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ISO 20816-4

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Mechanical vibration — Measurement and evaluation of machine vibration —

Part 4: Gas turbines in excess of 3 MW, with fluid-film bearings

Vibrations mécaniques — Mesurage et évaluation des vibrations de machines —

Partie 4: Turbines à gaz à paliers à film fluide, excédant 3 MW

Reference number
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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).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation on the voluntary nature of standards, 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.

This document was prepared by Technical Committee 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 20816-4 cancels and replaces ISO 7919-4:2009 and ISO 10816-4:2009, which have been technically revised. It also incorporates the Amendments ISO 7919-4/Amd.1:2017 and ISO 10816-4/Amd.1:2017.

The main change is that the scope has been reduced to exclude large gas turbines with power outputs greater than 40 MW, fluid-film bearings and rated speeds of 1 500 r/min, 1 800 r/min, 3 000 r/min or 3 600 r/min. Such gas turbines are now covered by ISO 20816-2.

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

Introduction

Documents in the ISO 20816 series have been and are being developed to combine and supersede the ISO 7919 and ISO 10816 series.

ISO 20816-1 is the basic part of the ISO 20816 series that gives the general requirements for evaluating the vibration of various machine types when the vibration measurements are made on both non-rotating and rotating parts. ISO 20816-2 deals with the measurement and evaluation of machine vibration of large gas turbines with certain rotational speeds.

This document provides specific provisions for assessing the vibration of the bearing housings or pedestals and rotating shafts of those gas turbines which are not covered by ISO 20816-2. Measurements at these locations characterize the state of vibration reasonably well. Evaluation criteria, based on previous experience, are presented. These can be used for assessing the vibratory condition of such machines. In those cases where there is a high ratio between the mass of the bearing supports and the rotor, lower values of vibration of the bearing housings or pedestals can be appropriate.

Two criteria are provided for assessing the machine vibration when operating under steady-state conditions. One criterion considers the magnitude of the observed vibration; the second considers changes in the magnitude. In addition, different criteria are provided for transient operating conditions.

The evaluation procedures presented in this document are based on broad-band measurements. However, because of advances in technology, the use of narrow-band measurements or spectral analysis has become increasingly widespread, particularly for the purposes of vibration evaluation, condition monitoring and diagnostics. The specification of criteria for such measurements is beyond the scope of this document. They are provided in greater detail in the relevant parts of ISO 13373 which establish provisions for the vibration condition monitoring of machines.

Mechanical vibration — Measurement and evaluation of machine vibration —

Part 4: Gas turbines in excess of 3 MW, with fluid-film bearings

1 Scope

This document is applicable to land-based gas turbines with fluid-film bearings and power outputs greater than 3 MW and an operating speed under load between 3 000 r/min and 30 000 r/min. In some cases (see the list of exclusions below), this includes other rotating machinery coupled either directly or through a gearbox. The evaluation criteria provided in this document are applicable to the vibration of the main input and output bearings of the gearbox but are not applicable to the vibration of the internal gearbox bearings nor to the assessment of the condition of those gears. Specialist techniques required for evaluating the condition of gears are outside the scope of this document.

This document is not applicable to the following:

i) gas turbines with power outputs greater than 40 MW at rated speeds of 1 500 r/min, 1 800 r/min, 3 000 r/min or 3 600 r/min (see ISO 20816-2);

ii) aero-derivative gas turbines (including gas turbines with dynamic properties similar to those of aero-derivatives);

NOTE ISO 3977-3 defines aero-derivatives as aircraft propulsion gas generators adapted to drive mechanical, electrical or marine propulsion equipment. Large differences exist between heavy-duty and aero-derivative gas turbines, for example, in casing flexibility, bearing design, rotor-to-stator mass ratio and mounting structure. Different criteria, therefore, apply for these two turbine types.

iii) gas turbines with outputs less than or equal to 3 MW (see ISO 7919-3 and ISO 10816-3);

iv) turbine driven generators (see ISO 20816-2, ISO 7919-3 and ISO 10816-3);

v) turbine driven pumps (see ISO 10816-7);

vi) turbine driven rotary compressors (see ISO 7919-3 and ISO 10816-3);

vii) the evaluation of gearbox vibration (see this clause) but does not preclude monitoring of gearbox vibration;

viii) the evaluation of combustion vibration but does not preclude monitoring of combustion vibration;

ix) rolling element bearing vibration.

This document establishes provisions for evaluating the severity of the following *in-situ* broad-band vibrations:

a) structural vibration at all main bearing housings or pedestals measured radial (i.e. transverse) to the shaft axis;

b) structural vibration at thrust bearing housings measured in the axial direction;

c) vibration of rotating shafts radial (i.e. transverse) to the shaft axis at, or close to, the main bearings.

These are in terms of the following:

— vibration under normal steady-state operating conditions;

— vibration during other (non-steady-state) conditions when transient changes are taking place, including run up or run down, initial loading and load changes;

— changes in vibration which can occur during normal steady-state operation.

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 20816-1:2016, *Mechanical vibration — Measurement and evaluation of machine vibration — Part 1: General guidelines*

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:

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

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

4 Measurement procedures

4.1 General

The measurement procedures and instrumentation shall comply with the general requirements given in ISO 20816-1.

ISO 20816-1, as well as this document, cover vibration of non-rotating parts and rotating shafts. However, this does not mean that both sets of measurements shall be taken on any particular machine. The choice of whether to measure vibration of non-rotating parts, rotating shafts or a combination of both is dependent on the particular application and shall always be agreed between the machine supplier and purchaser prior to installation. As a general guidance, it should be noted that bearing cap vibration is more sensitive where the rotor mass is large and the pedestal is flexible, whereas relative shaft vibration is more sensitive where the pedestal is large and stiff, and the rotor light and flexible.

The characteristics of the measurement system should be known with regard to the effects of the environment. Care should be taken to ensure that the measurement equipment is not unduly influenced by external sources, including

a) temperature variations,

b) electromagnetic fields,

c) airborne and structure-borne noise, e.g. from gear mesh vibration or combustion vibration or neighbouring machines,

d) transducer power source variations,

e) cable impedance,

f) transducer cable length,

g) transducer orientation, and

h) structural characteristics of the transducer attachment.

In special cases where significant low-frequency vibration can be transmitted to the machine, such as in earthquake regions, it can be necessary to filter the low-frequency response of the instrumentation.

If measurements at different times or from different machines are compared, care should be taken to ensure that the same frequency range is used and the data should be taken with the machine operating under stable conditions at the same rotational speed and load.

4.2 Measurements of vibration of non-rotating parts

For monitoring purposes, the measurement system shall be capable of measuring broad-band vibration over a frequency range from 10 Hz to at least three times the maximum normal operating frequency or 500 Hz, whichever is greater. If, however, the instrumentation is also used for diagnostic purposes, a wider frequency range and/or spectral analysis can be necessary. For example, in cases where the frequency corresponding to the first resonance speed (critical speed) of the coupled rotors is below 10 Hz, the lower limit of the linear range of the measurement system shall be reduced accordingly.

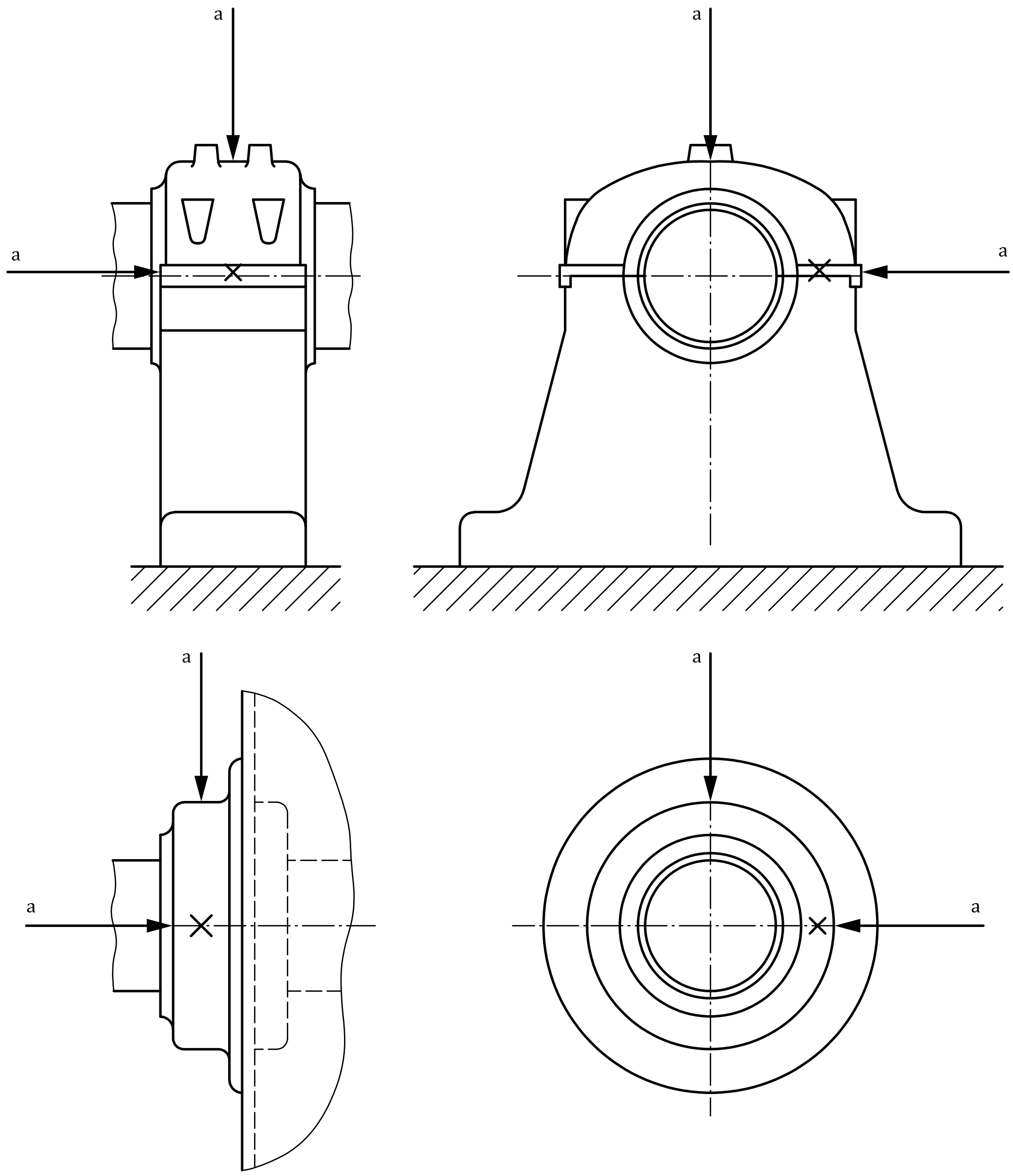
If the measurement is carried out using a velocity transducer and measurements below 10 Hz are required, it is important to linearize the velocity signal. This is particularly important when evaluating vibration velocity at lower speeds (see ISO 2954).

The locations of vibration measurements should be such that they provide adequate sensitivity to the dynamic forces of the machine. Typically, it involves measuring in two radial directions on each main bearing cap or pedestal with a pair of orthogonal transducers, as shown in Figure 1 and Figure 2. The transducers may be placed at any angular location on the bearing housings or pedestals, although vertical and horizontal directions (i.e. principal stiffness directions) are usually preferred.

A single radial transducer may be used on a bearing cap or pedestal in place of the more typical pair of orthogonal transducers if it is known to provide adequate information on the magnitude of the machine vibration. In general, however, caution should be observed when evaluating vibration from a single transducer at a measurement plane, since it might not be oriented to provide a reasonable approximation of the maximum value at that plane.

It is not common practice to measure axial vibration on the main radial load carrying bearings for continuous operational monitoring. Such measurements are primarily used during periodic vibration surveys or for diagnostic purposes. Hence, in this document, axial vibration criteria are only provided for thrust bearings where the vibration severity can be judged using the same criteria as for radial vibration (see Table A.1). For other bearings, where there are no axial restraints, a less stringent requirement may be used for the evaluation of axial vibration, provided that ancillary pipework and equipment are not adversely affected.

Particular attention should be given to ensuring that the vibration transducers are correctly mounted and that the mounting arrangement does not degrade the accuracy of the measurement (see e.g. ISO 2954 and ISO 5348).

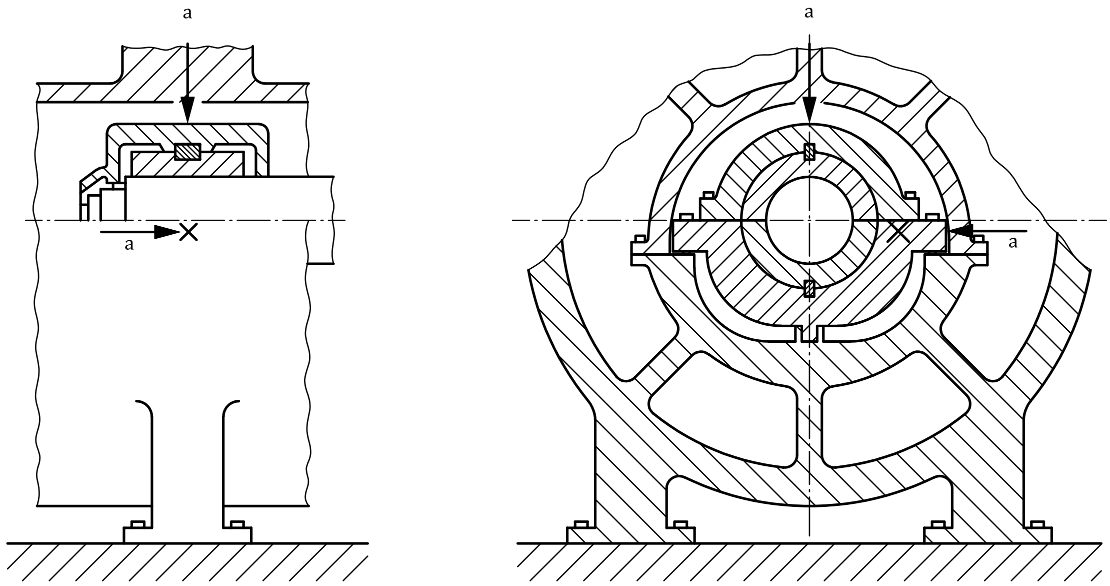


Key

a Direction of measurement.

NOTE The evaluation criteria in this document are applicable to radial vibration on all main bearings and to axial vibration on thrust bearings.

Figure 1 — Typical measuring points and directions on bearing pedestals and bearing caps



Key

a Direction of measurement.

NOTE The evaluation criteria in this document are applicable to radial vibration on all main bearings and to axial vibration on thrust bearings.

Figure 2 — Typical measuring points and directions on a gas turbine bearing

4.3 Measurements of vibration of rotating shafts

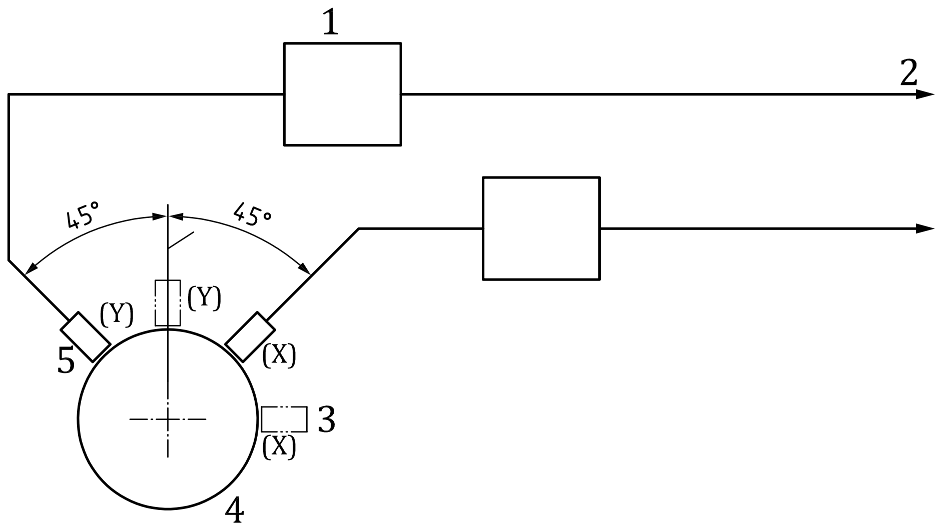
Shaft relative vibration (i.e. the vibration of the rotating shaft relative to the supporting structure) measurements are the preferred measurement quantity for shaft vibration of gas turbines.

For monitoring purposes, the measurement system shall be capable of measuring broad-band vibration over a frequency range from 1 Hz to at least three times the maximum normal operating frequency or 500 Hz, whichever is greater. If, however, the instrumentation is also used for diagnostic purposes, a wider frequency range (e.g. up to six times the maximum normal operating frequency) and/or spectral analysis can be necessary.

The locations of vibration measurements should be such that the transverse movement of the shaft at points of importance can be assessed. Typically, it involves measuring in two radial directions with a pair of orthogonal transducers at, or adjacent to, each main bearing. The transducers may be placed at any angular location, but it is common practice to select locations on the same bearing half which are either at ±45° to the vertical direction (top dead centre 12 o'clock position) or close to the vertical and horizontal directions (see Figure 3).

A single radial transducer may be used in place of the more typical pair of orthogonal transducers, if it is known to provide adequate information on the magnitude of the shaft vibration. In general, however, caution should be exercised when evaluating vibration from a single transducer at a measurement plane since it might not be oriented to provide a reasonable approximation of the maximum value at that plane.

It is not common practice to measure axial shaft vibration on gas turbines. Where measurements of axial position are made using non contacting transducers and are included within the vibration monitoring system, the assessment of the signal is not covered within this document.



Key

1 signal conditioning units

2 to signal processing

3 optional transducer orientations

4 shaft

5 non-contacting transducers

Figure 3 — Schematic diagram for measurement of relative motion of the shaft using non-contacting transducers

Particular attention shall be given to ensuring that the vibration transducers are correctly mounted and that the mounting arrangement does not degrade the accuracy of the measurement (see e.g. ISO 10817-1).

As far as is practically possible, the surface of the shaft at the location of the transducer shall be smooth and free from any geometric discontinuities, metallurgical non-homogeneities and local residual magnetism, which can cause false signals (so-called electrical runout). The combined electrical and mechanical "slow roll" runout, as measured by the transducer, should not exceed 25 % of the zone A/B boundary at rated speed (see Figure B.1).

Prior to running gas turbines up to rated speed, slow-roll measurements of shaft displacement may be carried out. If so, the low-frequency characteristics of the measurement system shall be adequate. Such measurements cannot normally be regarded as giving a valid indication of shaft runout under normal operating conditions, since they can be affected by, for example, temporary bows, erratic movements of the journal within the bearing clearance and axial movements. Subtraction of slow-roll measurements from rated speed vibration measurements should not be carried out without careful consideration of these factors, since the results can provide a misleading interpretation of the machine vibration (see ISO 20816-1).

5 Evaluation criteria

5.1 General

ISO 20816-1 provides a general description of the two evaluation criteria used to assess the vibration of various classes of machines.

a) One criterion considers the magnitude of the observed broad-band vibration;

b) the second criterion considers changes in magnitude, irrespective of whether they are increases or decreases.

The values presented are the result of experience with machinery of this type and, if due regard is paid to them, acceptable operation can be expected.

NOTE The values presented are based on previous International Standards, on the results of a survey which was carried out when the predecessor standards (see Foreword) were initially developed and on the feedback provided by users.

Criteria are presented for steady-state operating conditions at the specified rated speed and load ranges, including normal slow changes in power output. Alternative criteria are also presented for other non-steady-state conditions when transient changes are taking place. The vibration criteria represent target values which give provisions for ensuring that gross deficiencies or unrealistic requirements are avoided. In particular, the basic assumption for safe operation is that metal-to-metal contact between the rotating shaft and stationary components is avoided. They serve as a basis for defining acceptance test specifications (see 5.2.2.3).

The evaluation criteria relate to the vibration produced by the gas turbine and not to vibration transmitted from outside the machine. If it is suspected that there is a significant influence due to transmitted vibration (either steady-state or intermittent), measurements should be taken with the machine shut down. If the magnitude of the transmitted vibration is unacceptable, steps should be taken to remedy the situation.

5.2 Criterion I: vibration magnitude

5.2.1 General

This criterion is concerned with defining values for 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.

5.2.2 Vibration magnitude at rated speed under steady-state operating conditions

5.2.2.1 General

The maximum vibration magnitude observed at each measurement location is assessed against four evaluation zones.

5.2.2.2 Evaluation zones

The evaluation zones are defined to permit an assessment of the vibration of a given machine under steady-state conditions at rated speed and to provide guidelines on possible actions.

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

NOTE 1 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.

NOTE 2 For transient operation, see 5.2.4.

5.2.2.3 Acceptance criteria

Acceptance criteria should always be subject to agreement between the machine supplier and purchaser and prior negotiation is encouraged.

The evaluation zones provide a basis for defining acceptance criteria for new or refurbished machines but the numerical values assigned to the zone boundaries are not themselves intended to serve as acceptance specifications.

Historically, for new machines, acceptance criteria have been specified in zone A or zone B, but would normally not exceed 1,25 times the zone A/B boundary.

Different acceptance criteria can be agreed upon based on specific design characteristics and/or fleet experience with similar units.

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.

5.2.2.4 Evaluation zone boundaries

The zone boundary values are given in Annexes A and B. Annex A shall be used for radial vibration of non-rotating parts at all bearings and axial vibration of thrust bearing housings. Annex B shall be used for shaft relative vibration.

The zone boundaries apply to vibration measurements taken under steady-state conditions at rated speed. The numerical values assigned to the zone boundaries were established from representative data provided by manufacturers and users. There was inevitably a significant spread in the data but the values given do nevertheless give provisions for ensuring that gross deficiencies or unrealistic requirements are avoided.

Higher vibration is permitted at other measurement positions and during transient conditions (see 5.2.4).

In most cases, the values given in Annexes A and B are consistent with ensuring that the dynamic loads transmitted to the bearing support structure and foundation are acceptable and that running clearances are maintained. However, in certain cases, there can be specific features or available experience associated with a particular machine type which can require other values (higher or lower) to be used for the zone boundaries. Examples are given in a) to e).

a) The machine vibration can be influenced by its mounting system. For example, higher shaft relative vibration can be expected if stiff bearing supports are used. Conversely, for flexible bearing supports, lower shaft relative vibration can be expected. It can then be acceptable, based on demonstrated satisfactory operating history, to use different zone boundary values.

b) Lower vibration on non-rotating parts may apply where there is a high ratio (e.g. 10:1) between the structural mass associated with the bearing (e.g. pedestal and casing mass) and rotor mass. In such cases, it can be acceptable, based on satisfactory operating history, to use different zone boundary values.

c) Care should be taken to ensure that the shaft relative vibration does not indicate that the bearing clearance is exceeded. Furthermore, it should be recognized that the allowable vibration can be related to the journal diameter since, generally, running clearances are greater for larger diameter bearings. Where bearings with small clearance are used, the zone boundary values given in Annex B may be reduced. The degree to which the zone boundary values are to be reduced varies, dependent on the type of bearing used (e.g. circular, elliptical, tilting pad, etc.) and the relationship between the measurement direction and the minimum clearance. It is, therefore, not possible to give precise recommendations, but Annex E provides a representative example for a plain cylindrical bearing.

d) Other criteria based on the detailed machine design may be used for relatively lightly loaded bearings or other more flexible bearings whose static load is highly sensitive to rotor alignment.

e) Where shaft vibration measurements are made away from the bearing, other criteria may apply.

NOTE 1 This document does not provide different evaluation zone values for machines mounted on rigid and flexible foundations. However, this document might be revised in the future to give different criteria with respect to support flexibility, if additional analysis of survey data on such machines shows it to be warranted.

Different values can apply for measurements taken at different measurement locations on the same rotor line.

In general, when higher zone boundary values are used, it can be necessary to provide technical justification to confirm that the machine's reliability is not compromised by operating with higher vibration. This could be based, for example, on the detailed features of the machine or on successful operating experience with machines of similar structural design and support.

The common measurement parameter for assessing machine vibration on non-rotating parts is velocity. Annex A presents the evaluation zone boundaries based on broad-band RMS (root-mean-square) velocity measurements. In some cases, however, it was customary to measure vibration with instruments scaled to read peak rather than RMS vibration velocity values. If the vibration consists mainly of one frequency component (e.g. for gas turbines it is common for the vibration to be predominantly at the operating frequency of the machine), a simple relationship exists between the peak and RMS values. In such cases, the zone boundaries of Annex A can be readily expressed as zero-to-peak values by multiplying by $\sqrt{2}$. Alternatively, the measured peak vibration values can be divided by $\sqrt{2}$ and judged against the RMS criteria of Annex A. This approach cannot be used for complex waveforms with more than one significant frequency component.

NOTE 2 There is no general factor to relate the true peak value to the RMS value.

5.2.3 Operational limits for steady-state operation

5.2.3.1 General

For long-term steady-state operation at rated speed, it is common practice to establish operational vibration limits. These limits take the form of ALARMS and TRIPS.

ALARMS: To provide a warning that a defined vibration limit has been reached or a significant change has occurred, at which remedial action can be necessary. In general, if an ALARM occurs, operation can continue for a period while investigations are carried out (e.g. examine the influence of load, speed or other operational parameters) to identify the reason for the change in vibration and to define at which point any remedial action can be required.

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

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

5.2.3.2 Setting of ALARMS

The ALARM limits can vary for individual machines. The values chosen should normally be set relative to baseline values determined from experience for the measurement position or direction for that particular machine.

It is recommended that the ALARM limit be set higher than the baseline by an amount equal to 25 % of the zone boundary B/C. The ALARM limit should not normally exceed 1,25 times the zone boundary B/C. If the baseline value is low (i.e. less than 75 % of the zone boundary B/C), the ALARM limit can be less than the zone B/C boundary (see the example in Annex C).

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. In cases where no such data are available, the ALARM limit for steady-state operation at rated speed should not exceed the zone boundary B/C. After a period of time, the steady-state baseline values become established and the ALARM setting should be adjusted accordingly.

As explained in 5.2.2.4, suitable adjustments can be required to the shaft vibration ALARMS for units with small bearing and/or seal clearances (see Annex E).

Where the vibration signal is non-steady and non-repetitive, some method of averaging is required. Further detailed investigation of the cause of such behaviour is strongly recommended.

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

An example of establishing ALARM limits is given in Annex C.

5.2.3.3 Setting of TRIPS

The TRIP limits 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 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.

There can be differences for machines of different design and it is not possible to give more precise guidelines for absolute TRIP limits. In general, the TRIP limit is within zone C or D, but it should not exceed 1,25 times the zone boundary C/D. However, experience with a specific machine can prescribe a different limit.

As explained in 5.2.2.4, suitable adjustments can be required to the shaft vibration TRIPS for units with small bearing and/or seal clearances (see Annex E).

Gas turbines can be controlled by an automatic control system which shuts down the machine if the TRIP vibration limits are exceeded. In order to avoid unnecessary trips due to spurious signals, it is common practice to adopt a control logic using multiple transducers and to define a time delay before any automatic action is initiated to shut down the machine. Therefore, if a vibration TRIP signal is received, an action to proceed should only be acted upon if the signal is confirmed by at least two independent transducers and exceeds the defined limit for a specified finite delay time. Typically, a delay time in the range of 1 s to 3 s should be adequate to prevent premature tripping due to spurious signals while avoiding extreme damage due to prolonged exposure to high vibration. It might also be prudent to introduce a second alarm or warning to alert operators, so that corrective action (e.g. load reduction or other manufacturer's recommendations) can be taken to avoid tripping the unit.

Where the machine is subject to low-frequency vibration transmitted from an external source (e.g. in an earthquake zone), then it can be necessary to filter the signal and/or implement an appropriate time delay to avoid any nuisance trips.

5.2.4 Vibration magnitude during non-steady-state conditions (transient operation)

5.2.4.1 General

The vibration values given in Annexes A and B are specified with regard to the long-term operation of the gas turbine at the specified steady-state operating conditions. Higher vibration can be tolerated during the time that it takes for the gas turbine to reach thermal equilibrium when the operating conditions are changing at rated speed and during run up or run down. These higher values can exceed the steady-state ALARM and TRIP limits specified in 5.2.3. For such cases, a “trip multiplier” may be introduced which automatically raises the ALARM and TRIP limits for the period until steady-state conditions are established (see 5.2.4.4).

For gas turbines operating under non-steady-state conditions, such transient changes, which strongly influence the vibration behaviour, are generally associated with thermal variations (e.g. due to temperature changes during initial loading and load changes) and speed changes (e.g. run up, run down). Special design features are introduced to deal with such conditions, but it is inevitable that there are greater variations in the experienced vibration during speed changes (e.g. run up, run down) and while thermal changes are taking place (e.g. during start up, initial loading and load changes).

As with the steady-state vibration, any acceptance criteria for specific cases shall be subject to agreement between the machine supplier and purchaser. However, provisions are given in this clause which should ensure that gross deficiencies or unrealistic requirements are avoided.

For machines with synchronizing clutches, vibration step changes can occur due to normal variations in axial expansion and clutch engagement angle.

5.2.4.2 Vibration magnitude during transient operation at rated speed

This includes operation at no load, initial loading or during rapid load or power factor changes and any other operational conditions of relatively short duration. During such conditions, the vibration magnitude shall normally be considered to be acceptable provided it does not exceed the zone boundary C/D. The TRIP and ALARM limits should be adjusted accordingly.

5.2.4.3 Vibration magnitude during run up, run down and overspeed

The gas turbine shall have been adequately conditioned prior to running up to ensure that there are no temporary bends or bows present which would cause abnormal excitation. In particular, it is recommended that, where appropriate, a period of barring with the turning gear engaged and/or low-speed rotation be carried out before commencing to run up. Following this, if the gas turbine is fitted with shaft vibration transducers, slow-roll measurements may be carried out to assess the amount of shaft displacement obtained at low speed (where the measurements are not influenced by the lowest resonance speed), when stable bearing oil films have been established but centrifugal effects are negligible. The shaft displacement measured at this speed, together with other reference parameters, should be checked to be within previously established satisfactory experience. Such checks provide a basis for judging whether the state of the shaft line is satisfactory; for example, whether a temporary bend is present in the shaft or whether there is any lateral or angular misalignment between couplings ("crank effect"). Furthermore, during the run up, it is recommended that the vibration be assessed before a resonance speed is reached and compared with typical vibration vectors obtained under the same conditions during previous satisfactory runs. If any significant differences are observed, it can be advisable to take further action before proceeding (e.g. hold or reduce speed until the vibration stabilizes or returns to previous values, carry out a more detailed investigation or check operational parameters).

If there is no provision for barring or for measuring slow roll shaft displacement, observe alternative recommendations given by the supplier.

During run up, it can be necessary to hold at a particular speed (e.g. to allow temperature matching or during gas turbine purging). If so, care should be taken to ensure that there is an adequate margin between the hold speed and any resonance speeds where significant amplification of the vibration could occur. The specification of vibration limits during run up, run down and overspeed can vary depending on particular machine constructional features or the specific operational requirements. For example, higher vibration values can be acceptable for a base load production machine for which there is only a small number of starts, whereas more stringent limits can apply for a machine which undergoes regular two-shift operation and can be subject to specific time constraints for achieving guaranteed output levels. Furthermore, the vibration magnitude when passing through resonance speeds during run up and run down are strongly influenced by the damping and, to a lesser extent, by the rate of change of speed (for the sensitivity of machines to unbalance, see ISO 21940-31).

Different ALARM limits from those adopted for normal steady-state operating conditions apply during run up, run down and overspeed. They should normally be set relative to established values determined from experience during run up, run down or overspeed for the particular machine. It is recommended that the ALARM limit during run up, run down and overspeed be set above these values by an amount equal to 25 % of the zone boundary B/C for the rated speed.

In those cases where no reliable established data are available, the ALARM limit during run up, run down or overspeed should not exceed the values given in Table 1.

Table 1 — ALARM limit during run up, run down or overspeed

Speed range (in relation to rated speed)	**Vibration of non-rotating parts (see Table A.1)**	**Shaft relative vibration (see Formulae (B.1), (B.2), (B.3) and Figure B.1)**
<20 %	n/a (see Note)	1,5 × C/D boundary
20 % to 90 %	1,0 × C/D boundary	1,5 × C/D boundary
>90 %	1,0 × C/D boundary	1,0 × C/D boundary
NOTE The ratio of vibration displacement to velocity is inversely proportional to frequency. Hence, for measurements made on non-rotating parts, there are drawbacks in using a constant velocity criterion at speeds below 20 % of rated speed (see Annex D).		

As explained in 5.2.2.4, suitable adjustments can be required to the shaft vibration limits for bearings with small clearances (see Annex E).

Different approaches are used with regard to setting a TRIP during run up or run down. For example, if excessive vibrations build up during run up, it is possibly more appropriate to reduce speed rather than to initiate a TRIP. On the other hand, there is little point in initiating a high vibration TRIP during run down, since this does not change the action which has already been taken (i.e. to run down). However, if the gas turbine has an automatic control system, it can be necessary to define TRIP limits during run up or run down. In such cases, the TRIP limits during run up or run down should be derived by increasing the steady-state operational TRIP limits specified in 5.2.3.3 by the same proportion as for the ALARM limits.

NOTE During run up and run down, the highest vibration normally occurs when passing through resonance speeds due to dynamic magnification effects. At other speeds, lower vibration is normally expected.

5.2.4.4 Use of “trip multiplier”

In some cases, gas turbines are fitted with control systems which automatically shut down the turbine if the TRIP limits are exceeded. In order to avoid unnecessary trips when operating under transient conditions during which higher vibration is permitted, a “trip multiplier” may be introduced which automatically increases the steady-state ALARM and TRIP limits to reflect the revised values given in 5.2.4.2 and 5.2.4.3.

The “trip multiplier” would normally be activated during acceleration/deceleration of the rotor to/from rated speed (but not during any hold speed) and, if appropriate, during the initial loading transient after reaching rated speed, and for short periods after any sudden, substantial load changes to allow the thermal conditions to stabilize. Different “trip multiplier” settings, based on established experience, can apply for each of these operational conditions. The actual “trip multiplier” values vary for different machines and should be based on previous satisfactory operational experience.

5.3 Criterion II: change in vibration magnitude under steady-state conditions at rated speed

This criterion provides an assessment of a change in vibration magnitude from a previously established reference value for particular steady-state conditions. 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 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 at rated speed. This includes small changes in variables such as power output, but does not include large, rapid changes in output, which are dealt with in 5.2.4.2.

For machines with synchronizing clutches, vibration step changes can occur due to normal variations in axial expansion and clutch engagement angle.

The reference value for this criterion is the typical, reproducible normal vibration, known from previous measurements for the specific operating conditions. If the vibration magnitude changes by a significant amount (typically 25 % of the zone boundary B/C but other values may be used based on experience with a specific machine), steps should be taken to ascertain the reasons for the change. Such action should be taken regardless of whether the change causes an increase or decrease in the vibration magnitude. A decision on what action to take, if any, should be made after consideration of the maximum value of vibration and whether the machine has stabilized at a new condition. In particular, if the rate of change of vibration is significant, action should be taken even though the limit defined above has not been exceeded.

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.

It should be appreciated that a criterion based on change of vibration has limited application, since significant changes of varying magnitude and rates can and do occur in individual frequency components, but the importance of these is not necessarily reflected in the broad-band vibration signal (see ISO 20816-1). For example, the propagation of a crack in a rotor can introduce a progressive change in vibration at rotational frequency and components at multiples of rotational frequency, but their magnitude can be small relative to the amplitude of the once-per-revolution rotational frequency component. Consequently, it can be difficult to identify the effects of the crack propagation by looking at the change in the broad-band vibration only. Therefore, although monitoring the change in broad-band vibration gives some indication of potential problems, it can be necessary in certain applications to use measurement and analysis equipment which is capable of determining the trends of the vibration vector changes which occur for individual frequency components. This equipment can be more sophisticated than that used for normal supervisory monitoring and its use and application requires specialist knowledge. The specification of detailed criteria for measurements of this type is beyond the scope of this document (see 5.5).

5.4 Supplementary procedures/criteria

There is no simple way to relate bearing housing vibration to shaft vibration or vice versa. The difference between the shaft absolute and shaft relative measurements is related to the bearing housing vibration, but generally is not numerically equal to it because of the relative dynamic flexibility of the bearing oil film and support structure at the operating speed, the different positions at which the probes are mounted and the influence of phase angle differences. Thus, if vibration is measured on both non-rotating parts and on rotating shafts and the application of the different criteria leads to different assessments of vibration severity, the more restrictive zone classification generally applies, unless there is significant experience to the contrary.

5.5 Evaluation based on vibration vector information

The evaluation considered in this document is limited to broad-band vibration without reference to frequency components or phase. In most cases, this is adequate for acceptance testing and for operational monitoring purposes. However, for long-term condition monitoring purposes and for diagnostics, the use of vibration vector information is particularly useful for detecting and defining changes in the dynamic state of the machine. In some cases, these changes would go undetected when using only broad-band vibration measurements (see ISO 20816-1:2016, Annex D).

Phase and frequency related vibration information is commonly used for monitoring and diagnostic purposes. The specification of criteria for this, however, is beyond the scope of this document. They are dealt with in greater detail in the respective parts of ISO 13373, which give provisions for the vibration condition monitoring of machines.

Annex A
(normative)

Evaluation zone boundaries for vibration of non-rotating parts

The evaluation zone boundaries for vibration of non-rotating parts are given in Table A.1. The values given in Table A.1 apply to radial vibration measurements of all bearings and to axial vibration measurements of thrust bearings, when taken under steady-state operating conditions at rated speed for the frequency range specified in 4.2. Figures 1 and 2 show typical measurement positions. The values given are for ensuring that gross deficiencies or unrealistic requirements are avoided. In certain cases, specific features associated with a particular machine type can require the use of different zone boundary values (see 5.2.2.4). Higher vibration can be permitted at other measurement positions and under transient conditions (see 5.2.4).

NOTE Historically, acceptance criteria have been specified in zone A or zone B, but would normally not exceed 1,25 times the zone A/B boundary (see 5.2.2.3).

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.

Table A.1 — Values for bearing housing or pedestal RMS vibration velocity at zone boundaries

Zone boundary	RMS vibration velocity at zone boundaries mm/s
A/B	4,5
B/C	9,3
C/D	14,7

Annex B
(normative)

Evaluation zone boundaries for vibration of rotating shafts

In accordance with accumulated experience of shaft vibration measurements in this field, the recommended values for the peak-to-peak shaft relative vibration displacement at the zone boundaries $S_{(p-p)}$, in micrometres, for gas turbines with outputs greater than 3 MW are inversely proportional to the square root of the maximum normal operating speed n (r/min). The recommended values for such gas turbines are given by Formulae (B.1), (B.2) and (B.3) and are illustrated in Figure B.1. Generally, the actual value used should be rounded to the nearest multiple of 5 µm:

Zone boundary A/B

$$S_{(p-p)} = \frac{4\,800}{\sqrt{n}} \tag{B.1}$$

Zone boundary B/C

$$S_{(p-p)} = \frac{9\,000}{\sqrt{n}} \tag{B.2}$$

Zone boundary C/D

$$S_{(p-p)} = \frac{13\,200}{\sqrt{n}} \tag{B.3}$$

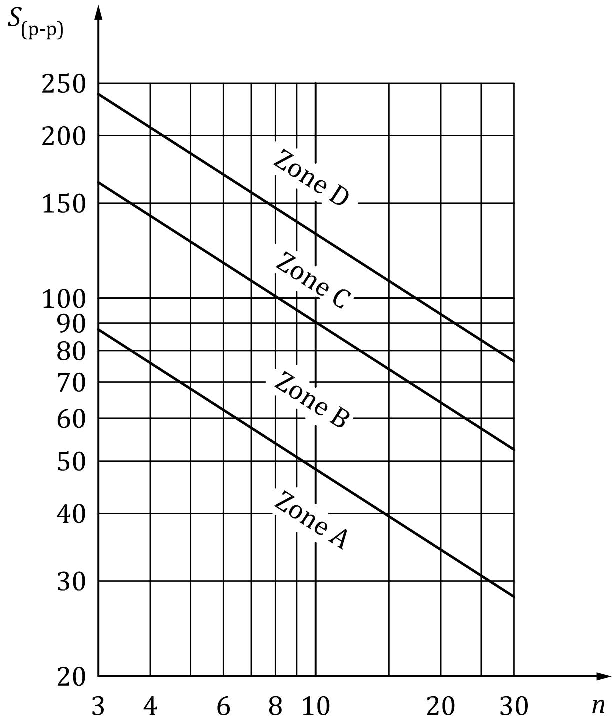
The values given by Formulae (B.1), (B.2) and (B.3) and in Figure B.1 apply to radial shaft vibration measurements at, or close to, the bearings, when taken under steady-state operating conditions at rated speed for the frequency range specified in 4.3. The values given are for ensuring that gross deficiencies or unrealistic requirements are avoided. In certain cases, specific features associated with a particular machine type can require the use of different zone boundary values (see 5.2.2.4). For example, care should be taken to ensure that the shaft relative vibration does not exceed the allowable bearing clearance (see Annex E). Higher vibration can be permitted at other measurement positions and under transient conditions (see 5.2.4).

The criteria given in this annex are stated in terms of the maximum peak-to-peak shaft vibration (see Method B of ISO 20816-1:2016, Annex A) at a particular measurement position. As a general guideline, if the outputs from a pair of orthogonal transducers at the measurement plane are used to derive S_{max} (see Method C of ISO 20816-1:2016, Annex A), the corresponding zone boundary values for S_{max} can be derived by dividing the values given in this annex by a factor of 1,85.

NOTE 1 Historically, acceptance criteria have been specified in zone A or zone B, but would normally not exceed 1,25 times the zone A/B boundary (see 5.2.2.3).

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.

NOTE 2 It is not common practice to measure shaft absolute vibration on gas turbines.



Key

n maximum normal operating speed × 1 000 (r/min)

$S_{(p-p)}$ peak-to-peak shaft relative vibration displacement (µm)

Figure B.1 — Recommended values for shaft relative vibration displacement at zone boundaries as a function of the maximum normal operating speed for gas turbines with power outputs greater than 3 MW

Annex C
(informative)

Example of setting ALARM and TRIP values

Consider the case of a 4 000 r/min gas turbine. The operational ALARM settings for a new machine, for which there is no prior knowledge of bearing vibration, is normally set within zone B. The specific value is often set by mutual agreement between the supplier and purchaser. For this example, assume it has been set initially for each bearing at the zone B/C boundary, which corresponds to an RMS value of 9,3 mm/s (see Table A.1).

After a period in service when the normal vibration characteristics have been established, consideration can be given to changing the ALARM limits to reflect the typical steady baseline values of vibration at each bearing. Using the procedure described in 5.2.3.2 as the basis, the ALARM may be set for each bearing to equal the sum of the typical steady-state value obtained from experience with the specific machine, and 25 % of the zone B/C boundary. For example, if the typical steady-state RMS vibration at a particular bearing were 4,0 mm/s, a new ALARM limit of 6,3 mm/s (i.e. 4,0 mm/s + 0,25 × 9,3 mm/s) would be adopted that is within zone B. If, at another bearing, the typical steady-state RMS vibration were 7,2 mm/s, the new ALARM limit would be 9,5 mm/s. However, in view of the small difference between this and the initial value, it could be decided that the initial value (9,3 mm/s) would be retained.

The TRIP limit would remain at an RMS value of 14,7 mm/s according to Criterion I. The basis for this is that the TRIP limit is a fixed value corresponding to the maximum vibration to which the machine should be subjected.

During transient operation, the above limits may be increased as described in 5.2.4.

Annex D
(informative)

Cautionary notes about the use of vibration velocity criteria at low rotational speeds

For the reasons explained in this annex, the velocity criteria provided in this document should not be applied for low frequencies. For monitoring the vibration at lower speeds, more specialized instrumentation can be required to allow evaluation against other criteria, such as constant displacement, which are outside the scope of this document.

The rationale for using vibration velocity measured on non-rotating parts as a basis for characterizing the severity of machine vibration has been derived from field experience (e.g. the pioneering work of Rathbone in the 1930s, see Reference [13]), together with an understanding of fundamental mechanics. Based on this, it has been accepted for many years that vibration with the same RMS velocity in the frequency range 10 Hz to 1 000 Hz can generally be considered to be of equal severity. A particular advantage of this is that if vibration velocity is used as an evaluation parameter, the same assessment criteria can be used regardless of the frequency of vibration or the operating speed of the machine. Conversely, if displacement or acceleration were used for evaluation, the assessment criteria would vary with frequency because the relationship between vibration displacement and velocity is inversely proportional to frequency and that between acceleration and velocity is directly proportional to frequency.

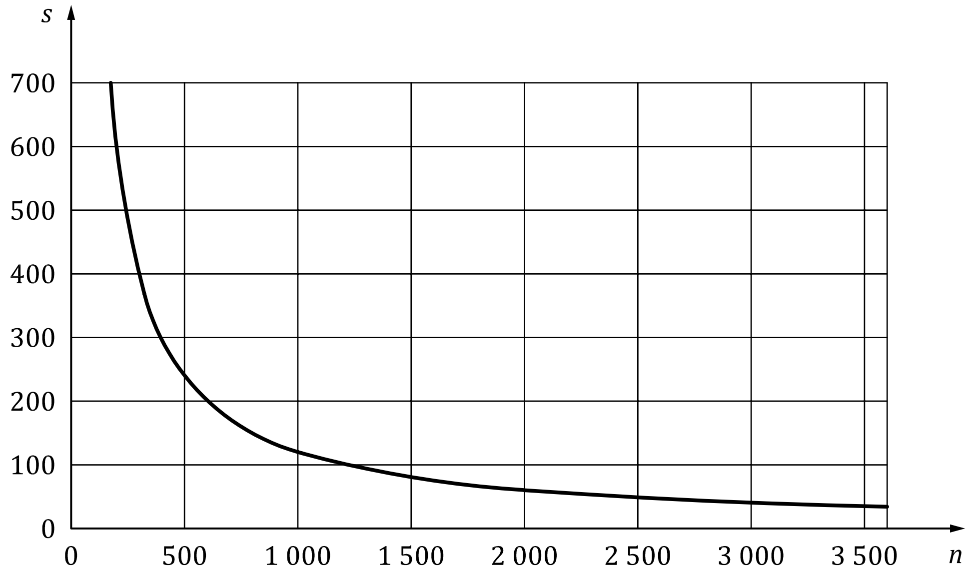
The use of a constant-velocity criterion alone becomes impractical at low and high frequencies where the influence of displacement and acceleration, respectively, becomes significant.

The ratio of vibration acceleration to velocity is directly proportional to frequency so that at high frequencies, the acceleration corresponding to a constant velocity increases linearly and becomes unacceptably large. However, for the gas turbines which are the subject of this document, the vibration frequencies are normally not high enough for acceleration to be of concern. Vibration associated with rolling element bearings or gear meshing fall into the category where criteria based on vibration acceleration are needed.

Conversely, the ratio of vibration displacement to velocity is inversely proportional to frequency so that at low frequencies, the displacement corresponding to a constant velocity increases rapidly. This is demonstrated in Figure D.1, which shows how the vibration displacement corresponding to a constant RMS velocity of 4,5 mm/s varies with speed for a once-per-revolution vibration frequency component (e.g. due to unbalance) during run down from 3 600 r/min.

Figure D.1 is simply a mathematical relationship which shows how a constant velocity transposes to displacement at different speeds and demonstrates how a constant velocity criterion can lead to a progressively increasing displacement of the bearing housing as the speed reduces. In such cases, although the dynamic forces transmitted to the bearing housing would be acceptable, the vibration displacements can be of concern at lower speeds for ancillary equipment attached to the bearing housing (e.g. oil pipes).

Figure D.1 should not be confused with a normal run-up or run-down response curve for which, apart from when passing through resonance speeds, the vibration velocity normally reduces as the speed is reduced. In practice, if the vibration velocity at rated speed is acceptable, it usually reduces at lower speeds and the corresponding vibration displacements at lower speeds would be acceptable. It follows that if significant vibration velocities are recorded at low speed during run up, even if they are below the values specified in this document and especially if they are significantly outside the range normally experienced at the same speed for that particular machine, action should be taken to understand the reasons for the higher vibration values and to establish whether it is safe to continue to higher speeds (see 5.2.4.3).



Key

n rotational speed (r/min)

s peak-to-peak vibration displacement (µm)

Figure D.1 — Variation of once-per-revolution vibration displacement component with speed for constant RMS vibration velocity of 4,5 mm/s

NOTE When using a velocity transducer and measurements below 10 Hz are required, it is important to linearize the velocity signal (see ISO 2954).

Annex E
(informative)

Evaluation zone boundary limits and bearing clearance

For machines supported by hydrodynamic bearings, the basic assumption for safe operation is that the shaft relative vibration displacement within the bearing oil film should be such that contact with the bearing is avoided. It should, therefore, be ensured that the shaft relative vibration limits for the evaluation zone boundaries given in Annex B are consistent with this assumption. In particular, where bearings with small clearances are used, it can be necessary to reduce the evaluation zone boundary values. The extent to which this is necessary is dependent on the type of bearing being used and the relationship between the measurement direction and the minimum clearance. A typical example is given in this annex.

Assume that a gas turbine with a rated speed of 4 000 r/min is supported by plain cylindrical bearings of 180 mm diameter and 0,1 % clearance ratio. In this case, the total (diametral) clearance of the bearing is 180 µm.

From Formulae (B.1), (B.2) and (B.3), the peak-to-peak zone boundary values for shaft relative vibration (rounded to the nearest multiple of 5 µm) are:

A/B 75 µm;

B/C 140 µm;

C/D 210 µm.

In this case, the B/C value is less than the bearing diametral clearance, but the C/D value is in excess of it. In such a case, it is recommended that the zone boundary limits be reduced, for example:

A/B 0,4 times bearing clearance = 72 µm (rounded up to 75 µm);

B/C 0,6 times bearing clearance = 108 µm (rounded up to 110 µm);

C/D 0,7 times bearing clearance = 126 µm (rounded up to 130 µm).

The factors 0,4, 0,6 and 0,7 have been chosen to illustrate the principle. Different factors, which should be agreed upon between the supplier and purchaser, apply for different bearing types.

The above example applies to the case where the shaft relative vibration is measured at, or very close to, the bearing oil film. Higher limits are acceptable at other measurement positions where the radial clearances are greater.

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