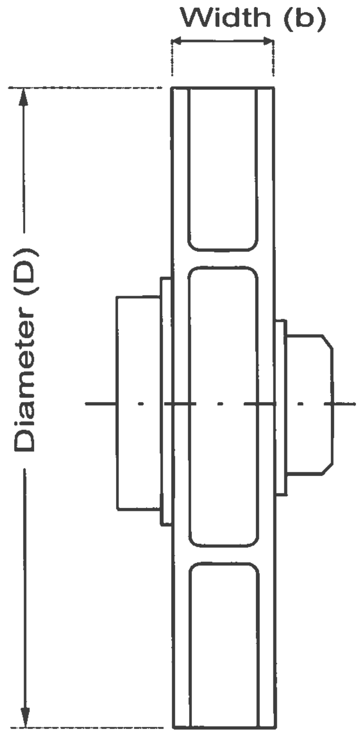


Figure B.3 — Single suction impeller

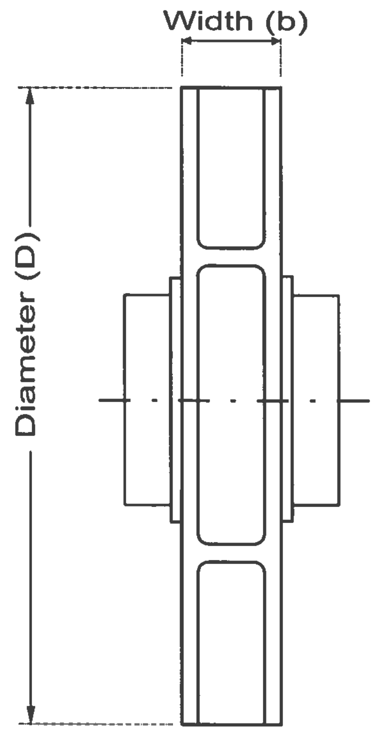


Figure B.4 — Double suction impeller

2) Keys/keyway geometry errors.

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

3) Excessive looseness between impeller hub and balancing arbor.

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

Impeller hub bore diameter	Maximum looseness (diametral)	
	1800 rpm	1800 to ≤ 3600 rpm
0 — 38 mm (1.499 in)	0.038 mm (0.0015 in)	0.038 mm (0.0015 in)
38.1 mm (1.5 in) — 50.8 mm (1.999 in)	0.051 mm (0.0020 in)	0.038 mm (0.0015 in)
50.9 mm (2.0 in) and above	0.065 mm (0.0025 in)	0.038 mm (0.0015 in)

4) Removal or addition of material.

   Material removal: This should be done in a manner that spreads the balance correction as evenly as possible over the surface. If a shroud is used, the thickness removed should be no more than one third of the original, and the subsequent finish should not have a greater RMS value than that of the remainder of the shroud. If the impeller vane is used for balance correction, no more than one quarter of the vane thickness should be removed, always from the low-pressure side. Removal by drilling and/or end milling should follow the same thickness guidelines, with appropriate consideration to minimizing flow discontinuities.

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

B.2 Residual mechanical unbalance of rotating parts - abrasive fluids

Pumps, especially those involving abrasive fluids (slurry and solids-handling pumps) can undergo a change in the residual balance due to wear. This will affect the pump vibration performance.

B.3 Pump and driver natural frequency and resonance

A resonant condition can result when the operating speed or vane-pass frequency of the pump excites a natural frequency of the structure. At resonance, the vibration levels will be amplified depending on the amount of damping present in the system.

Often much attention is given to the value of the natural frequency of the pump assembly; however, in a field installation, the structure comprises in addition to the pump, the foundation, the mounting, the piping and its supports, and may include the driver and coupling or driveshaft(s), as applicable. The natural frequency of the total structure may therefore differ significantly from the natural frequency of the pump alone.

The system designer should give this consideration to ensure that there is an appropriate separation margin between the natural frequency of the system and the operating speed. For systems that operate over a speed range, natural frequencies may fall within the operating range and critical speeds may be identified. In such cases, due care and consideration must be taken to evaluate the possibilities of avoiding such operating speeds or mitigating the possible effects on bearing housing (or other) structural resonances.

B.4 Miscellaneous mechanical problems

A common cause of pump vibration is misalignment of the pump and driver shaft. Shafts can have either parallel or angular misalignment, or both. It is not uncommon for a pump and driver shafts to exhibit both types of misalignment.

Damaged bearings, bent shafts, inadequate piping supports, and expansion joints without properly applied tie rods are also frequent causes of vibration. Piping supports that are adequate for empty pipes may not be adequate for pipes filled with the pumped liquid and the adequacy of the piping supports may vary with the pumped liquid.

Vibration may also be caused by inadequate practices pertaining to bed plates, such as improper grouting, inadequate tie-down bolts, inadequate bed material, inadequate bed rigidity, and the lack of proper internal locking of the bed plate to the foundation.

B.5 Hydraulic disturbances

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

a) Hydraulic forces produced between the impeller vanes and the volute cutwater or diffuser at vane-passing frequency.

b) Recirculation and radial forces at low flows.

c) Fluid separation at high flows.

d) Cavitation.

e) Flow disturbances in the pump intake due to improper intake design.

f) Air entrainment or aeration of the liquid.

g) Hydraulic resonance in the piping.

h) Solids contained in the fluids, such as sewage, impacting the pump and causing momentary mechanical unbalance or wedged in the impeller and causing continuous mechanical imbalance.

i) Hydraulic imbalance caused by the impeller vanes not spaced evenly enough.

B.6 Hydraulic resonance in piping

Vibration problems can be caused by hydraulic resonance of the liquid within the pump/piping system. Hydraulic resonance occurs when a piping system has a hydraulic resonant frequency that is excited by forces induced by operation of a pump. When normal pump-induced pressure pulsations are reflected by the piping systems and added in phase to the source pulse, the amplitude of the pulse is magnified. Resulting high pressures can ultimately cause mechanical fatigue failures in either the piping or the pump components, generate unacceptable noise levels, and cause vibration of the system components.

NOTE: A certain level of pressure fluctuation is unavoidable and has no detrimental effect. Excessive pressure pulsations, however, can excite pump and pipe vibrations and might even cause damage. Some typical problems are

- Bearing housing vibrations (possibly resonances) sometimes causing fatigue fracture of instrumentation or auxiliary pipes.
- Baseplate vibrations. Multiplying the pressure fluctuations by the cross sections of the suction and the discharge pipes provides a rough estimate of the excitation forces at work.
- Fatigue fracture of tie bolts in multistage segmental pumps or joining elements in the stage casings of barrel pumps.
- The pressure pulsations generated by the pump can give rise to standing waves (acoustic resonances) in the system that may upset the control system, induce pipe vibrations, cause breakage of instrument lines, or get into resonances with other components.
- Finally, pressure pulsations are also one of the most important sources for noise emissions from the pump; airborne noise is radiated from the casing, the piping, and the baseplates, and solidborne noise is radiated into the piping and the foundation.

Experience has shown that the following measures may prove effective in correcting hydraulic resonance:

a) Alter the resonant piping.

b) Change the pump speed.

c) Change the internal design characteristics of the pump.

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

B.7 Effect of rigidity

The amplitude of the vibration resulting from a given driving force is related to the rigidity of the vibrating structure. Structures that have lower centers of gravity tend to be more rigid.

For example, a low or short pump structure is more rigid than a high or tall pump structure. Identical vibration forces could be expected to result in higher vibration amplitude levels on a high or tall pump structure than on a low or short pump structure. In the same way, identical vibration forces would result in lower vibration amplitude levels on a low or short pump structure than on a high or tall pump structure. Users are encouraged to consult with the pump manufacturer for more specific recommendations

Appendix C

Recommended Default Initial Field Alarm and Trip Settings

This appendix is not part of the standard, and is presented for informative purposes only.

Users are encouraged to consult with the pump manufacturer for any specific initial field alarm and trip settings.

In the absence of site-specific requirements for field instrumentation alarm and trip settings, the following default initial settings are recommended.

Alarm – One and a half times the allowable field vibration

Trip – Twice the allowable field vibration

An appropriate time delay should be used to account for transient conditions that would otherwise cause nuisance tripping. The time delay value used should consider such factors as the time to open a valve to obtain steady state operating conditions.

For an alarm or trip event occurance, it is recommended that an evaluation be made of the event to determine if corrective action is needed.

For certain pump applications (such as slurry- and solids-handling pumps), wear occurs in the normal course of operation that will affect the vibration levels obtained.

As vibration trends for a particular pump are established, the site-specific alarm and trip values may be adjusted accordingly. For a further discussion of this topic, refer to ANSI/HI 9.6.5 *Rotodynamic (Centrifugal and Vertical) Pumps for Condition Monitoring.*

Appendix D

Vibration Source Identification Chart

This appendix is not part of the standard, and is presented for informative purposes only.

High Pump Vibration Source Identification			
Symptom(s)	**Frequency (CPM)**	**Possible Cause**	**Comments**
Radial plane vibration, proportional to unbalance and/or speed	1 × RPM	**Imbalance** • Impeller imbalance • Clogging • Weak foundation • Bad pipe support	Common source of vibration
Vibrates at one speed only	1 × RPM	**Mechanical Resonance** • Motor imbalance • Impeller imbalance • Pump design • Weak foundation • Bad pipe support	Confirm by bump test Natural frequency at run speed
	N × RPM		Confirm by bump test Natural frequency at blade-pass frequency N = Blade-pass frequency
	N × RPM	Acoustic Resonance	Confirm by waveform testing N = Blade-pass frequency
	N × RPM	Acoustic Resonance	Use pressure transducers to measure fluid pressure pulsations in the piping N = Blade-pass frequency
Axial component of vibration is +50% of radial levels	1 × RPM V × RPM	Vortexing Intake	Observe intake flow for stability V = number of impeller vanes
	1 × RPM 2 × RPM	Coupling Misalignment Bent Shaft	Confirm with dial indicators to document shaft runout
Erratic vibration	High > 6 × RPM	Bad Antifriction Bearings	Use velocity to measure Listen at bearing housings
Vibration stops instant power is shut off	1 × RPM 1 or 2 × Synch speed	Electrical	Bad motor, power source, or variable-frequency drive

Appendix E

Vibration Transducer Mounting and Sensitivity

This appendix is not part of the standard, and is presented for informative purposes only.

The type of mounting used for the vibration transducer has a significant effect on the frequency response of the transducer. Figure E.1 shows the typical response for the three types of mountings covered by Figure 9.6.4.2.2.2 of this standard:

- Permanent rigid mount (using a stud)
- Magnetic mount (temporary)
- Handheld transducer (temporary)

Other types of mountings not covered explicitly in Figure 9.6.4.2.2.2 of the standard include:

- Permanent rigid mount using welding
- Permanent rigid mount using adhesive

The permanent rigid mount (using welding or stud mounting) has the widest frequency range with a flat transducer response. This arrangement is recommended for any installation requiring frequency measurement above 1 kHz.

Adhesive mounting is also an acceptable attachment method but not as good as stud mounting. The manufacturer's installation guidelines must be followed regarding the type and application of the adhesive. If excess adhe-

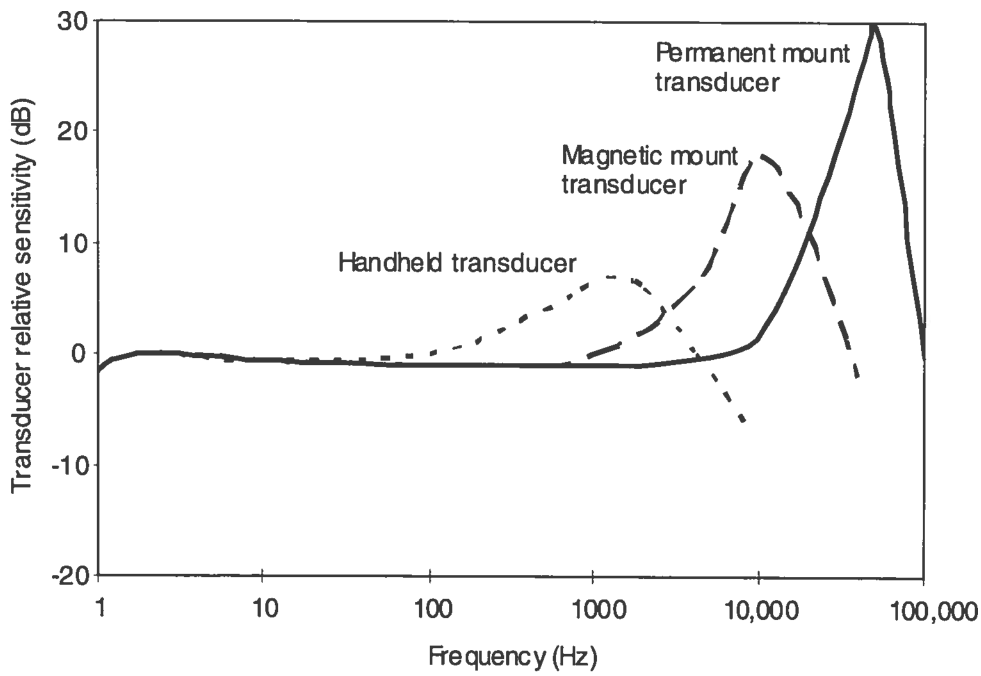


Figure E.1 — Typical response of three mounting types

sive or the wrong type of adhesive is used, there is a risk of excessive vibration dampening in the adhesive layer or failure of the adhesive mount.

The frequency range of both magnetic and handheld transducer mounting methods is greatly reduced when compared to permanent rigid mounting. Magnetic mount transducers have a better response than handheld. However, this assumes that the mounting surface is sufficiently flat. Magnetically mounted transducers can produce consistent results if any paint is removed under the magnet and the surface is flat or the magnets can be mounted firmly to avoid any rocking of the mount.

Handheld probes typically provide the poorest reading quality because the readings are affected by the probe length and the pressure applied by the person holding the probe. To prevent handheld probes from having a structural resonance in the frequency range of interest, they should be made of steel and should not be more than 150 mm (6 in) long.

Appendix F

Vibration Transducer Types

This appendix is not part of the standard, and is presented for informative purposes only.

There are three basic types of vibration sensors commonly used.

A proximity probe. This might be more properly called a *noncontact displacement probe* that measures the relative distance between the usually fixed probe and the rotating part. This measures the displacement in micrometers (microinches).

A velocity probe. This probe converts the velocity of the movement to an electrical output that is proportional to the velocity. It is usually mounted on the bearing housing or driver pedestal.

An accelerometer probe. This probe converts the acceleration of the movement into a proportional electrical signal. It is usually mounted on the bearing housing or driver pedestal.

For most acceptance and diagnostic testing situations, accelerometer type probes are used. These are commonly based on piezoelectric transducers. The piezoelectric transducer generates an electric voltage by the inertial movement of a small mass against the piezoelectric material in response to the vibration of the machinery that the transducer is attached. The transducer also has a signal amplifier to allow transmission over a length of cable to the data collector or analyzer.

Accelerometer transducers come in a variety of frequency ranges and sensitivities, and it is important to select the correct combination for the testing to be performed. The frequency range of an accelerometer is determined by the mass and piezoelectric material selected as well as the resulting natural frequency of the device. Usually, transducer frequency ranges are classified as low frequency, general purpose, and high frequency.

For equipment operating at or below 600 rpm, a low-frequency accelerometer is recommended. These transducers can detect vibration down to about 6 cycles per minute (CPM) (0.1 Hz).

For most equipment, general purpose transducers may be used. General purpose transducers have a useful frequency range of about 180 to 600,000 CPM (3 to 10,000 Hz).

High-frequency transducers are required for applications where the generated frequencies of interest are more than a few orders of magnitude above the rotating element speed. High-frequency transducers have a useful vibration range of 600 to 3,600,000 CPM (10 to 60,000 Hz). Gear mesh wear vibration frequencies on a high-speed centrifugal machine typically require a high-frequency transducer to accurately measure them.

Transducers also have differing sensitivities to the measured acceleration. Sensitivity is normally given in millivolts output signal per g of acceleration (mV/g), where g = 9.806 m/s^2 (32.174 ft/s^2).

For low-frequency transducers, a higher sensitivity is desirable to be sure the measured signal is not lost in low-frequency noise or radio frequency interference. Sensitivities for low-frequency transducers should be 500 mV/g and above.

General purpose transducer sensitivities should be in the 10 to 100 mV/g range.

High-frequency transducers should have a sensitivity range of between 0.4 and 10 mV/g.

In general, a higher sensitivity is desirable. However, higher sensitivity makes the accelerometer susceptible to damage. For example, a highly sensitive low-frequency accelerometer can easily be permanently damaged by a slight drop or even rough handling. Therefore the user should select a transducer with the highest sensitivity and the appropriate level of physical protection for the application.

Appendix G

Index

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

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

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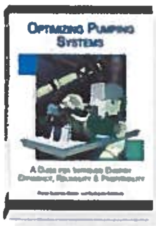

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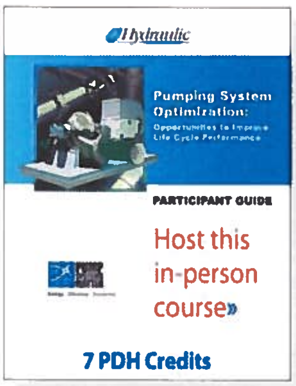

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9.6.4.2.5.2 Additional acceptance criteria, allowable pump bearing housing vibration, speeds of 600 rpm and below.

Table 9.6.4.2.5.2a — Additional acceptance criteria for pumps operating at 600 rpm and below[1]

Operating Conditions & Application Data	Field	Factory
Operation within the POR	80 µm (3.0 mils), peak to peak	100 µm (4.0 mils), peak to peak
Operation within the AOR	100 µm (4.0 mils), peak to peak	125 µm (5.0 mils), peak to peak

[1] Adjust these values using the applicable additive values in Table 9.6.4.2.5.2b.

Table 9.6.4.2.5.2b — Additive values to Table 9.6.4.2.5.2a

Application Data	Field	Factory
Additive value per meter of height when the measurement location per Figure 9.6.4.2.3.1 is greater than 1.5 m (5 ft) above foundation. (Adjust for fractions of a meter.)	50 µm (2.0 mils), peak to peak	
Additive value for slurry pumps	100 µm (4.0 mils), peak to peak	
Additive value for solids-handling wet pit pumps	50 µm (2.0 mils), peak to peak	
Additive value for solids-handling pumps	50 µm (2.0 mils), peak to peak	Not applicable