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Mechanical vibration — Measurement and evaluation of machine vibration —

Part 1:
General guidelines

Vibrations mécaniques — Mesurage et évaluation des vibrations de machines —

Partie 1: Lignes directrices générales

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

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For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see the following URL: www.iso.org/iso/foreword.html.

The committee responsible for this document is ISO/TC 108, *Mechanical vibration, shock and condition monitoring*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

This first edition of ISO 20816-1 cancels and replaces ISO 7919-1:1996, ISO 10816-1:1995 and ISO 10816-1:1995/Amd 1:2009 which have been merged and editorially revised.

A list of all parts in the ISO 20816 series can be found on the ISO website.

Introduction

Machines are now being operated at increasingly high rotational speeds and loads, as well as more flexible operation at part and full load, and under increasingly severe operating conditions. This has become possible, to a large extent, by the more efficient use of materials, although this has sometimes resulted in there being less margin for design and application errors.

At present, it is not uncommon for continuous operation to be expected and required for 2 years or 3 years between maintenance operations. Consequently, more restrictive requirements are being specified for operating vibration values of rotating machinery, in order to ensure continued safe and reliable operation.

This document is a basic document which establishes general guidelines for the measurement and evaluation of mechanical vibration of machinery, as measured on rotating and on non-rotating (and, where applicable, non-reciprocating) parts of complete machines, such as shafts or bearing housings. Recommendations for measurements and evaluation criteria pertaining to specific machine types are provided in additional parts of ISO 20816 as they become available as a replacement of the relevant parts of ISO 7919 and ISO 10816. ISO/TR 19201 gives an overview over these and further machinery vibration standards.

For some machines, measurements made on non-rotating parts are sufficient to characterize adequately their running conditions with respect to trouble-free operation. There are also types of machine, such as steam turbines, gas turbines and turbo compressors, all of which can have several modes of vibration in the service speed range, for which measurements on structural members, such as the bearing housings, might not adequately characterize the running condition of the machine, although such measurements are useful. Such machines generally contain flexible rotor shaft systems, and changes in the vibration condition can be detected more decisively and more sensitively by measurements on the rotating elements. Machines having relatively stiff and/or heavy casings in comparison to rotor mass are typical of those classes of machines for which shaft vibration measurements are frequently preferred.

Vibration measurements are used for a number of purposes, ranging from routine operational monitoring and acceptance tests to advanced experimental testing, as well as diagnostic and analytical investigations. These various measurement objectives lead to many differences in methods of interpretation and evaluation. To limit the number of these differences, this document is designed to provide guidelines primarily for operational monitoring and acceptance tests.

Three primary vibration quantities (displacement, velocity and acceleration) are defined and their limitations given. Adherence to the guidelines presented should, in most cases, ensure satisfactory service performance.

Mechanical vibration — Measurement and evaluation of machine vibration —

Part 1:
General guidelines

1 Scope

This document establishes general conditions and procedures for the measurement and evaluation of vibration using measurements made on rotating, non-rotating and non-reciprocating parts of complete machines. It is applicable to measurements of both absolute and relative radial shaft vibration with regard to the monitoring of radial clearances, but excludes axial shaft vibration. The general evaluation criteria, which are presented in terms of both vibration magnitude and change of vibration, relate to both operational monitoring and acceptance testing. They have been provided primarily with regard to securing reliable, safe, long-term operation of the machine while minimizing adverse effects on associated equipment. Guidelines are also presented for setting operational limits.

NOTE 1   The evaluation criteria for different classes of machinery will be included in other parts of ISO 20816 when they become available. In the meantime, guidelines are given in Clause 6.

NOTE 2   The term "shaft vibration" is used throughout ISO 20816 because, in most cases, measurements are made on machine shafts. However, the ISO 20816 series is also applicable to measurements made on other rotating elements if such elements are found to be more suitable, provided that the guidelines are respected.

For the purposes of ISO 20816, operational monitoring is considered to be those vibration measurements made during the normal operation of a machine. The ISO 20816 series permits the use of different measurement quantities and methods, provided that they are well-defined and their limitations are set out, so that the interpretation of the measurements is well-understood.

The evaluation criteria relate only to the vibration produced by the machine itself and not the vibration transmitted to it from outside.

This document does not include consideration of torsional vibration.

NOTE 3   For torsional vibration, see, for example, ISO 3046-5, ISO 22266-1 or VDI 2039.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 2954, *Mechanical vibration of rotating and reciprocating machinery — Requirements for instruments for measuring vibration severity*

ISO 5348, *Mechanical vibration and shock — Mechanical mounting of accelerometers*

ISO 10817-1, *Rotating shaft vibration measuring systems — Part 1: Relative and absolute sensing of radial vibration*

3 Terms and definitions

No terms and definitions are listed in this document.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— IEC Electropedia: available at http://www.electropedia.org/

— ISO Online browsing platform: available at http://www.iso.org/obp

4 Measurements

4.1 General

4.1.1 Overview

This clause describes the measurements, procedures and operating conditions recommended for assessing machine vibration. The guidelines given permit the evaluation of vibration in accordance with the general criteria and principles given in Clause 6.

4.1.2 Vibration measurements

It is common practice to measure vibration on non-rotating parts or to measure relative shaft vibration, or both. The measurement type for the protection system is normally based on the experience from the machine manufacturer.

4.1.3 Frequency range

The measurement of vibration shall be broad band, so that the frequency spectrum of the machine is adequately covered.

The frequency range depends on the type of machine being considered (e.g. the frequency range necessary to assess the integrity of rolling element bearings should include frequencies higher than those on machines with fluid-film bearings only).

Guidelines for instrumentation frequency ranges for specific machine classes are given in the appropriate parts of ISO 20816.

NOTE 1   In the past, broad-band measurements in the range 10 Hz to 1 000 Hz were the intended metric for full-load acceptance testing. This might not meet the requirements of a condition monitoring scheme and might need to be modified for the purposes of vibration monitoring and diagnostics.

NOTE 2   Vibration condition monitoring and diagnostics of machines are described in ISO 13373.

On certain equipment, e.g. gearboxes and rolling element bearings, it can be appropriate to use a different frequency range for acceptance purposes.

4.2 Types of measurements

4.2.1 Vibration measurements on non-rotating parts

Vibration measurements on non-rotating parts are generally carried out with a seismic transducer which senses the absolute velocity or acceleration of structure parts on which it is mounted (e.g. the bearing housing).

4.2.2 Relative shaft vibration measurements

Relative shaft vibration measurements are generally carried out with a non-contacting transducer which senses the vibratory displacement between the shaft and a structural member on which it is mounted (e.g. the bearing housing).

4.2.3 Absolute shaft vibration measurements

Absolute shaft vibration measurements are carried out by one of the following methods:

a) by a shaft-riding probe on which a seismic transducer (velocity type or accelerometer) is mounted so that it measures absolute shaft vibration directly;

b) by a non-contacting transducer which measures relative shaft vibration in combination with a seismic transducer (velocity type or accelerometer) which measures the support vibration. Both transducers shall be mounted close together so that they undergo the same absolute motion in the direction of measurement. Their conditioned outputs are summed to provide a measurement of the absolute shaft motion.

NOTE   In order to avoid incorrect results, it is important to ensure that the same datum time reference be used for the outputs from both the seismic and non-contacting transducers.

4.3 Measurement parameters

4.3.1 Measurement quantities

For the purposes of this document, the following measurement quantities can be used:

a) vibration displacement, measured in micrometres;

b) vibration velocity, measured in millimetres per second;

c) vibration acceleration, measured in metres per square second.

The use, application and limitations of these quantities are discussed in Clause 6.

Generally, there is no simple relationship between broad-band acceleration, velocity and displacement; nor is there between peak (0–p), peak-to-peak (p–p), root-mean-square (r.m.s.) and average values of vibration. The reasons for this are briefly discussed in A.1, which also defines some precise relationships between the above quantities when the harmonic content of the vibration waveform is known.

In order to avoid confusion and to ensure correct interpretation, it is important at all times to identify clearly the measurement quantity and its unit, e.g. peak-to-peak displacement in μm (1 μm = 10^{-6} m), r.m.s. velocity in mm/s.

NOTE   Vibration is a vector quantity and therefore, when comparing two different values, it can be necessary to consider the phase angle between them (see Annex D).

Generally, it can be stated that the preferred measurement quantity for the measurement of vibration of non-rotating parts is r.m.s. velocity while the preferred measurement quantity for the measurement of shaft vibration is peak-to-peak displacement.

Since this document applies to both relative and absolute shaft vibration measurements, displacement is further defined as follows:

— relative displacement which is the vibratory displacement of the shaft with reference to support structure, such as a bearing housing or machine casing;

— absolute displacement which is the vibratory displacement of the shaft with reference to an inertial reference system.

It should be clearly indicated whether displacement values are relative or absolute.

Relative and absolute displacements are further defined by several different displacement quantities, each of which is now in widespread use. These include the following:

S_{max} maximum vibratory displacement in the plane of measurement, measured from time-integrated mean position, see Formula (A.10);

$S_{(p-p)}$ peak-to-peak vibratory displacement in the direction of measurement defined as $S_{(p-p)} = \max [S_{A(p-p)}, S_{B(p-p)}]$;

$S_{(p-p)max}$ maximum peak-to-peak vibratory displacement in the plane of measurement.

Any of these displacement quantities may be used for the measurement of shaft vibration. However, the quantities shall be clearly identified so as to ensure correct interpretation of the measurements in terms of the criteria of Clause 6. The relationships between these quantities are shown in Figure A.3.

4.3.2 Vibration magnitude

The result of measurements made with an instrument which complies with the requirements of Clause 5 is called the vibration magnitude at a specific measuring position and direction.

It is common practice, based on experience, when evaluating broad-band vibration of rotating machinery to consider the r.m.s. value of vibration velocity, since this can be related to the vibration energy. However, other quantities such as displacement or acceleration and peak values instead of r.m.s. values may be preferred. In this case, alternative criteria, which are not necessarily simply related to criteria based on r.m.s. values, are required.

For shaft vibration measurement, it is common practice to consider peak-to-peak values.

4.3.3 Vibration severity

Normally, measurements are made at various measuring positions and in one, two or three measuring directions, leading to a set of different vibration magnitude values. The maximum broad-band magnitude value measured under agreed machine support and operating conditions is defined as the vibration severity.

For most machine types, one value of vibration severity characterizes the vibratory state of that machine. However, for some machines, this approach can be inadequate and the vibration severity should then be assessed independently for measurement positions at a number of locations.

4.4 Measuring positions

4.4.1 Positions for measurements on non-rotating parts

Measurements on non-rotating parts should be taken on the bearings, bearing support housing or other structural parts which significantly respond to the dynamic forces transmitted from the rotating elements at the bearing locations and characterize the overall vibration of the machine. Typical measurement locations are shown in Figure 1 to Figure 5.

To determine the vibrational behaviour at each measuring position, it is necessary to take measurements in three mutually perpendicular directions. The full complement of measurements represented in Figure 1 to Figure 5 is generally only required for acceptance testing.

The requirement for operational monitoring is usually met by performing one or both measurements in the radial direction (i.e. normally in the horizontal transverse and/or vertical directions). These can be supplemented by a measurement of the vibration in the axial direction, but are to be evaluated only on thrust bearings. The latter can be of significance at thrust bearing locations where direct axial dynamic forces are transmitted.

Detailed recommendations for specific machine types are provided in the additional parts of ISO 20816.
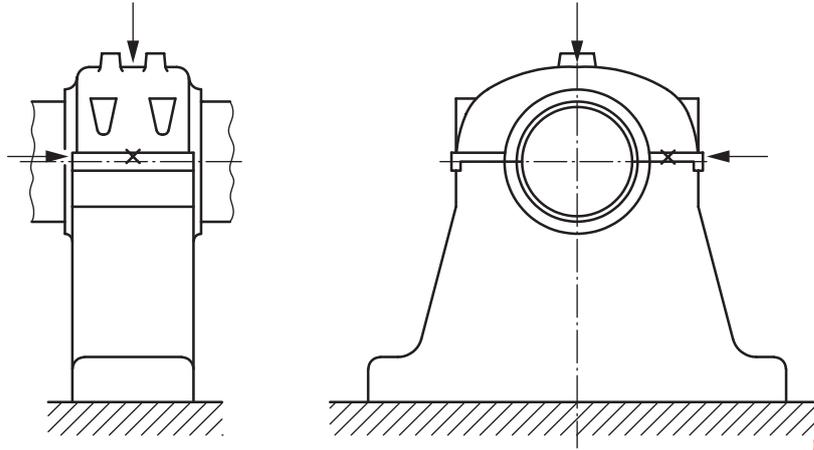
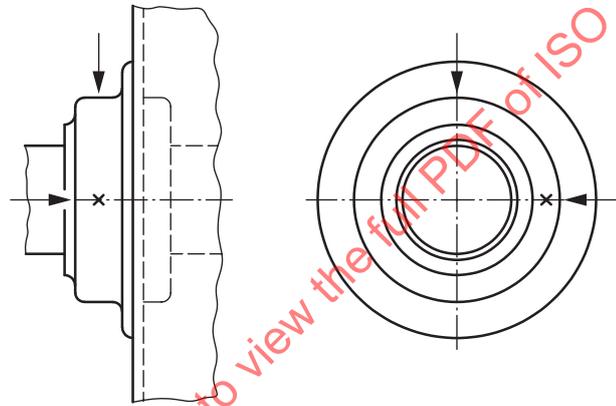**Figure 1 — Measuring points for pedestal bearings**
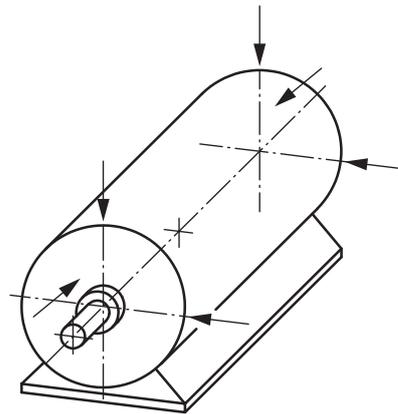
Figure 2 — Measuring points for housing-type bearings
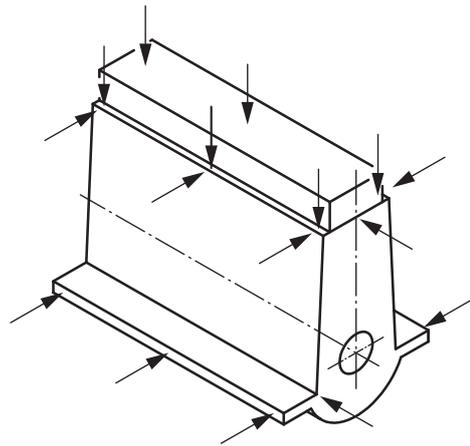
Figure 3 — Measuring points for small electrical machines

Figure 4 — Measuring points for reciprocating engines close to the bearing locations
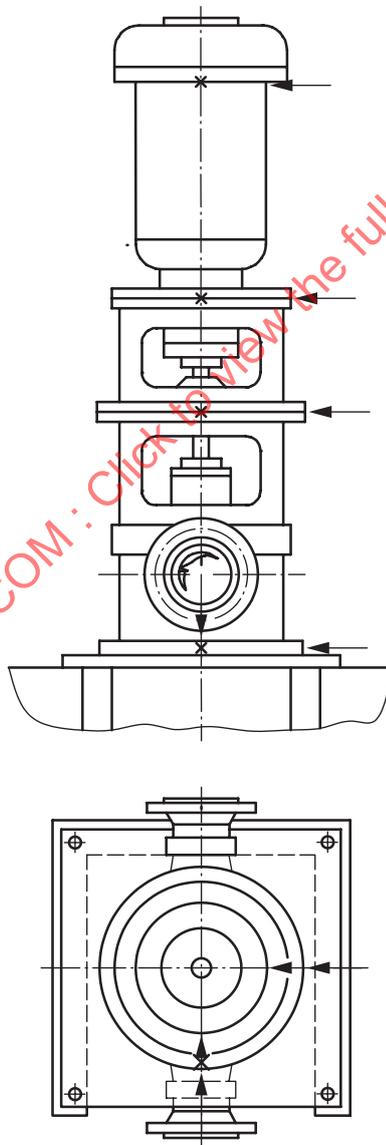
Figure 5 — Measuring points for vertical machine sets

4.4.2 Positions for measurements on rotating shafts

4.4.2.1 General

For the measurements on rotating shafts, it is desirable to locate transducers at positions such that the lateral movement of the shaft at points of importance can be assessed. It is recommended that, for both relative and absolute measurements, two transducers be located at, or adjacent to, each machine bearing, see Figure 6. They should be radially mounted in the same transverse plane perpendicular to the shaft axis or as close as practicable, with their axes within ±5° of a radial line. It is preferable to mount both transducers 90° ± 5° apart on the same bearing half and the positions chosen should be the same at each bearing. See Figure 7.

A single transducer may be used at each measurement plane in place of the more typical pair of orthogonal transducers if it is known to provide adequate information about the shaft vibration.

It is recommended that special measurements be made in order to determine the total non-vibration runout, which is caused by shaft surface metallurgical non-homogeneities, local residual magnetism and shaft mechanical runout. It should be noted that, for anisotropic rotors such as, for example, two-pole generators, the effect of gravity can cause a false runout signal.

Key
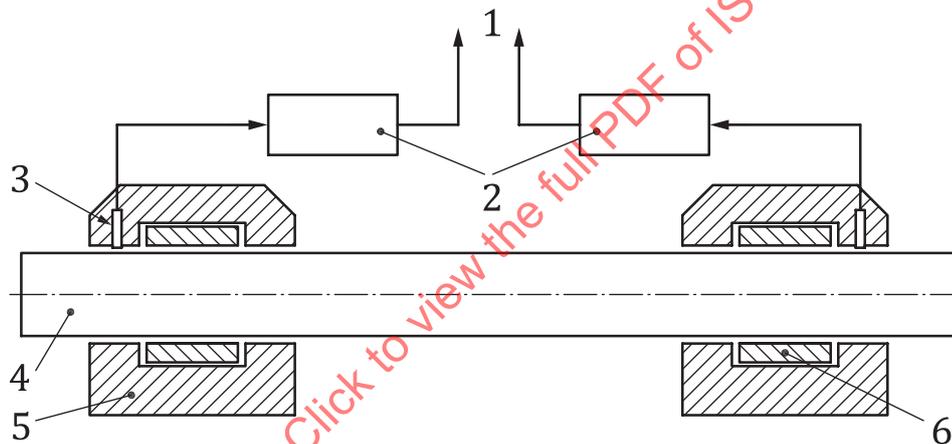
1   to signal processing
2   signal conditioning units
3   non-contacting transducers
4   shaft
5   bearing housings
6   bearings

Figure 6 — Measuring points for measurements on rotating shafts
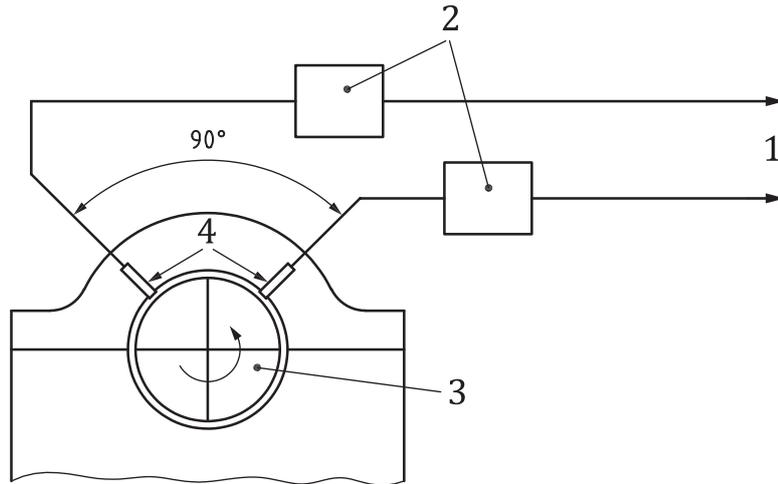
Key

1   to signal processing
2   signal conditioning units
3   shaft
4   non-contacting transducers

Figure 7 — Mounting of non-contacting probes for the measurement of shaft relative vibration

4.4.2.2   Positions for relative shaft vibration measurements

Relative vibration transducers of the non-contacting type are normally mounted in tapped holes in the bearing housing or by rigid brackets adjacent to the bearing housing or bearing shell. Where the transducers are mounted in the bearing, they should be located so as not to interfere with the lubrication pressure wedge. However, special arrangements for mounting transducers in other axial locations may be made, but different vibration criteria for assessment will then have to be used. For bracket-mounted transducers, the bracket shall be free from natural frequencies which adversely affect the capability of the transducer to measure the relative shaft vibration.

The surface of the shaft at the location of the transducer, taking into account the total axial float of the shaft under all thermal conditions, shall be smooth and free from any geometric discontinuities (such as keyways, lubrication passages and threads), metallurgical non-homogeneities and local residual magnetism which can cause false signals. In some circumstances, an electroplated or metallized shaft surface may be acceptable, but it should be noted that the calibration can be different. It is recommended that the total combined electrical and mechanical runout, as measured by the transducer, does not exceed 25 % of the allowable vibration displacement, specified in accordance with 6.3.2.2, or 6 μm, whichever is greater. For measurements made on machines already in service, where provision was not originally made for shaft vibration measurements, it can be necessary to use other runout criteria.

4.4.2.3   Positions for absolute shaft vibration measurements using combined seismic and non-contacting relative vibration transducers

If a combination of seismic and non-contacting relative vibration transducers is used, the absolute shaft vibration is obtained by

a)   integrating the signal from the seismic transducer to convert the acceleration or velocity output to displacement, and

b)   summing the displacement outputs from both transducers.

NOTE 1   In order to avoid incorrect results, it is important to ensure that the same datum time reference is used for the outputs from both the seismic and non-contacting transducers.

The mounting and other requirements for the non-contacting transducer are as specified in 4.4.2.2. In addition, the seismic transducer shall be rigidly mounted to the machine structure (e.g. the bearing housing) close to the non-contacting transducer so that both transducers undergo the same absolute vibration of the support structure in the direction of measurement. The sensitive axes of the non-contacting and seismic transducers shall be parallel, so that their summed, conditioned signals result in an accurate measure of the absolute shaft vibration. See Figure 8.

NOTE 2   Information on mounting accelerometers for seismic measurements is given in ISO 5348.

NOTE 3   In the past, absolute shaft vibration measurements have also been performed using a shaft-riding mechanism with a seismic transducer.
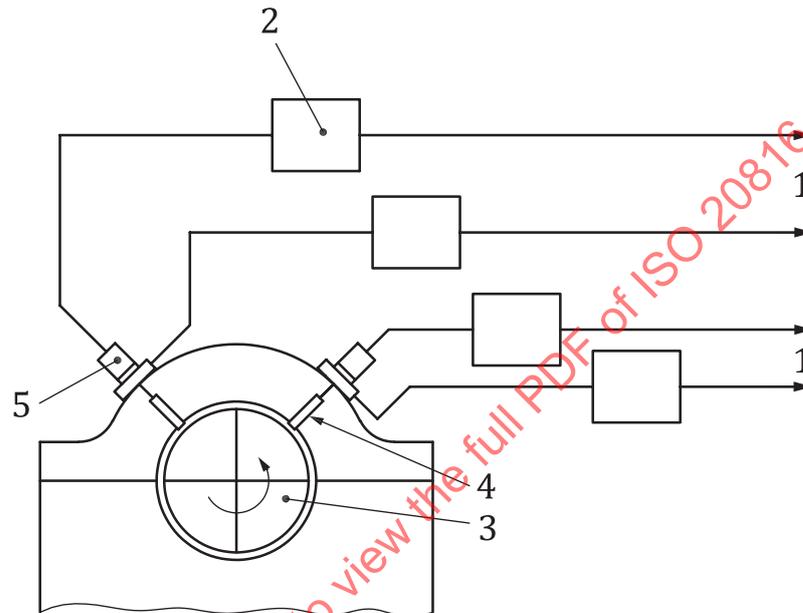
Key
1   to signal processing
2   signal conditioning units
3   shaft
4   non-contacting transducers
5   seismic transducers

Figure 8 — Mounting of non-contacting and seismic probes for the measurement of shaft absolute vibration

4.5   Machine support structure for acceptance testing

4.5.1   General

The acceptance criteria should be agreed between the customer and manufacturer.

4.5.2   In-situ tests

When acceptance is carried out in situ, the support structure shall be that supplied for the machine. In this case, it is important to ensure that all the major components of the machine and structure are installed when the test is carried out.

Valid comparisons of vibration for machines of the same type but on different foundations or sub-foundations can only be made if the foundations concerned have similar dynamic characteristics.

4.5.3 In a test facility

There are many classes of machines for which, because of economic or other reasons, acceptance is carried out on a test bed which can have different support structure characteristics from those at the site. The support structure can significantly affect the measured vibration and every attempt should be made to ensure that the natural frequencies of the complete test arrangement do not coincide with the rotational frequencies of the machine or with any of its significant harmonics.

The test arrangement normally meets these requirements if the vibration magnitude measured in the horizontal and vertical directions at the machine feet, or at the base frame near the bearing support or stator feet, does not exceed 50 % of the vibration magnitude measured in the same measuring direction at that bearing. Additionally, the test arrangement shall not cause a substantial change in any of the major resonance frequencies.

If a significant support resonance is present during acceptance test and it cannot be eliminated, the vibration acceptance may have to be carried out on the fully installed machine in situ.

For some classes of machines (e.g. small electrical machinery), acceptance tests can be carried out when machines are supported by a resilient system (see, for example, IEC 60034-14). In this case, all the rigid-body mode frequencies of the machine on its support system shall be less than one-half of the lowest significant excitation frequency of the machine. Appropriate support conditions can be achieved by mounting the machine on a resilient support baseplate or by free suspension on a soft spring.

4.6 Machine operating conditions

Shaft vibration measurements shall be made after achieving contractually agreed normal operating conditions (speed, load, temperature, pressure, etc.). Additional vibration measurements that may be taken under other conditions are not applicable for evaluation in accordance with Clause 6.

4.7 Evaluation of vibration from other sources

If the measured vibration magnitude exceeds the recommended limit, it can then be necessary to take measurements of environmental vibration with the machine shut down to ensure that this is not making a significant contribution to the observed vibration. Where possible, steps should be taken to reduce the magnitude of environmental vibration if it is greater than one-third of the recommended limits.

5 Instrumentation

The instrumentation used shall be designed to operate satisfactorily in the environment for which it is to be used, for example, with respect to temperature and humidity. Specification for instruments for measuring vibration severity is given in ISO 2954 and instruments for measuring shaft vibration are specified in ISO 10817-1.

Particular attention shall be given to ensure that the vibration transducer is correctly mounted and that its presence does not affect the vibration response characteristics of the machine. Specifications for mounting accelerometers are given in ISO 5348, which, in principle, are applicable also to velocity transducers.

Modern instrumentation can provide multiple methods to deliver a measurement value. The acceptance criteria here are based on r.m.s. velocity in mm/s for measurements on non-rotating parts and peak-to-peak displacement in micrometres for measurements on rotating parts. The acceptance criteria may be scaled suitably to match the units in common use at the site, using the same assumptions as programmed into the distributed control systems.

It is desirable that the measurement system has provision for on-line calibration of the readout instrumentation and, in addition, has suitable isolated outputs to permit further analysis as required.

6 Evaluation criteria

6.1 General

6.1.1 Overview

This clause specifies general criteria and principles for the evaluation of machine vibration. The evaluation criteria relate to both operational monitoring and acceptance testing, and they apply only to the vibration produced by the machine itself and not the vibration transmitted from outside.

The specification of evaluation criteria for machine vibration is dependent upon a wide range of factors and the criteria adopted vary significantly for different types of machine and, in some cases, for different rotors in the same coupled line. It is important, therefore, to ensure that valid criteria are adopted for a particular machine and that the criteria which relate to a certain type of machine are not erroneously applied to other types.

EXAMPLE   Evaluation criteria for a high-speed compressor operating in a petrochemical plant are likely to be different from those for large turbo-generators.

This clause establishes a basis for specifying evaluation criteria for measurements on non-rotating parts, as well as on rotating shafts. No attempt has been made to specify vibration values. If the procedures of both kinds of measurements are applicable, the one which is more restrictive shall generally apply. Specific criteria for different classes and types of machinery are given in the relevant parts of ISO 20816.

The criteria presented in this clause are applicable only to the vibration due to excitation from the rotating elements. The criteria are not applicable to the evaluation of the electromagnetic vibration excited at twice the line frequency (i.e. twice the electrical system frequency) and transmitted from the generator stator core to the bearings.

6.1.2 Types of measurement on rotating shafts

6.1.2.1   There are two principal factors by which shaft vibration is judged:

a)   absolute vibration of the shaft;

b)   vibration of the shaft relative to the structural elements.

6.1.2.2   If the evaluation criterion is the change in shaft vibration, then

a)   when the vibration of the structure, on which the shaft-relative transducer is mounted, is small (i.e. less than 20 % of the relative shaft vibration), either the relative shaft vibration or absolute shaft vibration may be used as a measure of shaft vibration, and

b)   when the vibration of the structure, on which the shaft-relative transducer is mounted, is 20 % or more of the relative shaft vibration, the absolute shaft vibration shall be measured and, if found to be larger than the relative shaft vibration, it shall be used as the measure of shaft vibration.

6.1.2.3   If the evaluation criterion is the dynamic load on the bearing, the relative shaft vibration shall be used as the measure of shaft vibration.

6.1.2.4   If the evaluation criterion is stator/rotor clearance, then

a)   when the vibration of the structure, on which the shaft-relative transducer is mounted, is small (i.e. less than 20 % of the relative shaft vibration), the relative shaft vibration shall be used as a measure of clearance absorption, and

b)   when the vibration of the structure, on which the shaft-relative transducer is mounted, is 20 % or more of the relative shaft vibration, the relative shaft vibration measurement may still be used as a

measure of clearance absorption unless the vibration of the structure, on which the shaft-relative transducer is mounted, is not representative of the total stator vibration. In this latter case, special measurements are required.

6.1.2.5 The shaft vibration associated with a particular evaluation range depends on the size and mass of the vibrating body, the characteristics of the mounting system and the output and use of the machine. It is therefore necessary to take into account the various purposes and circumstances concerned when specifying different ranges of shaft vibration for a specific class of machinery. Where appropriate, reference should be made to the product specification.

6.2 Factors affecting evaluation criteria

There are a wide range of different factors which need to be taken into account when specifying evaluation criteria for vibration measurements. Amongst these are the following:

a) the purpose for which the measurement is made (e.g. the requirement for ensuring that running clearances are maintained is, in general, different from that if the avoidance of excessive dynamic load on the bearing is the main concern);

b) the type of measurement made (vibration of non-rotating parts, relative or absolute shaft vibration);

c) the quantities measured (see Annex A);

d) the positions where the measurements are made;

e) the rotational speed of the shaft;

f) the bearing type, clearance and diameter;

g) the function, output and size of the machine;

h) the stiffness of the bearings, pedestals and foundations;

i) the rotor mass and stiffness.

Clearly, this wide range of factors makes it impossible to define unique evaluation criteria which can be applied to all machines. Different criteria, which have been derived from operational experience, are necessary for different machines, but at best, they can only be regarded as guidelines and there will be occasions where machines will operate safely and satisfactorily outside any general recommendations or are unable to operate despite vibration values are well within guideline values.

6.3 Types of evaluation criteria

6.3.1 General

Two evaluation criteria are used to assess vibration severity on various classes of machines. One criterion considers the magnitude of observed broad-band vibration; the second considers changes in magnitude, irrespective of whether they are increases or decreases.

6.3.2 Criterion I: Vibration magnitude at rated speed under steady operation conditions

6.3.2.1 Non-rotating parts

For non-rotating parts, this criterion is concerned with defining limits for absolute vibration magnitude consistent with acceptable dynamic loads on the bearings and acceptable vibration transmission into the support structure and foundation. The maximum vibration magnitude observed at each bearing or pedestal is assessed against four evaluation zones established from international experience. This maximum magnitude of vibration measured is defined as the vibration severity (see 4.3.3).

The vibration of a particular machine depends on its size, the characteristics of the vibrating body and mounting system and the purpose for which it is designed. It is therefore necessary to take account of the various purposes and circumstances concerned when specifying ranges of vibration measurement for different machine types. For nearly all machines, regardless of the type of bearings used, in general, measurements of the broad-band r.m.s. vibration velocity on structural parts, such as bearing housings, adequately characterize the running conditions of the rotating shaft elements with respect to their trouble-free operation.

In most cases, it has been found that vibration velocity is sufficient to characterize the severity of vibration over a wide range of machine operating speeds. However, it is recognized that the use of a single value of velocity, regardless of frequency, can lead to unacceptably large vibration displacements. This is particularly so for machines with low operating speeds when the once-per-revolution vibration component is dominant. Similarly, constant-velocity criteria for machines with high operating speeds, or with vibration at high frequencies generated by machine component parts, can lead to unacceptable accelerations. Consequently, evaluation criteria based on velocity take the general form of Figure 9. This indicates the upper and lower frequency limits, f_u and f_l, and shows that below a defined frequency f_x and above a defined frequency f_y, the allowable vibration velocity is a function of frequency (see also C.2). However, for vibration frequencies between f_x and f_y, a constant-velocity criterion applies.

NOTE   The r.m.s. velocity values for non-rotating parts listed in Table C.1 refer to this constant-velocity region.

The precise nature of the evaluation criteria and the values of f_l, f_u, f_x and f_y for specific machine types are given in the additional parts of ISO 20816.

For many machines, the broad-band vibration consists primarily of a single frequency component, often shaft rotational frequency. In this case, the allowable vibration is obtained from Figure 9 as the vibration velocity corresponding to that frequency.

For less-common machines, where there can be significant vibratory energy beyond the breakpoints f_x and f_y of Figure 9, a number of different approaches are possible. Examples are as follows.

a) In addition to the usual broad-band velocity, broad-band displacement may be measured when there is significant energy below f_x. Similarly, broad-band acceleration may be measured when there is significant energy above f_y. The allowable vibration displacement and acceleration should be consistent with the velocity corresponding to the sloped portions of Figure 9.

b) The velocity, displacement or acceleration at each significant component throughout the frequency spectrum may be determined using a frequency analyser. The equivalent broad-band velocity can be obtained using Formula (A.2) after applying appropriate weighting factors, consistent with Figure 9, for those components whose frequencies are below f_x or above f_y. This value should then be evaluated relative to the constant velocity between f_x and f_y. It should be noted that, except for the case when the broad-band vibration consists primarily of a single frequency component, a direct comparison of the frequency spectrum components with the curves of Figure 9 would yield misleading results.

c) A composite broad-band measurement encompassing the entire spectrum may be carried out using an instrument incorporating weighting networks consistent with the shape of Figure 9. This value should then be evaluated relative to the constant velocity between f_x and f_y.

The evaluation criteria for specific machine types are given in the additional parts of ISO 20816. C.2 provides additional guidance. For certain machine types, it can be necessary to define further criteria beyond those described by Figure 9 (see, for example, 6.5.3).

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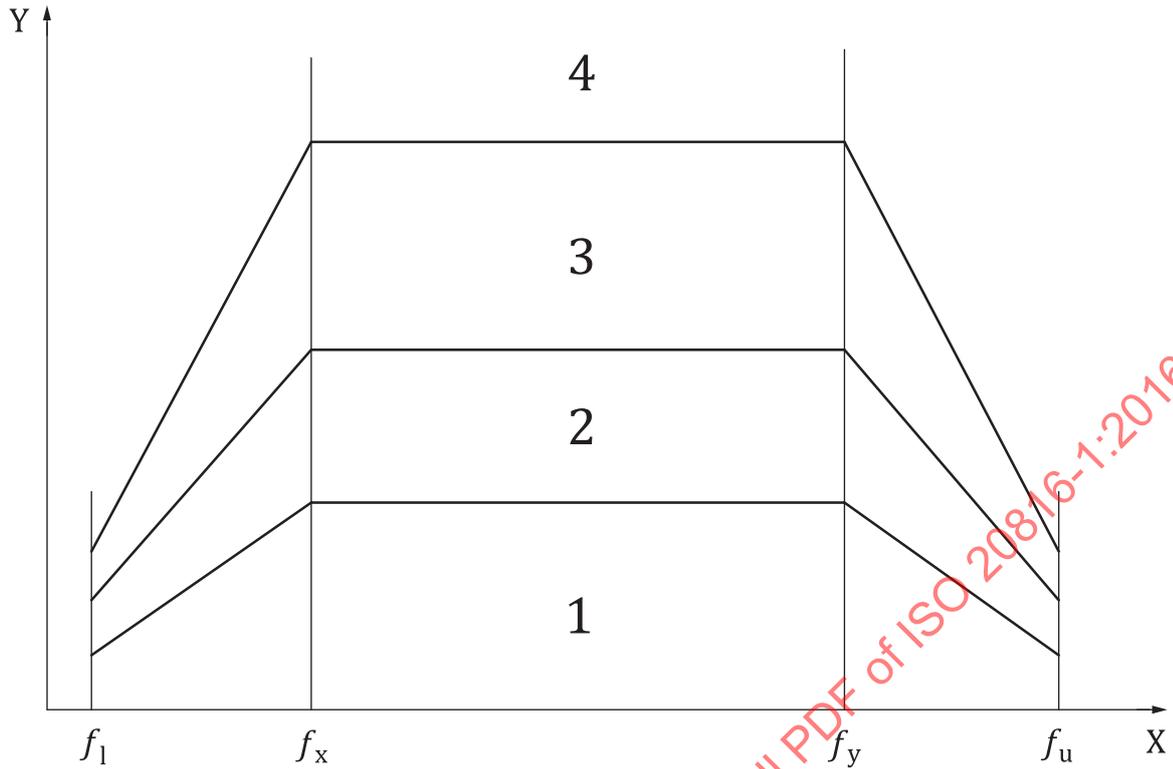
Key

X   frequency
Y   r.m.s. vibration velocity
1   zone A
2   zone B
3   zone C
4   zone D

Figure 9 — General form of vibration velocity evaluation criteria

6.3.2.2 Rotating shafts

For rotating shafts, this criterion is concerned with defining limits for shaft vibration magnitude consistent with acceptable dynamic loads on the bearings, adequate margins on the radial clearance envelope of the machine and acceptable vibration transmission into the support structure and foundation. The maximum shaft vibration magnitude observed at each bearing is assessed against four evaluation zones established from international experience.

Figure 10 shows a plot of allowable vibration, in terms of peak-to-peak shaft vibration, against the operating speed range. It is generally accepted that limiting vibration values decrease as the operating speed of the machine increases, but the actual values and their rate of change with speed varies for different types of machines.
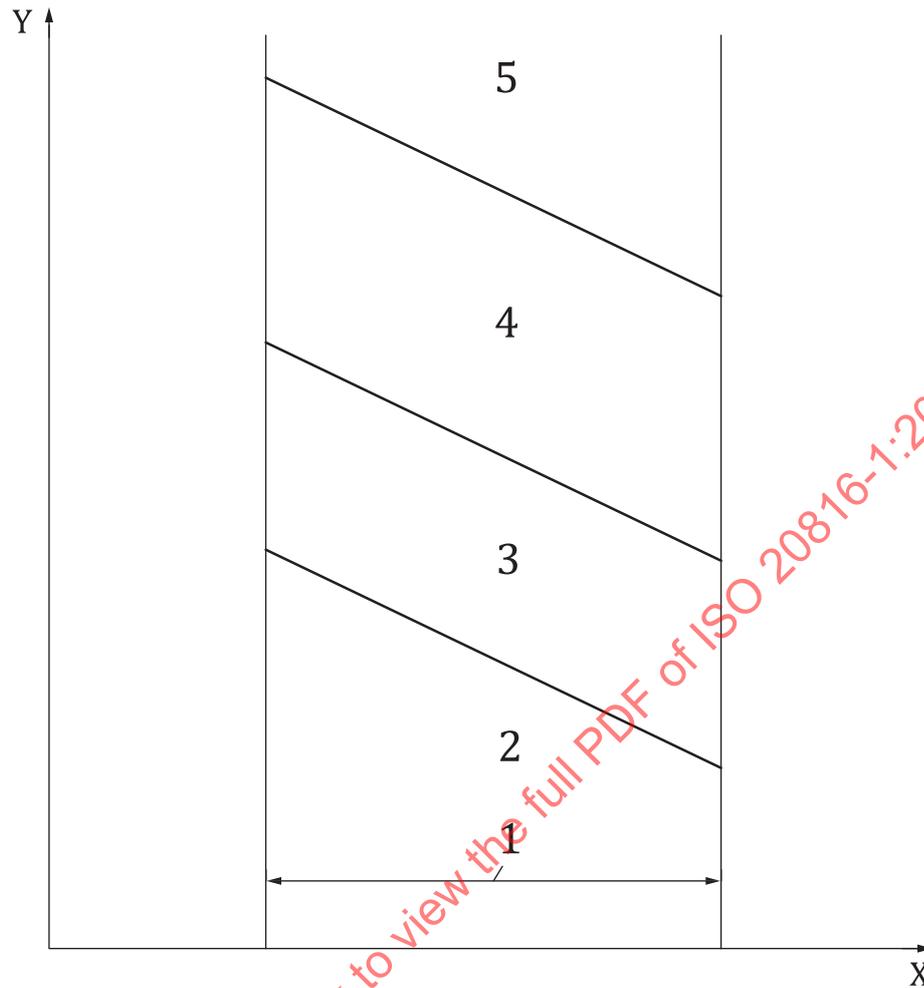
Key

X   shaft rotational speed

Y   peak-to-peak shaft vibration

1   relevant speed range

2   zone A

3   zone B

4   zone C

5   zone D

NOTE   The actual values for vibration at the zone boundaries and the relevant speed range vary for different types of machines. It is important to select the relevant criteria and to avoid incorrect extrapolation.

Figure 10 — Generalized example of evaluation criteria for shaft vibration

6.3.2.3   Evaluation zones

The following evaluation zones are defined to permit a qualitative assessment of the vibration on a given machine under steady-state conditions at normal operating speed and to provide guidelines on possible actions. Different categorization and number of zones may apply for specific machine types. These are provided in additional parts of ISO 20816.

Zone A: The vibration of newly commissioned machines normally falls within this zone.

NOTE   The effort required to achieve vibration within zone A can be disproportionate and unnecessary.

Zone B: Machines with vibration within this zone are normally considered acceptable for unrestricted long-term operation.

Zone C: Machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.

Zone D: Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine.

6.3.2.4 Evaluation zone limits

Numerical values assigned to the zone boundaries, which are provided in additional parts of ISO 20816, provide guidelines for ensuring that gross deficiencies or unrealistic requirements are avoided. In certain cases, there can be specific features associated with a particular machine which would require different zone boundary values (higher or lower) to be used. In such cases, it is normally necessary to explain the reasons for this and, in particular, to confirm that the machine will not be endangered by operating with higher vibration values.

NOTE   For the vibration evaluation for non-rotating parts of those machines for which no specific International Standard exists, see Annex C.

6.3.2.5 Acceptance criteria

The acceptance criteria should be subject to agreement between the machine manufacturer and customer, and prior negotiation is encouraged. The evaluation zones provide a basis for defining acceptance criteria but the numerical values assigned to the zone boundaries are not themselves intended to serve as acceptance specifications.

Contractual acceptance tests shall be carried out under clearly defined duration and operating parameters, e.g. load, temperature, pressure.

After major component replacement, maintenance or service activities, acceptance criteria shall take into account the scope of activity and the vibration behaviour prior to servicing.

6.3.3 Criterion II: Change in vibration magnitude

This criterion provides an assessment of a change in vibration magnitude from a previously established reference value. A significant increase or decrease in vibration magnitude can occur, which requires some action even though zone C of Criterion I has not been reached. Such changes can be instantaneous or progressive with time and can indicate that damage has occurred or can be a warning of an impending failure or some other irregularity. Criterion II is specified on the basis of the change in vibration magnitude occurring under steady-state operating conditions.

When Criterion II is applied, the vibration measurements being compared shall be taken at the same transducer location and orientation and under approximately the same machine operating conditions. Significant changes from the normal vibration magnitudes should be investigated so that a dangerous situation can be avoided.

Criteria for assessing changes of vibration for monitoring purposes are given in the additional parts of ISO 20816. However, it should be noted that some changes might not be detected unless the discrete frequency components are monitored (see 6.5.1).

6.4 Operational limits

6.4.1 General

For long-term operation, it is common practice for some machine types to establish operational vibration limits. These limits take the form of ALARMS and TRIPS.

ALARMS: To provide a warning that a defined value of vibration has been reached or a significant change has occurred, at which remedial action can be necessary. In general, if an ALARM situation occurs, operation may continue for a period whilst investigations are carried out to identify the reason for the change in vibration and define any remedial action.

TRIPS: To specify the magnitude of vibration beyond which further operation of the machine can cause damage. If the TRIP value is exceeded, immediate action should be taken to reduce the vibration or the machine should be shut down.

Different operational limits, reflecting differences in dynamic load, bearing/seal clearances and support stiffness may be specified for different measurement positions and directions.

Where appropriate, guidelines for specifying ALARM and TRIP criteria for specific machine types are given in other parts of ISO 20816.

6.4.2 Setting of ALARMS

The ALARM values can vary considerably, up or down, for different machines. The values chosen will normally be set relative to a baseline value determined from experience for the measurement position or direction for that particular machine.

Where there is no established baseline (e.g. with a new machine), the initial ALARM setting should be based either on experience with other similar machines or relative to agreed acceptance values. After a period of time, the steady-state baseline value will be established and the ALARM setting should be adjusted accordingly.

If the steady-state baseline changes (e.g. after a machine overhaul), the ALARM setting should be revised accordingly. Different operational ALARM settings can then exist for different bearings on the machine, reflecting differences in dynamic load and bearing support stiffnesses.

6.4.3 Setting of TRIPS

The TRIP values generally relate to the mechanical integrity of the machine and are dependent on any specific design features which have been introduced to enable the machine to withstand abnormal dynamic forces. The values used are, therefore, generally the same for all machines of similar design and would not normally be related to the steady-state baseline value used for setting ALARMS.

6.5 Additional factors

6.5.1 Vibration frequencies and vectors

The evaluation considered in this document is limited to broad-band vibration without reference to frequency components or phase. This is, in most cases, adequate for acceptance testing and operational monitoring purposes. However, in some cases, the use of vector information for vibration assessment on certain machines types can be desirable.

Vector change information is particularly useful in detecting and defining changes in the dynamic state of a machine. In some cases, these changes would go undetected when using broad-band vibration measurements. This is demonstrated in Annex D.

The specification of criteria for vector changes is beyond the scope of this document.

6.5.2 Vibration sensitivity of the machine

The vibration measured on a particular machine can be sensitive to changes in the steady-state operational condition. In most cases, this is not significant. In other cases, the vibration sensitivity can be such that although the vibration magnitude for a particular machine is satisfactory when measured under certain steady-state conditions, it can become unsatisfactory if these conditions change.

It is recommended that, in cases where some aspect of the vibration sensitivity of a machine is in question, agreement be reached between the customer and supplier about the necessity and extent of any testing or theoretical assessment.

6.5.3 Techniques for rolling element bearings

Alternative approaches other than broad-band vibration measurements are continuing to be evolved for assessing the conditions of rolling element bearings. These are discussed further in Annex B. The definition of evaluation criteria for such additional methods is beyond the scope of this document.

Annex A
(informative)

Explanation of measurement quantities

A.1 Vibration of non-rotating parts

It has been recognized for many years that using the measurement of r.m.s. velocity to characterize the vibratory response of a wide range of machine classes has been very successful and continues to be so. For simple alternating waveforms which are made up of a discrete number of harmonic components of known amplitude and phase, and do not contain significant random vibration or shock components, it is possible, by means of Fourier analysis, to relate various fundamental quantities (displacement, velocity, acceleration, peak, peak-to-peak, r.m.s., average, etc.) using rigorously determined mathematical relationships. A number of relationships are summarized in this annex.

From measured vibration velocity versus time records, the r.m.s. value of the velocity, v_{rms}, can be calculated by Formula (A.1):

$$v_{rms} = \sqrt{\frac{1}{T}\int_0^T v^2(t)\,dt}$$ (A.1)

where

$v(t)$ is the time-dependent vibration velocity;

T is the sampling time which is longer than the period of any of the major frequency components of which $v(t)$ is composed.

Acceleration, velocity and/or displacement magnitudes (a_j, v_j, s_j, respectively; j = 1, 2, …, n) can be determined for different frequencies ($f_1, f_2, …, f_n$) from spectral analyses.

If the peak-to-peak displacement values of the vibration, $s_1, s_2, …, s_n$, in micrometres, or the r.m.s. velocity values, $v_1, v_2, …, v_n$, in millimetres per second, or the r.m.s. acceleration values, $a_1, a_2, …, a_n$, in metres per square second, and the frequencies, $f_1, f_2, …, f_n$, in hertz, are known, the associated r.m.s. velocity, v_{rms}, characterizing the motion is given by Formula (A.2):

$$v_{rms} = 10^{-3}\pi\sqrt{\frac{1}{2}\left[(s_1 f_1)^2 + (s_2 f_2)^2 + … + (s_n f_n)^2\right]}$$
$$= \sqrt{v_1^2 + v_2^2 + … + v_n^2}$$
$$= \frac{10^3}{2\pi}\sqrt{\left(\frac{a_1}{f_1}\right)^2 + \left(\frac{a_2}{f_2}\right)^2 + … + \left(\frac{a_n}{f_n}\right)^2}$$ (A.2)

In the case where the vibration consists of only two significant frequency components giving beats of r.m.s. value, v_{min} and v_{max}, the associated r.m.s. vibration velocity, v_{rms}, can be determined approximately from Formula (A.3):

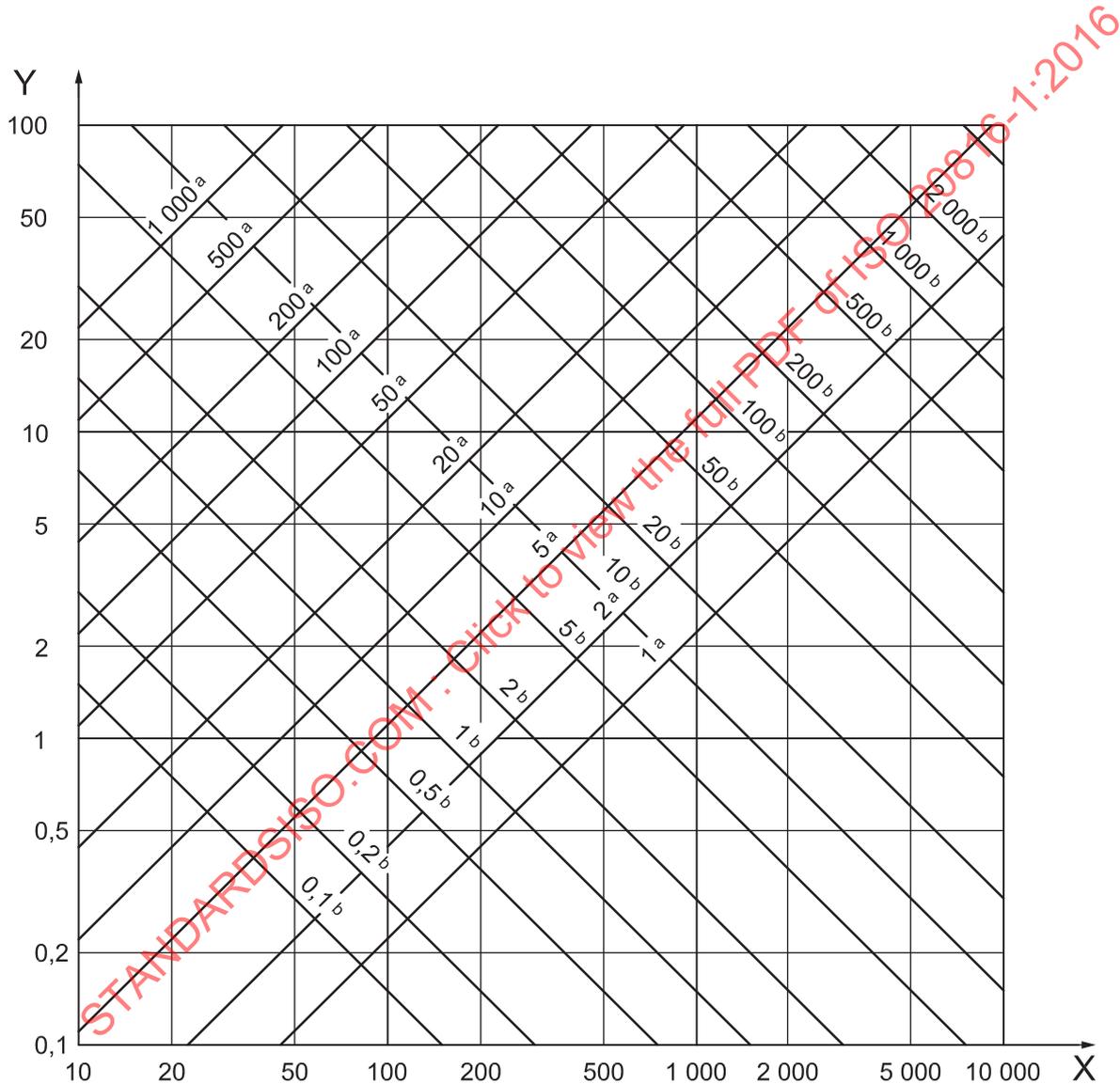$$v_{rms} = \sqrt{\frac{1}{2}\left(v_{max}^2 + v_{min}^2\right)}$$ (A.3)

The operation of interchanging vibration acceleration, velocity or displacement values can be accomplished only for single-frequency harmonic components using, for example, Figure A.1. If the

A.2 Vibration of rotating shafts

A.2.1 Time-integrated mean vibration

The mean values of the shaft displacement in any two specified orthogonal directions, $\bar{x}, \bar{y}$, relative to a reference position, as shown in Figure A.2, are defined by integrals with respect to time, as shown in Formula (A.5) and Formula (A.6):

$$\bar{x} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} x(t)\, \mathrm{d}t \tag{A.5}$$

$$\bar{y} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} y(t)\, \mathrm{d}t \tag{A.6}$$
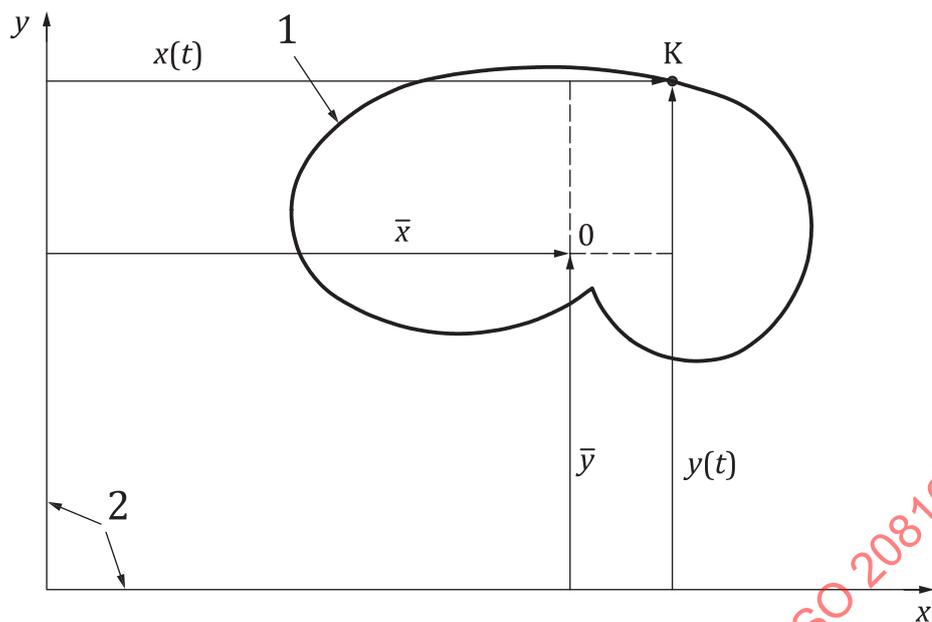
where

$x(t)$ and $y(t)$ are the time-dependent alternating values of displacement relative to the reference position;

$t_2 - t_1$ is the duration of integration which is large relative to the period of the lowest frequency vibration component.

In the case of absolute vibration measurements, the reference position is fixed in space. For relative vibration measurements, these mean values give an indication of the mean position of the shaft relative to the non-rotating parts at the axial location where the measurements are made. Changes in the mean values can be due to a number of factors, such as bearing/foundation movements, changes in oil film characteristics, which normally occur slowly relative to the period of the vibration components which make up the alternating mean values.

It should be noted that, in general, the time-integrated mean position in any direction differs from the position defined by taking half the sum of the maximum and minimum displacement values (see Figure A.3). However, when the shaft vibration is a single frequency and sinusoidal, then the locus of the shaft centre is an ellipse. In such circumstances, the time-integrated mean position in any direction of measurement is the same as the position identified by taking half the sum of the maximum and minimum displacement values.

Key

1	kinetic orbit of shaft
2	fixed reference axes x and y
$x(t), y(t)$	time-dependent alternating values of displacement relative to the reference position
$\overline{x}, \overline{y}$	time-integrated mean values of shaft displacement
0	time-integrated mean position of orbit
K	instantaneous position of shaft centre

Figure A.2 — Kinetic orbit of shaft
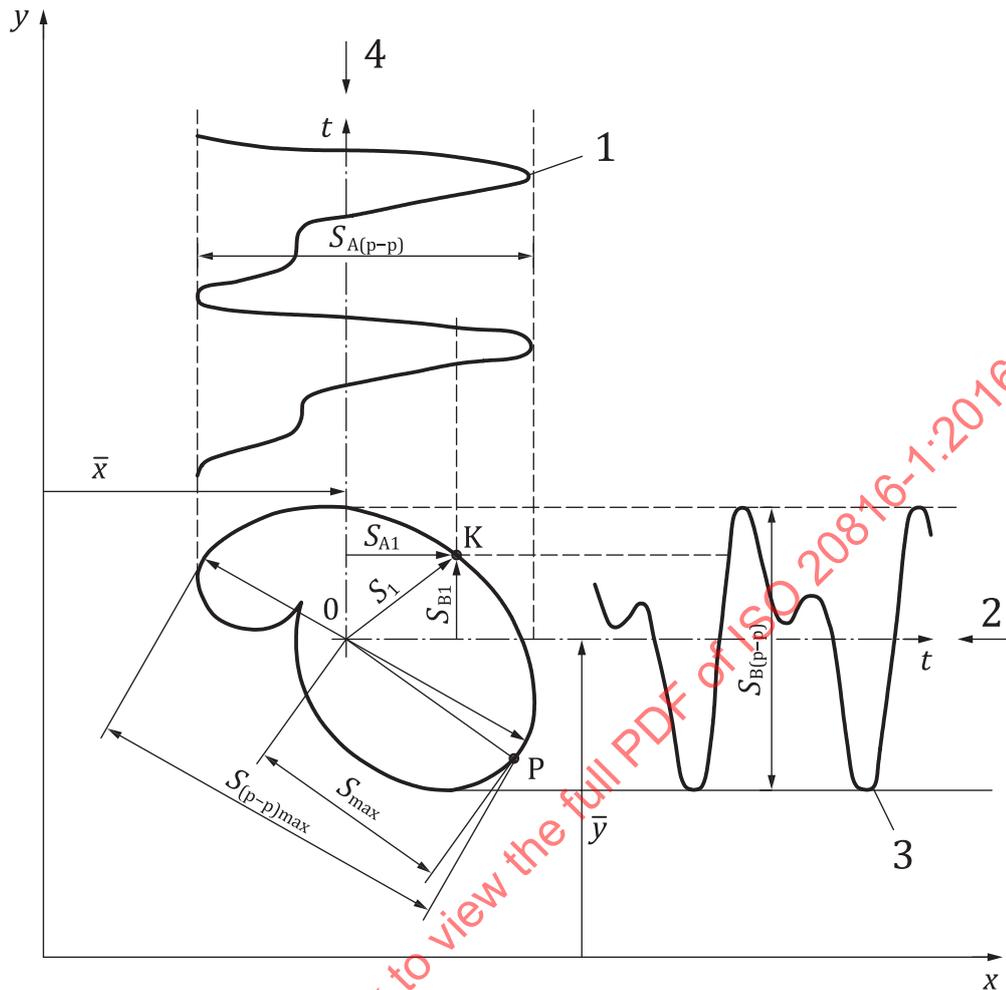
Key

1	transducer A waveform
2	transducer A
3	transducer B waveform
4	transducer B
x, y	fixed reference axes
0	time-integrated mean position of orbit
$\bar{x}, \bar{y}$	time-integrated mean values of shaft displacement
K	instantaneous position of shaft centre
P	position of shaft for maximum displacement from time-integrated mean position
S_1	instantaneous value of shaft displacement
S_{max}	maximum value of shaft displacement from time-integrated mean position 0
S_{A1}, S_{B1}	instantaneous values of shaft displacement in directions of transducers A and B, respectively
$S_{(p-p)max}$	maximum value of peak-to-peak displacement
$S_{A(p-p)}$, $S_{B(p-p)}$	peak-to-peak values of shaft displacement in directions of transducers A and B

NOTE   In this example sketch, $S_{A(p-p)} = S_{(p-p)}$ since $S_{A(p-p)} > S_{B(p-p)}$.

Figure A.3 — Definition of shaft displacement quantities

A.2.2 Peak-to-peak displacement of shaft vibration

A.2.2.1 General

The primary quantities of interest in shaft vibration measurements are the alternating values which describe the shape of the orbit. Consider the kinetic shaft orbit shown in Figure A.3 and assume that there are two transducers, A and B, mounted 90° apart, which are used to measure the shaft vibration. At some instant, the shaft centre will be coincident with the point K on the orbit and the corresponding instantaneous value of shaft displacement from the mean position will be S_1. However, in the direction of the transducers, A and B, the instantaneous values of shaft displacement from the mean position will be S_{A1} and S_{B1}, respectively. The relationship between these quantities is given by Formula (A.7):

$$S_1^2 = S_{A1}^2 + S_{B1}^2 \tag{A.7}$$

The values of S_1, S_{A1} and S_{B1} vary with time as the shaft centre moves around the orbit; the corresponding waveforms measured by each transducer are shown in Figure A.3.

NOTE   If the orbit is elliptical, then these waveforms are pure sine waves of the same frequency.

The peak-to-peak value of the displacement in the direction of transducer A, $S_{A(p-p)}$, is defined as the difference between the maximum and minimum displacements of transducer A and similarly for S_B for transducer B. Clearly, $S_{A(p-p)}$ and $S_{B(p-p)}$ values will not be equal and, in general, they will be different from similar measurements made in other radial directions. Hence, the value of the peak-to-peak displacement is dependent on the direction of the measurement.

Since these measurement quantities are independent of the absolute value of the mean position, it is not necessary to use systems which can measure both the mean and alternating values.

Peak-to-peak displacement is the quantity which has been used most frequently for monitoring vibration of rotating machines.

Whereas measurement of the peak-to-peak displacement in any two given orthogonal directions is a simple matter, the amount and angular position of the maximum peak-to-peak displacement shown in Figure A.3 is difficult to measure directly. However, in practice, it has been found acceptable to use alternative measurement quantities which enable a suitable approximation for the maximum peak-to-peak displacement value to be obtained. For more precise determinations, it is necessary to examine the shaft orbit in more detail, for example, with an oscilloscope. The three most common methods for obtaining satisfactory approximations are described in A.2.2.2 to A.2.2.4.

A.2.2.2 Method A: Resultant value of peak-to-peak shaft displacement values measured in two orthogonal directions

The value of $S_{(p-p)max}$ can be approximated from Formula (A.8):

$$S_{(p-p)max} = \sqrt{S_{A(p-p)}^2 + S_{B(p-p)}^2} \tag{A.8}$$

The use of Formula (A.8) as an approximation when the vibration is predominantly at rotational frequency generally over-estimates the value of $S_{(p-p)max}$ with a maximum error of approximately 40 %.

The maximum error occurs for a circular orbit and progressively reduces as the orbit becomes flatter, with a zero error for the degenerate case of a straight line orbit.

A.2.2.3 Method B: Taking the maximum value of peak-to-peak shaft displacement values measured in two orthogonal directions

The value of $S_{(p-p)max}$ can be approximated from Formula (A.9):

$$S_{(p-p)max} = S_{A(p-p)} \text{ or } S_{B(p-p)} \tag{A.9}$$

whichever is greater.

The use of Formula (A.9) as an approximation when the vibration is predominantly at rotational frequency generally under-estimates the value of $S_{(p-p)max}$, with a maximum error of approximately 30 %.

The maximum error occurs for a flat orbit and progressively reduces as the orbit becomes circular, with a zero error when the orbit is circular.

A.2.2.4 Method C: Measurement of S_{max}

The instantaneous value of the shaft displacement can be defined by S_1, as shown in Figure A.3, which is derived from the transducer measurements, S_{A1} and S_{B1}, using Formula (A.7). There is a point on the orbit, defined by P in Figure A.3, where the displacement from the mean position 0 is a maximum. The value of S_1 corresponding to this position is denoted by S_{max} which is defined as the maximum value of displacement according to Formula (A.10):

$$S_{max} = \left[S_1\left(t\right) \right]_{max} = \left[\sqrt{S_{A1}^2\left(t\right) + S_{B1}^2\left(t\right)} \right]_{max} \tag{A.10}$$

The point on the orbit where S_{max} occurs does not necessarily coincide with the points where S_{A1} and S_{B1} are at their maximum values. Clearly, for a particular orbit, there is one value of S_{max} and this is independent of the position of the measuring transducers provided that the mean position 0 does not change.

The value of $S_{(p-p)max}$ can be approximated from Formula (A.11):

$$S_{(p-p)max} = 2\,S_{max} \tag{A.11}$$

Formula (A.11) is correct when the two orthogonal measurements from which S_{max} is derived are of single-frequency sinusoidal form. In most other cases, this formula over-estimates $S_{(p-p)max}$ since this depends on the nature of the harmonic vibration components present.

It should be noted that implicit in the definition of S_{max} is the requirement to know the time-integrated mean value of the shaft displacement. The measurement of S_{max} is, therefore, limited to those measuring systems which can measure both the mean and alternating values. Furthermore, the evaluation of S_{max} from the signals produced by two vibration transducers is a relatively complex computational procedure requiring specialized instrumentation.

Annex B
(informative)

Techniques for detection of problems in rolling element bearings

B.1 General

The use of a simple broad-band measuring technique on the raw acceleration data from a rolling element bearing housing, as described in this document, often provides sufficient information to give guidance on the running conditions of that particular bearing. It is realized that this simple technique is not successful in all circumstances. In particular, errors in assessment can arise if there are significant resonance effects in the bearing or its housing within the measurement frequency range, or if significant vibration signals are transmitted to the bearing from other sources such as gear-meshing vibration.

Mainly as a result of the above deficiencies, alternative measuring equipment and various analysis techniques have evolved which, in some instances, can be more suitable for identifying problems in rolling element bearings. None of these available instruments or techniques has, however, been successfully proven in all situations. For instance, not all types of bearing defects can be identified by any one technique and, whereas a particular technique can be perfectly satisfactory in identifying major bearing problems on one machine, it can be totally unsuitable for other installations. In all cases, the general vibration characteristics and patterns are mainly dependent on the specific type of bearing, the structure incorporating it, the instrumentation and even the signal processing. All of these phenomena need to be well understood; otherwise, no objective bearing evaluation method can apply.

NOTE   Information on measurement of structure-borne sound of rolling element bearings in machines for evaluation of rolling element condition is given in VDI 3832, whereas measuring methods for rolling element vibration are described in ISO 15242.

Selection of suitable techniques for specific applications requires specialist knowledge of both the technique and the machinery to which it is to be applied.

B.2 to B.5 briefly mention some of the available measuring equipment and analysis techniques which have been shown to have had some success in selected applications. However, insufficient information is available on suitable evaluation criteria values.

B.2 Raw data analysis (overall vibration measurements)

Various claims have been made in support of simple alternatives to the measurement of broad-band r.m.s. acceleration of the raw vibration signal for revealing defects in rolling element bearings.

These alternatives are as follows:

a)   measurement of peak acceleration values;

b)   measurement of the peak-to-r.m.s. ratio (crest factor) of the acceleration;

c)   computation of the product of r.m.s. and peak acceleration measurements.

B.3 Frequency analysis

The individual frequency components of a complex vibration signal can be identified with a variety of filtering arrangements or by spectrum analysis. If sufficient data are available about the particular bearing, its characteristic frequencies for a variety of defects can be calculated and compared with

the frequency components of the vibration signal. This, therefore, can give not only recognition that a bearing is giving concern, but can also identify the nature of the defect.

To give greater definition of the bearing-related frequencies in cases where high background vibration exists, processing techniques such as coherent averaging, adaptive noise cancellation or spectral subtraction techniques can be beneficially applied. A further technique is the spectral analysis of envelope waveforms which are generated by rectifying and smoothing of high-pass filtered vibration signals (or bandpass filtered in the high-frequency range). Thus, low-frequency background vibration is suppressed and the sensitivity for repetitive small pulses is significantly increased.

A useful variant to the spectral analysis approach is to consider sidebands (sum and difference frequencies) of the fundamental bearing characteristic frequencies rather than the fundamentals themselves. Although mainly used for detecting gear-meshing defects, cepstrum analysis (defined as "the power spectrum of the logarithm of the power spectrum") can be applied to identify sideband effects.

B.4 Shock-pulse techniques

A number of commercial instruments are available which rely on the fact that defects in rolling element bearings generate short pulses, usually called shock pulses.

Because of the sharpness of the shock pulses, they contain components at very high frequency. The instruments detect these high-frequency components and process them using proprietary techniques to form a value which can be related to the bearing condition.

An alternative technique is the spectral analysis of the raw shock-pulse envelope.

B.5 Alternative techniques

There are several techniques available which allow problems in rolling element bearings to be revealed in isolation of any vibration measurement. These include acoustic noise analysis, thermography and wear debris analysis (ferrography), but none can be claimed to be successful in all cases or even applicable in some instances.