BRITISH STANDARD

BS ISO 7919-4:2009

Mechanical vibration — Evaluation of machine vibration by measurements on rotating shafts

Part 4: Gas turbine sets with fluid-film bearings

ICS 17.160; 27.040

British Standards

National foreword

This British Standard is the UK implementation of ISO 7919-4:2009. It supersedes BS ISO 7919-4:1996 which is withdrawn.

The UK participation in its preparation was entrusted to Technical Committee GME/21/5, Vibration of machines.

A list of organizations represented on this committee can be obtained on request to its secretary.

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Mechanical vibration — Evaluation of machine vibration by measurements on rotating shafts —

Part 4: **Gas turbine sets with fluid-film bearings**

Vibrations mécaniques — Évaluation des vibrations des machines par mesurages sur les arbres tournants —

Partie 4: Turbines à gaz à paliers à film fluide

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

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.

ISO 7919-4 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 second edition cancels and replaces the first edition (ISO 7919-4:1996), of which it constitutes a technical revision. The main changes are:

— clarification that the document applies only to gas turbine sets with fluid-film bearings;

— emphasis on acceptance specifications always being agreed on between the supplier and the purchaser of the gas turbine set prior to installation;

— the addition of provisions for evaluating the vibration of coupled gas turbine sets during transient operation;

— closer alignment of this part of ISO 7919 with ISO 7919-2, ISO 10816-2 and ISO 10816-4.

ISO 7919 consists of the following parts, under the general title *Mechanical vibration — Evaluation of machine vibration by measurements on rotating shafts*:

— *Part 1: General guidelines*[1)]

— *Part 2: Land-based steam turbines and generators in excess of 50 MW with normal operating speeds of 1 500 r/min, 1 800 r/min, 3 000 r/min and 3 600 r/min*

— *Part 3: Coupled industrial machines*

— *Part 4: Gas turbine sets with fluid-film bearings*

— *Part 5: Machine sets in hydraulic power generating and pumping plants*

1) It is anticipated that when ISO 7919-1 is revised, it will have the same general title as the other parts of ISO 7919.

Introduction

ISO 7919-1 is the basic part of ISO 7919 giving the general requirements for evaluating the vibration of various machine types when the vibration measurements are made on rotating shafts. This part of ISO 7919 gives specific provisions for assessing the severity of radial shaft vibration measured at, or close to, the bearings of gas turbine sets. 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.

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. However, shaft vibration does not form the only basis for judging the severity of vibration. For gas turbine sets, it is also common to judge the vibration based on measurements taken on non-rotating parts. For such vibration measurement requirements, see ISO 10816-1 and ISO 10816-4.

The evaluation procedures presented in this part of ISO 7919 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 part of ISO 7919. They are dealt with in greater detail in ISO 13373 (all parts), which establish provisions for the vibration condition monitoring of machines.

Mechanical vibration — Evaluation of machine vibration by measurements on rotating shafts —

Part 4: Gas turbine sets with fluid-film bearings

1 Scope

This part of ISO 7919 establishes provisions for evaluating the severity of *in-situ*, broad-band shaft vibration measured radial (i.e. transverse) to the shaft axis at, or close to, the main bearings. These are in terms of:

— 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.

This part of ISO 7919 is applicable to heavy-duty gas turbine sets used in electrical and mechanical drive applications, with fluid-film bearings, outputs greater than 3 MW and an operating speed range under load between 3 000 r/min and 30 000 r/min. This includes gas turbines coupled to other rotating machinery either directly or through a gearbox. In some cases, this part of ISO 7919 is not applicable to the evaluation of the vibration of the coupled equipment (see the list of exclusions in this clause).

EXAMPLE For single-shaft combined-cycle power units in which a gas turbine is coupled to a steam turbine and/or generator, the evaluation of the gas turbine vibration is according to this part of ISO 7919, but that of the steam turbine and generator is according to ISO 7919-2 or ISO 7919-3.

This part of ISO 7919 is not applicable to the following:

a) 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.

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

c) gas turbine driven pumps (see ISO 7919-3);

d) coupled steam turbines and/or generators with outputs less than or equal to 50 MW (see ISO 7919-3);

e) coupled steam turbines and/or generators with outputs greater than 50 MW (see ISO 7919-2);

f) synchronizing clutches which couple the gas turbine to a steam turbine or generator (see ISO 7919-2);

g) coupled compressors (see ISO 7919-3);

h) gearbox vibration (see this clause);

i) rolling element bearing vibration.

This part of ISO 7919 is applicable to other driven equipment not included in this list of exclusions.

This part of ISO 7919 is applicable to machines which can be coupled to a gearbox, but does not address the evaluation of the vibration condition of those gears. Specialist techniques are required for evaluating the vibration condition of gears which are outside the scope of this part of ISO 7919.

The numerical values specified are not intended to serve as the only basis for judging the severity of vibration. For gas turbine sets, it is also common to judge the vibration based on measurements taken on non-rotating parts. For such vibration measurement requirements, see ISO 10816-1 and ISO 10816-4.

2 Normative references

The following referenced documents are indispensable for the application 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 7919-1:1996, *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria — Part 1: General guidelines*

ISO 10816-4:2009, *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 4: Gas turbine sets with fluid-film bearings*

3 Measurement procedures

The measurement procedures and instrumentation shall comply with the general requirements of ISO 7919-1 and are as follows.

In gas turbine sets, shaft vibration relative to the bearing is normally measured. Therefore, unless stated otherwise, the vibration displacements referred to in this part of ISO 7919 conform to this convention. In view of the relatively high operating speeds involved with gas turbine sets, measuring methods using non-contacting transducers are most common and are generally preferred on rotors with operating speeds of 3 000 r/min and above.

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. If, however, the instrumentation is also used for diagnostic purposes, a wider frequency range and/or spectral analysis can be necessary. 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 and/or implement an appropriate time delay. If measurements from different machines are compared, care should be taken to ensure that the same frequency range is used.

The locations of vibration measurements should be such that the transverse movement of the shaft at points of importance can be assessed. Care should be taken to avoid locating measurement positions at any vibration nodes and to ensure that the measurement equipment is not unduly influenced by external sources, such as combustion vibration, gear mesh vibration, and airborne and structure-borne noise. Typically, this requires 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 or close to the vertical and horizontal directions.

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 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 shaft vibration on gas turbine sets.

The characteristics of the measurement system should be known with regard to the effects of the environment, including:

a) temperature variations;

b) magnetic fields;

c) airborne and structure-borne noise;

d) power source variations;

e) cable impedance;

f) transducer cable length;

g) transducer orientation;

h) stiffness of the transducer attachment.

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 10817-1).

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 normal operating speed (see Figure A.1 and Table A.1).

Prior to running gas turbine sets up to 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. Vector subtraction of slow-roll measurements from operating 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 7919-1).

4 Evaluation criteria

4.1 General

ISO 7919-1 provides a general description of the two evaluation criteria used to assess the shaft vibration on various classes of machines. One criterion considers the magnitude of the observed broad-band shaft vibration; 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 These values are based on previous International Standards, on the results of a survey which was carried out when ISO 7919 (all parts) and ISO 10816 (all parts) were initially developed and on the feedback provided by the experts of ISO/TC 108.

Criteria are presented for steady-state operating conditions at the specified normal operating speed or speeds 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 specifications (see 4.2.2.3).

The criteria relate to the vibration produced by the gas turbine set and not to vibration transmitted from outside the machinery set. 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 gas turbine set shut down. If the magnitude of the transmitted vibration is unacceptable, steps should be taken to remedy the situation.

It should be noted that an overall judgement of the vibratory state of a machine is often made on the basis of measurements made on both rotating shafts and non-rotating parts.

4.2 Criterion I: Vibration magnitude

4.2.1 General

This criterion is concerned with defining values 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.

4.2.2 Vibration magnitude at normal operating speeds under steady-state operating conditions

4.2.2.1 General

The maximum shaft vibration magnitude observed at each bearing is assessed against four evaluation zones established from international experience.

4.2.2.2 Evaluation zones

The following evaluation zones are defined to permit an assessment of the shaft vibration of a given machine under steady-state conditions at normal operating speed (or speeds) and to provide guidelines on possible actions.

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

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 For transient operation, see 4.2.4.

4.2.2.3 Acceptance criteria

Acceptance criteria shall always be subject to agreement between the machine supplier and purchaser prior to installation. The evaluation zones provide a basis for defining acceptance criteria for new or refurbished machines.

NOTE 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.

4.2.2.4 Evaluation zone boundaries

For gas turbines operating with directly coupled steam turbines and/or generators and normal operating speed of 3 000 r/min or 3 600 r/min, the zone boundary values are given in Table A.1.

In accordance with accumulated experience of shaft vibration measurements in this field, the recommended values for the zone boundaries, in micrometres, for other gas turbine sets with outputs greater than 3 MW are inversely proportional to the square root of the maximum normal operating speed n (in r/min). The recommended values for such gas turbines are given in Equations (1), (2) and (3) and illustrated in Figure A.1. Generally the actual value used should be rounded to the nearest multiple of 5 µm:

Zone boundary A/B

$$S_{(\text{p–p})} = \frac{4\,800}{\sqrt{n}} \tag{1}$$

Zone boundary B/C

$$S_{(\text{p–p})} = \frac{9\,000}{\sqrt{n}} \tag{2}$$

Zone boundary C/D

$$S_{(\text{p–p})} = \frac{13\,200}{\sqrt{n}} \tag{3}$$

NOTE 1 For a definition of $S_{(\text{p–p})}$, see ISO 7919-1.

The values given in Table A.1 and Figure A.1 apply to radial shaft relative vibration measurements at or close to the bearings, when taken under steady-state conditions at normal operating speed (or speeds). 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. 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 4.2.4).

In most cases, the values given in Table A.1 and Figure A.1 are consistent with ensuring that adequate running clearances are maintained and that the dynamic loads transmitted to the bearing support structure and foundation are acceptable. 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. The following are examples.

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

b) 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 Table A.1 and Figure A.1 may be reduced. The degree to which the zone boundary values are to be reduced varies, dependent on the type of bearing used (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 B provides a representative example for a plain cylindrical bearing.

c) For relatively lightly loaded bearings or other more flexible bearings, other criteria based on the detailed machine design may be used.

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

NOTE 2 Different values can apply for measurements taken at different bearings on the same rotor line.

In general, when higher zone boundary values are used, it can be necessary for technical justification to be provided 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.

NOTE 3 This part of ISO 7919 does not provide different evaluation zone values for gas turbine sets mounted on rigid and flexible foundations. This is consistent with ISO 10816-4, which deals with vibration on non-rotating parts for the same class of machines. However, it is possible that this part of ISO 7919 and ISO 10816-4 will 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.

4.2.3 Operational limits for steady-state operation

4.2.3.1 General

For long-term steady-state operation, 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 may be necessary. In general, if an ALARM occurs, operation can continue for a period whilst 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 any remedial action.

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 or the machine should be shut down.

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

4.2.3.2 Setting of ALARMS

The ALARM limits may vary for individual machines. It is recommended that the values chosen 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, the ALARM limit may be less than the zone B/C boundary.

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 normal operating 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.

Where the vibration signal is non-steady and non-repetitive, some method of averaging is required.

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

An example of establishing ALARM limits is given in ISO 10816-4:2009, Annex B.

4.2.3.3 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 is recommended that it not exceed 1,25 times the zone boundary C/D. However, experience with a specific machine may prescribe a different limit.

Gas turbine sets are often 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 automatically. 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, the delay time should be in the range of 1 s to 3 s. It might also be prudent to introduce a second ALARM between the ALARM and TRIP limits to alert operators that they are approaching the TRIP limit, so that they can take any corrective action (e.g. load reduction or other manufacturer's recommendations) to avoid tripping the unit from full load.

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

4.2.4.1 General

The vibration values given in Annex A 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 normal operating speed (or speeds) and during run up or run down. These higher values may exceed the steady-state ALARM and TRIP limits specified in 4.2.3. 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 4.2.4.4).

For gas turbines operating under non-steady-state conditions, such transient changes are generally associated with significant thermal variations, which strongly influence the vibration behaviour of the engine. Special design features are introduced to deal with such conditions, but it is inevitable for there to be a greater variation 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) than for other machine types, where more gradual vibration changes are experienced. Consequently higher transient vibrations are normally acceptable for gas turbines.

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.

4.2.4.2 Vibration magnitude during transient operation at normal operating 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.

4.2.4.3 Vibration magnitude during run up, run down and overspeed

The gas turbine set 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 and/or low-speed rotation be carried out before commencing to run up. Following this, slow-roll shaft displacement measurements may be carried out to assess the amount of runout 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 shaft vibration be assessed before a critical/resonant 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. during purging). If so, care should be taken to ensure that there is an adequate margin between the hold speed and any critical/resonant 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 may be acceptable for a base load unit for which there may be only a small number of starts, whereas more stringent limits can apply for a unit 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 critical/resonant speeds during run up and run down is strongly influenced by the damping and the rate of change of speed (for the sensitivity of machines to unbalance, see ISO 10814).

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 maximum normal operating 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 zone boundary C/D for the maximum normal operating speed.

As explained in 4.2.2.4, suitable adjustments may be required for bearings with small clearances (see Annex B).

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 (i.e. to run down) which has already been taken. 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 should be increased by a similar proportion to that adopted for the ALARM limits.

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

4.2.4.4 Use of "trip multiplier"

In some cases, gas turbine sets are fitted with control systems which automatically shut down the set 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 4.2.4.2 and 4.2.4.3.

The "trip multiplier" would normally be activated during acceleration/deceleration of the rotor to/from normal operating speed (but not during any hold speed) and, if appropriate, during the initial loading transient (normally the first few hours) after reaching normal operating 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, may apply for each of these operational conditions. The actual “trip multiplier” values vary for different gas turbines and should be based on previous satisfactory operational experience.

4.3 Criterion II: Change in vibration magnitude under steady-state conditions at normal operating 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 shaft 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 shaft vibration magnitude occurring under steady-state operating conditions at normal operating 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 4.2.4.2.

CAUTION — Care should be taken in applying this criterion for machines with synchronizing clutches where vibration step changes can occur due to normal variations in axial expansion.

The reference value for this criterion is the typical, reproducible normal vibration, known from previous measurements for the specific operating conditions. If the shaft vibration magnitude changes by a significant amount (typically 25 % of the zone boundary B/C), 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. At bearings with small clearances, the limiting values should be adjusted accordingly.

When Criterion II is applied, the vibration measurements being compared shall be taken at the same transducer location and orientation and, as far as practicable, under 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 shaft vibration signal (see ISO 7919-1). For example the propagation of a crack in a rotor can introduce a progressive change in vibration 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 part of ISO 7919 (see 4.5).

4.4 Supplementary procedures/criteria

The measurement and evaluation of vibration given in this part of ISO 7919 may be supplemented or replaced by measurements made on non-rotating parts (see ISO 10816-4). There is no simple way to relate shaft vibration to bearing housing 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, when the criteria of this part of ISO 7919 and those of ISO 10816-4 are both applied in the assessment of machine vibration, independent shaft and bearing housing (or pedestal) vibration measurements shall be made. If 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.

4.5 Evaluation based on vibration vector information

The evaluation considered in this part of ISO 7919 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 10816-1:1995, Annex D).

Phase- and frequency-related vibration information is being used increasingly for monitoring and diagnostic purposes. The specification of criteria for this, however, is beyond the scope of this part of ISO 7919. They are dealt with in greater detail in ISO 13373 (all parts), which give provisions for the vibration condition monitoring of machines.

Annex A
(normative)

Evaluation zone boundaries

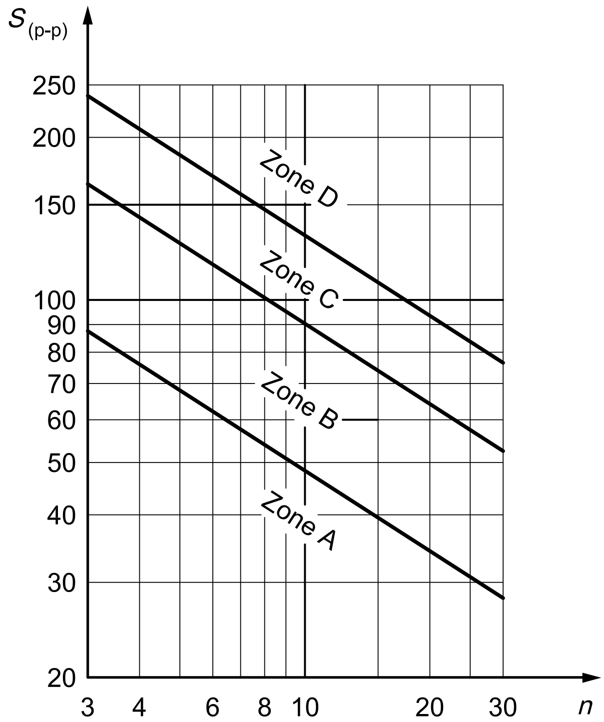
The values given in Table A.1 and Figure A.1 apply to radial shaft relative vibration measurements at, or close to, the bearings, when taken under steady-state operating conditions at normal operating speed. Table A.1 applies to gas turbine sets which are directly coupled to steam turbines and/or generators with normal operating speeds of 3 000 r/min or 3 600 r/min. Figure A.1 applies for other gas turbine sets. 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 4.2.2.4). For example care should be taken to ensure that the shaft relative vibration does not exceed the allowable bearing clearance (see Annex B). Higher vibration can be permitted at other measurement positions and under transient conditions (see 4.2.4).

The criteria given in Table A.1 and Figure A.1 are given in terms of the peak-to-peak shaft vibration at a particular measurement position (see Method B of ISO 7919-1:1996, Annex B). If the outputs from a pair of orthogonal transducers at the measurement plane are used to derive S_{max} (see Method C of ISO 7919-1:1996, Annex B), smaller zone boundary values should be used which are dependent on the shaft orbit. As a general guideline, the values given in Table A.1 and Figure A.1 should be divided by a factor of 1,85.

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 4.2.2.3).

Table A.1 — Recommended values for shaft relative vibration displacement at zone boundaries for gas turbines with directly coupled steam turbines and/or generators and normal operating speeds of 3 000 r/min or 3 600 r/min

Zone boundary	**Shaft rotational speed** r/min	
	3 000	3 600
	Peak-to-peak shaft relative vibration displacement at zone boundaries µm	
A/B	90	80
B/C	165	150
C/D	240	220



Key

n maximum normal operating speed × 1 000, r/min

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

For gas turbines with directly coupled steam turbines and/or generators and normal operating speeds of 3 000 r/min and 3 600 r/min, use the values given in Table A.1.

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 4.2.2.3).

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

Annex B
(informative)

Evaluation zone boundary limits and bearing clearance

For machines supported by hydrodynamic bearings, the basic assumption for safe operation is that the shaft 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 Table A.1 and Figure A.1 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 particular gas turbine set with a normal operating speed of 3 000 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 Table A.1, the peak-to-peak zone boundary values are:

A/B 90 µm;

B/C 165 µm;

C/D 240 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 on 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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