INTERNATIONAL STANDARD

ISO 7919-1

Second edition
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Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria —

Part 1:
General guidelines

Vibrations mécaniques des machines non alternatives — Mesurages sur les arbres tournants et critères d'évaluation —

Partie 1: Directives générales

Reference number
ISO 7919-1:1996(E)

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.

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.

International Standard ISO 7919-1 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

This second edition of ISO 7919-1 cancels and replaces the first edition (ISO 7919-1:1986), which has been technically revised.

ISO 7919 consists of the following parts, under the general title *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria*:

— *Part 1: General guidelines*

— *Part 2: Large land-based steam turbine generator sets*

— *Part 3: Coupled industrial machines*

— *Part 4: Gas turbine sets*

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

Annex A forms an integral part of this part of ISO 7919. Annexes B, C, D and E are for information only.

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Introduction

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

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

ISO 10816-1 establishes a basis for the evaluation of mechanical vibration of machines by measuring the vibration response on non-rotating, structural members only. There are many types of machine, however, for which measurements on structural members, such as the bearing housings, may not adequately characterize the running condition of the machine, although such measurements are useful. Such machines generally contain flexible rotor shaft systems, and changes in the vibration condition may be detected more decisively and more sensitively by measurements on the rotating elements. Machines having relatively stiff and/or heavy casings in comparison to rotor mass are typical of those classes of machines for which shaft vibration measurements are frequently to be preferred.

For machines such as steam turbines, gas turbines and turbo-compressors, all of which may have several modes of vibration in the service speed range, measurements on non-rotating parts may not be totally adequate. In such cases, it may be necessary to monitor the machine using measurements on the rotating and non-rotating parts, or on the rotating parts alone.

The guidelines presented in this part of ISO 7919 are complemented by those given in ISO 10816-1. If the procedures of both standards are applied, the one which is more restrictive generally applies.

Shaft vibration measurements are used for a number of purposes, ranging from routine operational monitoring and acceptance tests to advanced experimental testing, as well as diagnostic and analytical investigations. These various measurement objectives lead to many differences in methods of interpretation and evaluation. To limit the number of these differences, this part of ISO 7919 is designed to provide guidelines primarily for operational monitoring and acceptance tests.

During the preparation of this part of ISO 7919, it was recognized that there was a need to establish quantitative criteria for the evaluation of machinery shaft vibration. However, there is a significant lack of data on this subject at present and, consequently, this part of ISO 7919 has been structured to allow such data to be incorporated as it becomes available.

Specific criteria for different classes and types of machinery will be given in the relevant parts of ISO 7919 as they are developed.

Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria —

Part 1: General guidelines

1 Scope

This part of ISO 7919 sets out general guidelines for measuring and evaluating machinery vibration by means of measurements made directly on rotating shafts for the purpose of determining shaft vibration with regard to

a) changes in vibrational behaviour;

b) excessive kinetic load;

c) the monitoring of radial clearances.

It is applicable to measurements of both absolute and relative radial shaft vibration, but excludes torsional and axial shaft vibration. The procedures are applicable for both operational monitoring of machines and to acceptance testing on a test stand and after installation. Guidelines are also presented for setting operational limits.

NOTES

1 Evaluation criteria for different classes of machinery will be included in other parts of ISO 7919 when they become available. In the meantime, guidelines are given in annex A.

2 The term "shaft vibration" is used throughout ISO 7919 because, in most cases, measurements will be made on machine shafts; however, ISO 7919 is also applicable to measurements made on other rotating elements if such elements are found to be more suitable, provided that the guidelines are respected.

For the purposes of ISO 7919, operational monitoring is considered to be those vibration measurements made during the normal operation of a machine. ISO 7919 permits the use of several different measurement quantities and methods, provided that they are well defined and their limitations are set out, so that the interpretation of the measurements will be well understood.

This part of ISO 7919 does not apply to reciprocating machinery.

2 Normative reference

The following standard contains provisions which, through reference in this text, constitute provisions of this part of ISO 7919. At the time of publication, the edition indicated was valid. All standards are subject to revision, and parties to agreements based on this part of ISO 7919 are encouraged to investigate the possibility of applying the most recent edition of the standard indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 10816-1:1995, *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 1: General guidelines.*

3 Measurements

3.1 Measurement quantities

3.1.1 Displacement

The preferred measurement quantity for the measurement of shaft vibration is displacement. The unit of measurement is the micrometer ($1\ \mu m = 10^{-6}\ m$).

NOTE 3 Displacement is a vector quantity and, therefore, when comparing two displacements, it may be necessary to consider the phase angle between them (see also annex D).

Since this part of ISO 7919 applies to both relative and absolute shaft vibration measurements, displacement is further defined as follows:

a) relative displacement, which is the vibratory displacement between the shaft and appropriate structure, such as a bearing housing or machine casing; or

b) absolute displacement, which is the vibratory displacement of the shaft with reference to an inertial reference system.

NOTE 4 It should be clearly indicated whether displacement values are relative or absolute.

Absolute and relative displacements are further defined by several different displacement quantities, each of which is now in widespread use. These include:

$S_{(p-p)}$ vibratory displacement peak-to-peak in the direction of measurement;

S_{max} maximum vibratory displacement in the plane of measurement.

Either of these displacement quantities may be used for the measurement of shaft vibration. However, the quantities shall be clearly identified so as to ensure correct interpretation of the measurements in terms of the criteria of clause 5. The relationships between each of these quantities are shown in figures B.1 and B.2.

NOTE 5 At present, the greater of the two values for peak-to-peak displacement, as measured in two orthogonal directions, is used for evaluation criteria. In future, as relevant experience is accumulated, the quantity $S_{(p-p)max}$, defined in figure B.2, may be preferred.

3.1.2 Frequency range

The measurement of relative and absolute shaft vibration shall be broad band so that the frequency spectrum of the machine is adequately covered.

3.2 Types of measurement

3.2.1 Relative vibration measurements

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

3.2.2 Absolute vibration measurements

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

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

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

3.3 Measurement procedures

3.3.1 General

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

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

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

Recommendations for instrumentation are given in annex C.

3.3.2 Procedures for relative vibration measurements

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

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

3.3.3 Procedures for absolute vibration measurements using combined seismic and non-contacting relative vibration transducers

If a combination of seismic and non-contacting relative vibration transducers is used, the absolute vibration is obtained by vectorially summing the outputs from both transducers. The mounting and other requirements for the non-contacting transducer are as specified in 3.3.2. In addition, the seismic transducer shall be rigidly mounted to the machine structure (e.g. the bearing housing) close to the non-contacting transducer so that both transducers undergo the same absolute vibration of the support structure in the direction of measurement. The sensitive axes of the non-contacting and seismic transducers shall be parallel, so that their vectorially summed, conditioned signals result in an accurate measure of the absolute shaft vibration.

3.3.4 Procedures for absolute vibration measurements using a shaft-riding mechanism with a seismic transducer

The seismic transducer (velocity type or accelerometer) shall be mounted radially on the shaft-riding mechanism. The mechanism shall not chatter or bind in a manner modifying the indicated shaft vibration. The mechanism shall be mounted as described for transducers in 3.3.1.

The shaft surface against which the shaft-riding tip rides, taking into account the total axial float of the shaft under all thermal conditions, shall be smooth and free from shaft discontinuities, such as keyways and threads. It is recommended that the mechanical runout of the shaft should not exceed 25 % of the allowable vibration displacement, specified in accordance with annex A, or 6 µm, whichever is the greater.

There may be surface speed and/or other limitations to shaft-riding procedures, such as the formation of hydrodynamic oil films beneath the probe, which may give false readings and, consequently, manufacturers should be consulted about possible limitations.

3.4 Machine operating conditions

Shaft vibration measurements should be made under agreed conditions over the operating range of the machine. These measurements should be made after achieving agreed thermal and operating conditions. In addition, measurements may also be taken under conditions of, for example, slow roll, warming-up speed, critical speed, etc. However, the results of these measurements may not be suitable for evaluation in accordance with clause 5.

3.5 Machine foundation and structures

The type of machine foundation and structures (for example piping) may significantly affect the measured vibration. In general, a valid comparison of vibration values of machines of the same type can only be made if the foundations and structures have similar dynamic characteristics.

3.6 Environmental vibration and evaluation of measurement system

Prior to measuring the vibration of an operating machine, a check with the same measuring system and stations should be taken with the machine in an inoperative state. When the results of such measurements exceed one-third of the values specified for the operating speed, steps should be taken to eliminate environmental vibration effects.

4 Instrumentation

The instrumentation used for the purpose of compliance with this part of ISO 7919 shall be so designed as to take into account temperature, humidity, the presence of any corrosive atmosphere, shaft surface speed, shaft material and surface finish, operating medium (e.g. water, oil, air or steam) in contact with the transducer, vibration and shock (three major axes), airborne noise, magnetic fields, metallic masses in proximity to the tip of the transducer, and power-line voltage fluctuations and transients.

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

5 Evaluation criteria

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

a) absolute vibration of the shaft;

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

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

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

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

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

5.4 If the evaluation criterion is stator/rotor clearances, then

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

b) when the vibration of the structure, on which the shaft-relative transducer is mounted, is 20 % or more of the relative shaft vibration, the relative shaft vibration measurement may still be used as a measure of clearance absorption unless the vibration of the structure, on which the shaft-relative transducer is mounted, is not representative of the total stator vibration. In this latter case, special measurements will be required.

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

5.6 General principles for evaluation of shaft vibration on different machines are given in annex A. The evaluation criteria relate to both operational monitoring and acceptance testing, and apply only to the vibration produced by the machine itself and not to vibration transmitted from outside. For certain classes of machinery, the guidelines presented in this part of ISO 7919 are complemented by those given in ISO 10816-1 for measurements taken on non-rotating parts. If the procedures of both International Standards are applied, the one which is more restrictive shall generally apply.

Specific criteria for different classes and types of machinery will be given in the relevant parts of ISO 7919 as they are developed.

5.7 The evaluation considered in this basic document is limited to broad-band vibration without reference to frequency components or phase. This will in most cases be adequate for acceptance testing and operational monitoring purposes. However, in some cases the use of vector information for vibration assessment on certain machine types may be desirable. Vector change information is particularly useful in detecting and defining changes in the dynamic state of a machine, which in some cases could go undetected when using broad-band vibration measurements. This is demonstrated in annex D.

The specification of criteria for vector changes is beyond the present scope of this part of ISO 7919.

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

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

Annex A
(normative)

General principles for adopting evaluation criteria for different types of machine

Introduction

The specification of evaluation criteria for shaft vibration is dependent upon a wide range of factors and the criteria adopted will vary significantly for different types of machine and, in some cases, for different rotors in the same coupled line. It is important, therefore, to ensure that valid criteria are adopted for a particular machine and that criteria which relate to certain types of machine are not erroneously applied to other types. (For example, evaluation criteria for a high-speed compressor operating in a petrochemical plant are likely to be different from those for large turbo-generators.)

At present, there are a limited number of published standards on shaft vibration. Many of these are for specialized machinery and do not have widespread applications in other fields.

This annex establishes a basis for specifying evaluation criteria in terms of peak-to-peak vibration values (see annex B). No attempt has been made to specify vibration values; these will be given for different classes and types of machinery in the relevant parts of ISO 7919 as they are developed.

A.1 Factors affecting evaluation criteria

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

a) the purpose for which the measurement is made (for example, the requirements for ensuring that running clearances are maintained will, in general, be different from those if the avoidance of excessive kinetic load on the bearing is the main concern);

b) the type of measurement made — absolute or relative vibration;

c) the quantities measured (see annex B);

d) the position where the measurement is made;

e) the rotational frequency of the shaft;

f) the bearing type, clearance and diameter;

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

h) the relative flexibility of the bearings, pedestals and foundations;

i) the rotor mass and flexibility.

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

A.2 Evaluation criteria

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

A.2.1 Criterion I: Vibration magnitude at rated speed under steady operating conditions

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

Figure A.1 shows a plot of allowable vibration, in terms of peak-to-peak shaft vibration, against the operating speed range. It is generally accepted that limiting vibration values will decrease as the operating speed of the machine increases, but the actual values and their rate of change with speed will vary for different types of machine.

A.2.1.1 Evaluation zones

The following typical evaluation zones are defined to permit a qualitative assessment of the shaft vibration on a given machine and provide guidelines on possible actions.

Zone A: The vibration of newly commissioned machines would normally fall 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.

A.2.1.2 Evaluation zone limits

Numerical values assigned to the zone boundaries are not intended to serve as acceptance specifications, which shall be subject to agreement between the machine manufacturer and the customer. However, these values provide guidelines for ensuring that gross deficiencies or unrealistic requirements are avoided. In certain cases, there may be specific features associated with a particular machine which would require different zone boundary values (higher or lower) to be used. In such cases, it is normally necessary to explain the reasons for this and, in particular, to confirm that the machine will not be endangered by operating with higher vibration values.

A.2.2 Criterion II: Change in vibration magnitude

This criterion provides an assessment of a change in vibration magnitude from a previously established reference value. A significant increase or decrease in broad-band vibration magnitude may 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 may 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 broad-band vibration magnitude occurring under steady-state operating conditions.

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

Criteria for assessing changes in broad-band vibration for monitoring purposes are given in other parts of ISO 7919. However, it should be noted that some changes may not be detected unless the discrete frequency components are monitored (see 5.7).

A.2.3 Operational limits

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

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

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

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

Where appropriate, guidelines for specifying ALARM and TRIP criteria for specific machine types are given in other parts of ISO 7919.

A.2.3.1 Setting of ALARMS

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

It is recommended that the ALARM value should be set higher than the baseline by an amount equal to a proportion of the upper limit of zone B. If the baseline is low, the ALARM may be below zone C.

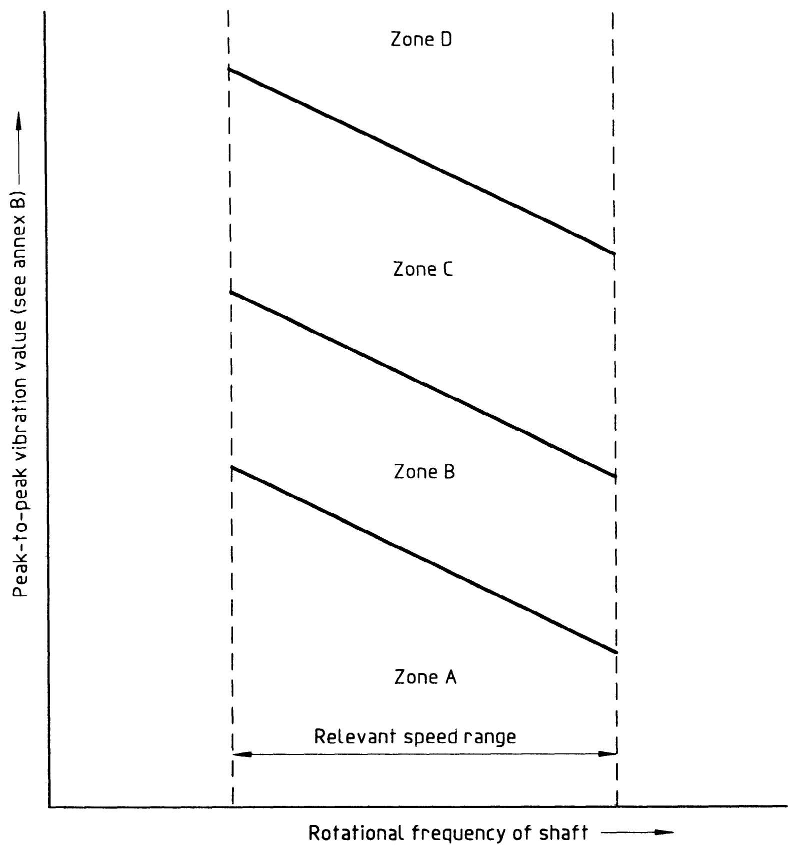
Where there is no established baseline, for example with a new machine, the initial ALARM setting should be based either on experience with other similar machines or relative to agreed acceptance values. After a period of time, the steady-state baseline value will be established and the ALARM setting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

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

A.2.3.2 Setting of TRIPS

The TRIP values will generally relate to the mechanical integrity of the machine and be dependent on any specific design features which have been introduced to enable the machine to withstand abnormal dynamic forces. The values used will, therefore, generally be 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 may, however, be differences for machines of different design and it is not possible to give guidelines for absolute TRIP values. In general, the TRIP value will be within zone C or D.



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

Figure A.1 — Generalized example of evaluation criteria

Annex B
(informative)

Derivation of measurement quantities

B.1 Mechanics of shaft vibration

The vibration of a rotating shaft is characterized at any axial location by a kinetic orbit, which describes how the position of the shaft centre varies with time. Figure B.1 shows a typical orbit. The shape of the orbit depends upon the dynamic characteristics of the shaft, the bearings and the bearing supports or foundations, the axial location on the rotor and the form of vibration excitation. For example, if the excitation takes the form of a single-frequency sinusoidal force, the orbit is an ellipse, which can in certain circumstances be a circle or straight line, and the time taken for the shaft centre to complete one circuit of the ellipse is equal to the period of the excitation force. One of the most important excitation forces is rotor unbalance, in which the excitation frequency is equal to the rotational frequency of the shaft. However, there are other forms of excitation, such as rotor cross-section asymmetry, for which the frequency is equal to multiples of the rotational frequency of the shaft. Where the vibration arises as a result of, for example, destabilizing self-excited forces, the orbit will not normally be of a simple shape, but will change form over a period of time and it will not necessarily be harmonically related. In general, the vibration of the shaft may arise from a number of different sources and, therefore, a complex orbit will be produced, which is the vectorial sum of the effects of the individual excitation forces.

B.2 Measurement of shaft vibration

At any axial location, the orbit of the shaft can be obtained by taking measurements with two vibration transducers mounted in different radial planes, separated by 90° (this is the preferred separation, but small deviations from this do not cause significant errors). If the angle between the transducer locations is substantially different from 90°, a vector resolution into the orthogonal directions will be required. If the transducers measure absolute vibration, then the orbit will be the absolute orbit of the shaft independent of the vibratory motion of the non-rotating parts. If the transducers measure relative vibration, then the measured orbit will be relative to that part of the structure upon which the transducers are mounted.

B.3 Measurement quantities

B.3.1 Time-integrated mean position

The mean values of the shaft displacement $(\bar{x},\bar{y})$, in any two specified orthogonal directions, relative to a reference position, as shown in figure B.1, are defined by integrals with respect to time, as shown in the following equations:

$$\bar{x} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} x(t)\,\mathrm{d}t \qquad \text{... (B.1)}$$

$$\bar{y} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} y(t)\,\mathrm{d}t \qquad \text{... (B.2)}$$

where $x(t)$ and $y(t)$ are the time-dependent alternating values of displacement relative to the reference position, and $(t_2 - t_1)$ is large relative to the period of the lowest frequency vibration component. In the case of absolute vibration measurements, the reference position is fixed in space. For relative vibration measurements, these values give an indication of the mean position of the shaft relative to the non-rotating parts at the axial location where the measurements are made. Changes in the values may be due to a number of factors, such as bearing/foundation movements, changes in oil film characteristics, etc., which normally occur slowly relative to the period of the vibration components which make up the alternating values.

It should be noted that, in general, the time-integrated mean position in any direction differs from the position defined by taking half the summation of the maximum and minimum displacement values (see figure B.2). However, when the shaft vibration is a single frequency and sinusoidal, then the locus of the shaft centre will be an ellipse. In such circumstances, the time-integrated mean position in any direction of measurement will be the same as the position identified by taking half the summation of the maximum and minimum displacement values.

B.3.2 Peak-to-peak displacement of the vibration

The primary quantities of interest in shaft measurements are the alternating values which describe the shape of the orbit. Consider the kinetic shaft orbit shown in figure B.2 and assume that there are two transducers A and B mounted 90° apart, which are used to measure the shaft vibration. At some instant, the shaft centre will be coincident with the point K on the orbit and the corresponding instantaneous value of shaft displacement from the mean position will be S_1. However, in the plane of the transducers A and B, the instantaneous values of shaft displacement from the mean position will be S_{A1} and S_{B1}, respectively, where

$$S_1^2 = S_{A1}^2 + S_{B1}^2 \qquad \text{... (B.3)}$$

The values of S_1, S_{A1} and S_{B1} will vary with time as the shaft centre moves around the orbit; the corresponding waveforms measured by each transducer are shown in figure B.2.

NOTE 6 If the orbit is elliptical, then these waveforms would be pure sine waves of the same frequency.

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

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

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

Whereas measurement of the peak-to-peak displacement in any two given orthogonal directions is a simple matter, the value and angular position of the maximum peak-to-peak displacement shown in figure B.2 is difficult to measure directly. However, in practice, it has been found acceptable to use alternative measurement quantities which enable a suitable approximation for the maximum peak-to-peak displacement value to be obtained. For more precise determinations, it is necessary to examine the shaft orbit in more detail, as for example with an oscilloscope. The three most common methods for obtaining satisfactory approximations are described in B.3.2.1 to B.3.2.3.

B.3.2.1 Method A: Resultant value of the peak-to-peak displacement values measured in two orthogonal directions

The value of $S_{(p-p)max}$ can be approximated from the following equation:

$$S_{(p-p)max} = \sqrt{S_{A(p-p)}^2 + S_{B(p-p)}^2} \qquad \text{... (B.4)}$$

The use of equation (B.4) as an approximation when the vibration is predominantly at rotational frequency will generally over-estimate the value of $S_{(p-p)max}$ with a maximum error of approximately 40 %.

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

B.3.2.2 Method B: Taking the maximum value of the peak-to-peak displacement values measured in two orthogonal directions

The value of $S_{(p-p)max}$ can be approximated from the following equation:

$$S_{(p-p)max} = S_{A(p-p)} \text{ or } S_{B(p-p)} \qquad \text{... (B.5)}$$

whichever is the greater.

The use of equation (B.5) as an approximation when the vibration is predominantly at rotational frequency will generally under-estimate the value of $S_{(p-p)max}$, with a maximum error of approximately 30 %.

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

B.3.2.3 Method C: Measurement of S_{max}

The instantaneous value of the shaft displacement can be defined by S_1, as shown in figure B.2, which is derived from the transducer measurements S_{A1} and S_{B1} using equation (B.3). There is a point on the orbit, defined by point P in figure B.2, where the displacement from the mean position is a maximum. The value of S_1 corresponding to this position is denoted by S_{max}, which is defined as the maximum value of displacement

$$S_{max} = [S_1(t)]_{max} = \left[\sqrt{[S_A(t)]^2 + [S_B(t)]^2}\right]_{max} \quad \ldots (B.6)$$

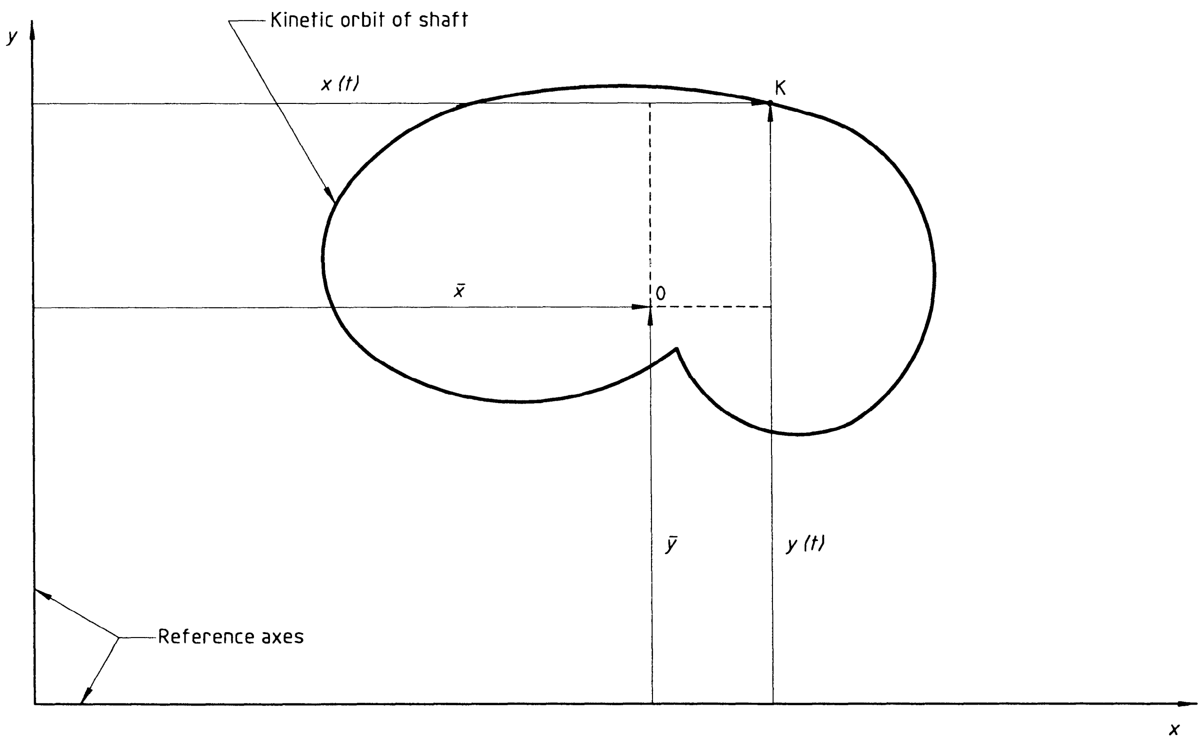
The point on the orbit where S_{max} occurs does not necessarily coincide with the point where S_A and S_B are at their maximum values. Clearly, for a particular orbit, there is one value of S_{max} and this is independent of the position of the measuring transducers provided that the mean position 0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

The value of $S_{(p\text{-}p)max}$ can be approximated from the following equation:

$$S_{(p\text{-}p)max} = 2\ S_{max} \quad \ldots (B.7)$$

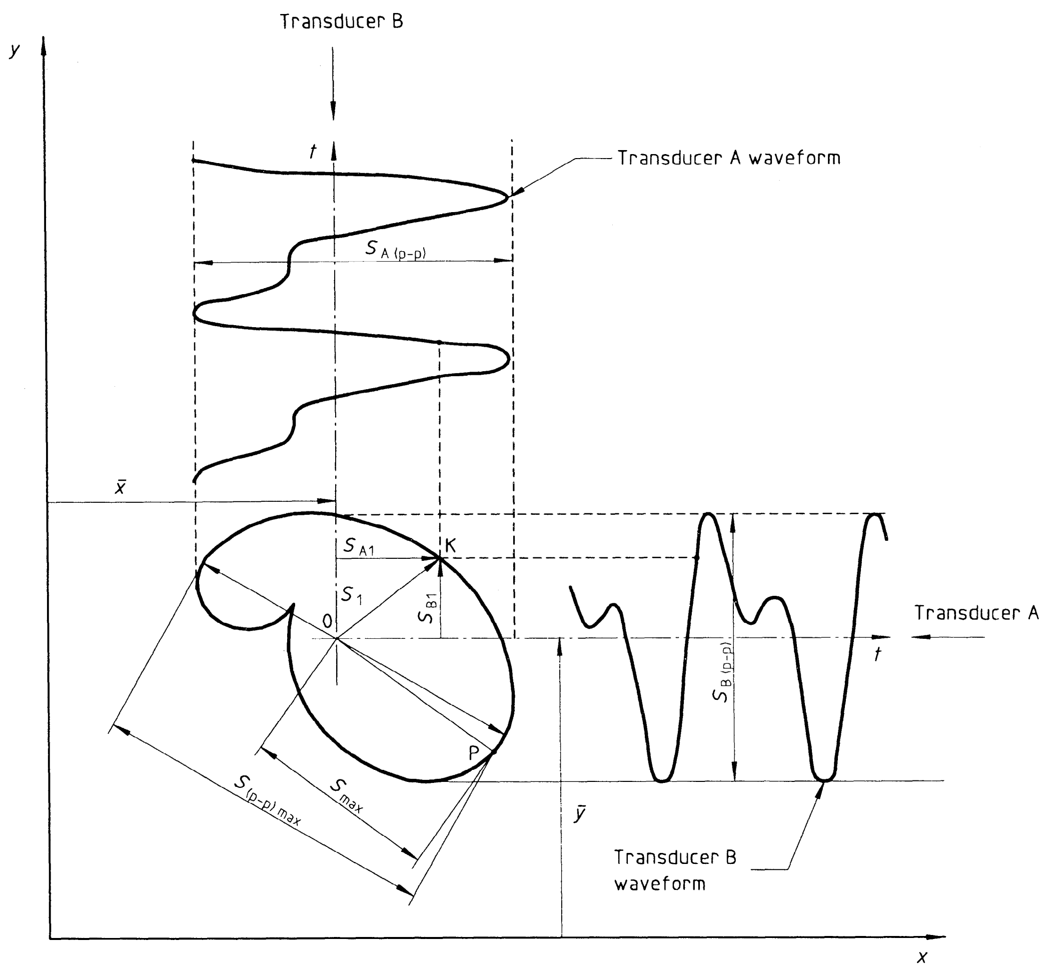
Equation (B.7) will be correct when the two orthogonal measurements from which S_{max} is derived are of single-frequency sinusoidal form. In most other cases, this equation will over-estimate $S_{(p\text{-}p)max}$, since this depends on the nature of the harmonic vibration components present.

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



0 Mean position of orbit
K Instantaneous position of shaft centre
$\left.\begin{array}{l}\bar{x}\\ \bar{y}\end{array}\right\}$ Mean values of shaft displacement
$\left.\begin{array}{l}x(t)\\ y(t)\end{array}\right\}$ Time-dependent alternating values of shaft displacement

Figure B.1 — Kinetic orbit of shaft



x, y	Fixed reference axes
0	Time-integrated mean position of orbit
$\bar{x}$, $\bar{y}$	Time-integrated mean values of shaft displacement
K	Instantaneous position of shaft centre
P	Position of shaft for maximum displacement from time-integrated mean position
S_1	Instantaneous value of shaft displacement
S_{max}	Maximum value of shaft displacement from time-integrated mean position 0
S_{A1}, S_{B1}	Instantaneous values of shaft displacement in directions of transducers A and B, respectively
$S_{(p-p)max}$	Maximum value of peak-to-peak displacement
$S_{A(p-p)}$, $S_{B(p-p)}$	Peak-to-peak values of shaft displacement in directions of transducers A and B, respectively

Figure B.2 — Kinetic orbit of shaft — Definition of displacement

Annex C
(informative)

Recommendations for instrumentation to be used for measuring relative and absolute shaft vibration

Introduction

Three types of measurement system are in common use for the measurement of transverse shaft vibration, each using either one or two measuring directions. One type uses non-contacting transducers which measure the relative motion between the shaft and a bearing; another uses shaft-riding seismic transducers which measure the absolute motion of the shaft; and the third provides measurement of the absolute motion of the shaft by combining the outputs of non-contacting transducers and structurally mounted (e.g. on the bearing housing) seismic transducers.

NOTE 7 In the examples given in clauses C.1 to C.3, two transducers are described mounted 90° apart in the same transverse plane perpendicular to the shaft axis. However, in certain cases, measurement in one direction will suffice. (See 3.3.)

C.1 Relative-motion measurement system (non-contacting transducers)

Figure C.1 presents a schematic diagram of a typical instrument system used to measure the relative motion between the shaft and a structural member (for example, the bearing housing). This system consists of a non-contacting transducer, a local signal conditioner and a readout instrument.

During initial installation of the transducer, it is desirable to conduct an *in situ* calibration of the transducer output versus the gap. It should be noted that machine operating conditions can alter the average gap position. Therefore, care should be taken to ensure operation within the linear range of the transducer.

When mounting non-contacting transducers, care should also be taken to ensure that they sense only the shaft vibration and that the accuracy is not affected by any conductive material or magnetic fields in proximity to the transducer.

It is recommended that the measurement system should be capable of indicating both the time-dependent alternating displacement value over the frequency range of interest and the mean position of the shaft relative to the support structure. The latter provides a means of setting the transducer at the correct gap position and assessing the amount of runout obtained at low speed when stable bearing oil films have been established but centrifugal effects are negligible. (For example, on a machine of rated speed 3 000 r/min, the runout could be assessed at a speed of the order of 200 r/min.)

It is recognized that other measuring systems, such as shaft-riding relative motion probes, are used.

NOTE 8 Caution should be exercised when interpreting slow-roll runout measurements, as they can be affected by, for example, temporary bows, erratic movements of the journal within the bearing clearance, axial movements, etc.

C.2 Absolute-motion measurement system (shaft-riding with seismic transducers)

Figure C.2 presents a schematic diagram of a typical instrument system used to measure the absolute motion of the shaft. This system consists of a seismic transducer (velocity type or accelerometer) mounted on a shaft-rider, a shaft-rider support system which permits the shaft-rider to follow accurately the motion of the shaft, and a readout instrument.

NOTE 9 No measurement of the shaft mean position relative to the structure is possible with this system.

The shaft-riding mechanism should accurately transmit the shaft vibration to the seismic transducer, and should be free from chatter or natural frequencies which could adversely affect, or distort, the measurements of shaft vibration within the frequency range of interest.

The output of the seismic transducers should be suitably conditioned to provide a signal which gives an accurate measure of the time-dependent alternating displacement value for the shaft.

C.3 Absolute-motion measurement system (combined non-contacting and seismic transducers)

Figure C.3 presents a schematic diagram of a typical instrument system used to measure absolute shaft motion, and which can also be used to measure the absolute motion of the bearing housing and the relative motion of the shaft. This system consists of a non-contacting type of relative-displacement transducer, a seismic transducer (i.e. velocity type or accelerometer), a signal conditioner and a readout instrument.

The two transducers should be mounted on a common rigid structure close together with their sensitive axes parallel to ensure that both undergo the same absolute structural motion.

The non-contacting transducer portion of the system should be similar to that described in clause C.1, and should provide an output proportional to the relative shaft vibration displacement, as well as gap distance. This output is the resultant of two motions: the motion of the shaft, and the motion of the structure on which the non-contacting transducer is mounted.

The output of the seismic transducer, which is proportional to the motion of the structure to which it and the non-contacting transducer are mounted, should be processed as is necessary to provide a displacement signal. The amplitude and phase of this signal, together with that of the non-contacting relative-motion transducer, should be adjusted and vectorially combined to provide an accurate measure of the absolute motion of the shaft. It should be noted that individual measuring systems can modify the phase and amplitude of the basic signal. Therefore, corrections should be made for these inherent characteristics before vectorial addition. The seismic transducer also provides an output proportional to the absolute vibration of the non-rotating member (for example, that of the bearing housing).

The measurement system should be capable of indicating the mean position of the shaft relative to the support structure and the time-dependent alternating displacement value of absolute shaft vibration, which is the vectorial sum of the absolute motion of the structure and the relative motion of the shaft.

C.4 Overall instrument system performance and environmental considerations

System performance and environmental considerations will form the subjects of future International Standards.

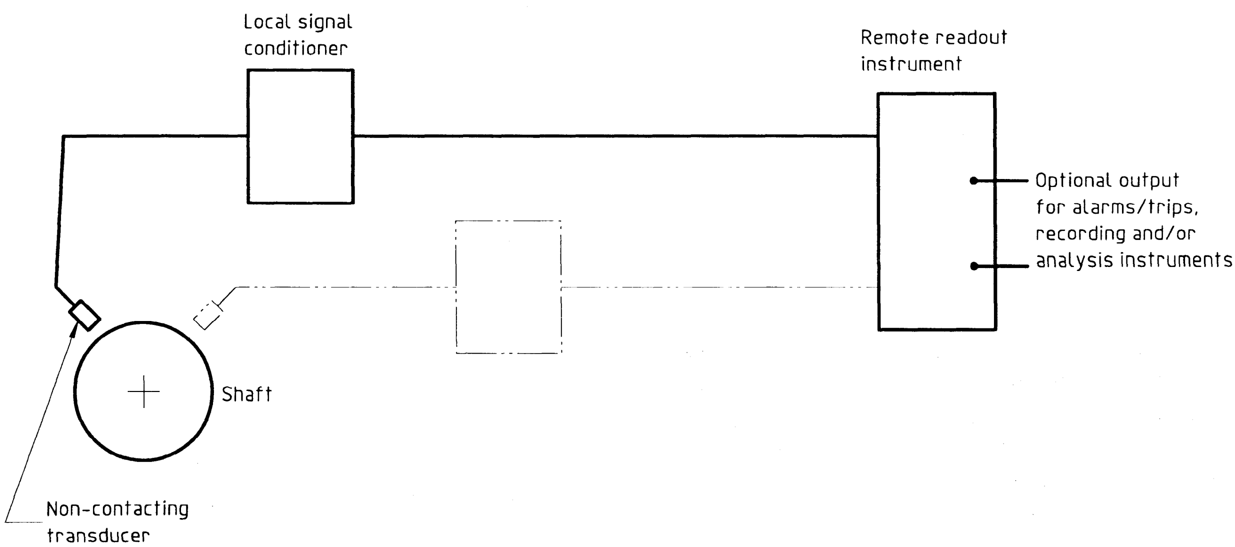


Figure C.1 — Schematic diagram of a relative-motion measurement system using non-contacting transducers (see clause C.1)

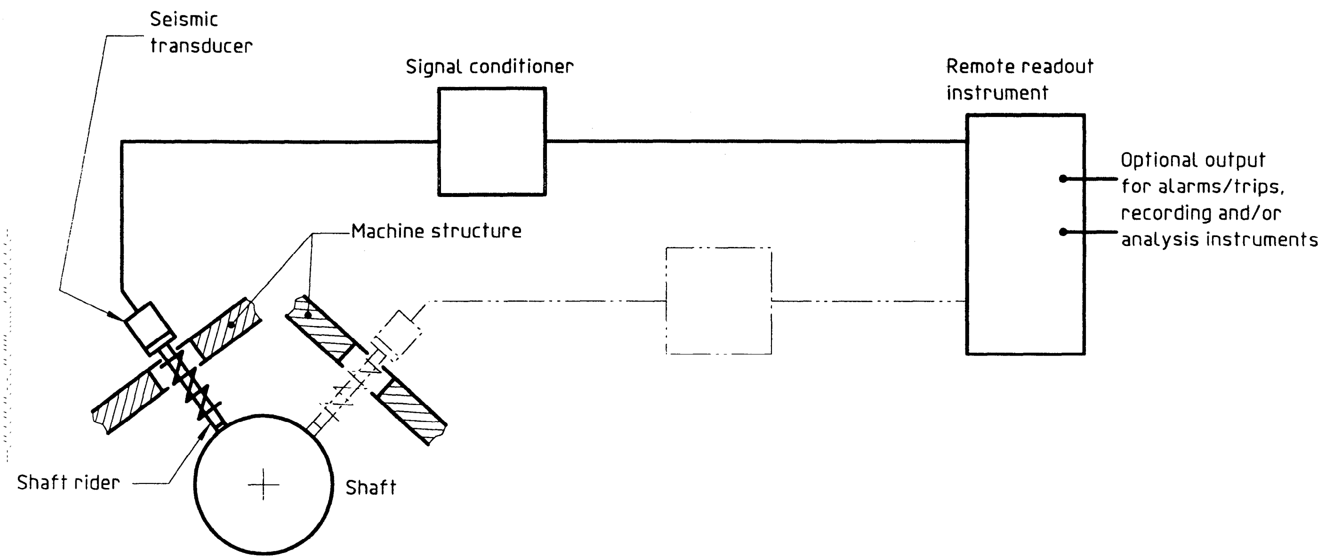


Figure C.2 — Schematic diagram of an absolute-motion measurement system using a shaft-rider mechanism with seismic transducers (see clause C.2)

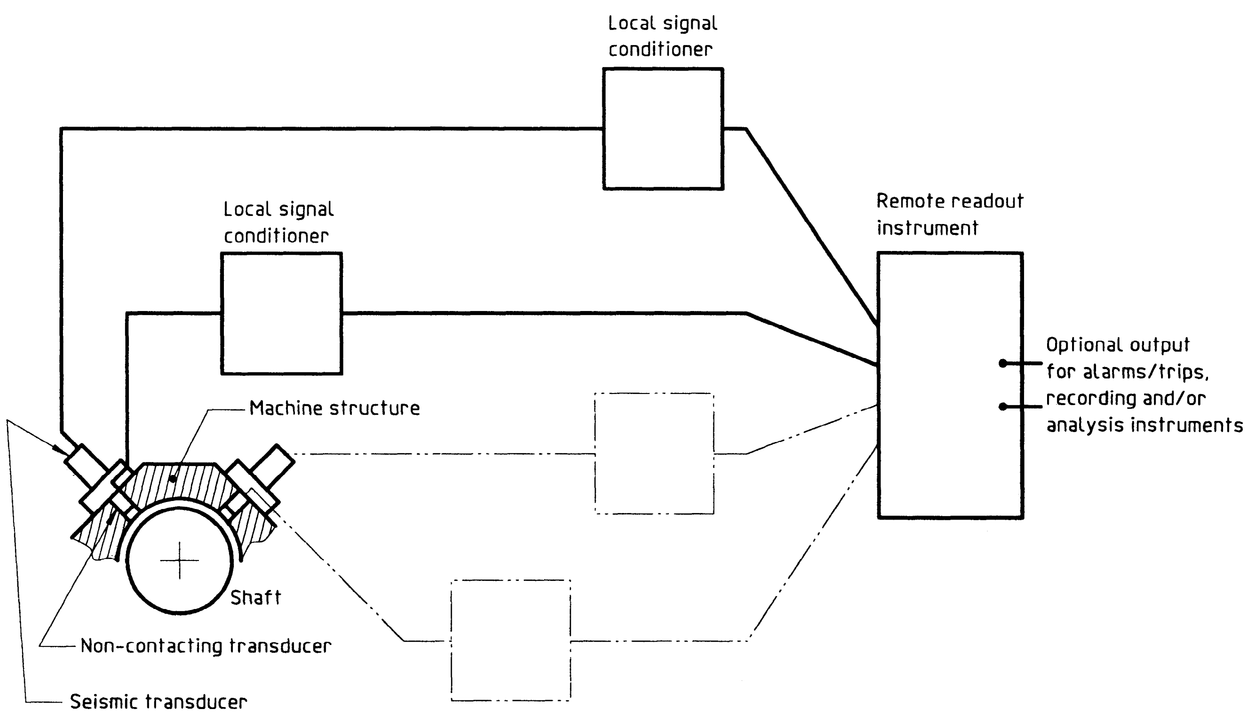


Figure C.3 — Schematic diagram of an absolute-motion measurement system using a combination of non-contacting and seismic transducers (see clause C.3)

Annex D
(informative)

Vector analysis of change in vibration

Introduction

Evaluation criteria are presented in annex A in terms of the normal steady running value of shaft vibration and any changes that may occur in the magnitude of these steady values. However, some changes may only be identified by vector analysis of the individual frequency components.

D.1 General

The overall steady vibration signal measured on a rotating shaft is complex in nature and is made up of a number of different frequency components. Each of these components is defined by its frequency, amplitude and phase relative to some known datum. Conventional vibration-monitoring equipment measures the magnitude of the overall complex signal and does not differentiate between the individual frequency components. However, modern diagnostic equipment is capable of analysing the complex signal so that the amplitude and phase of each frequency component can be identified. This information is of great value to the vibration engineer, since it facilitates the diagnosis of likely reasons for abnormal vibration behaviour.

Changes in individual frequency components, which may be significant, are more readily seen as vector changes than as changes in broad-band vibration.

D.2 Importance of vector change

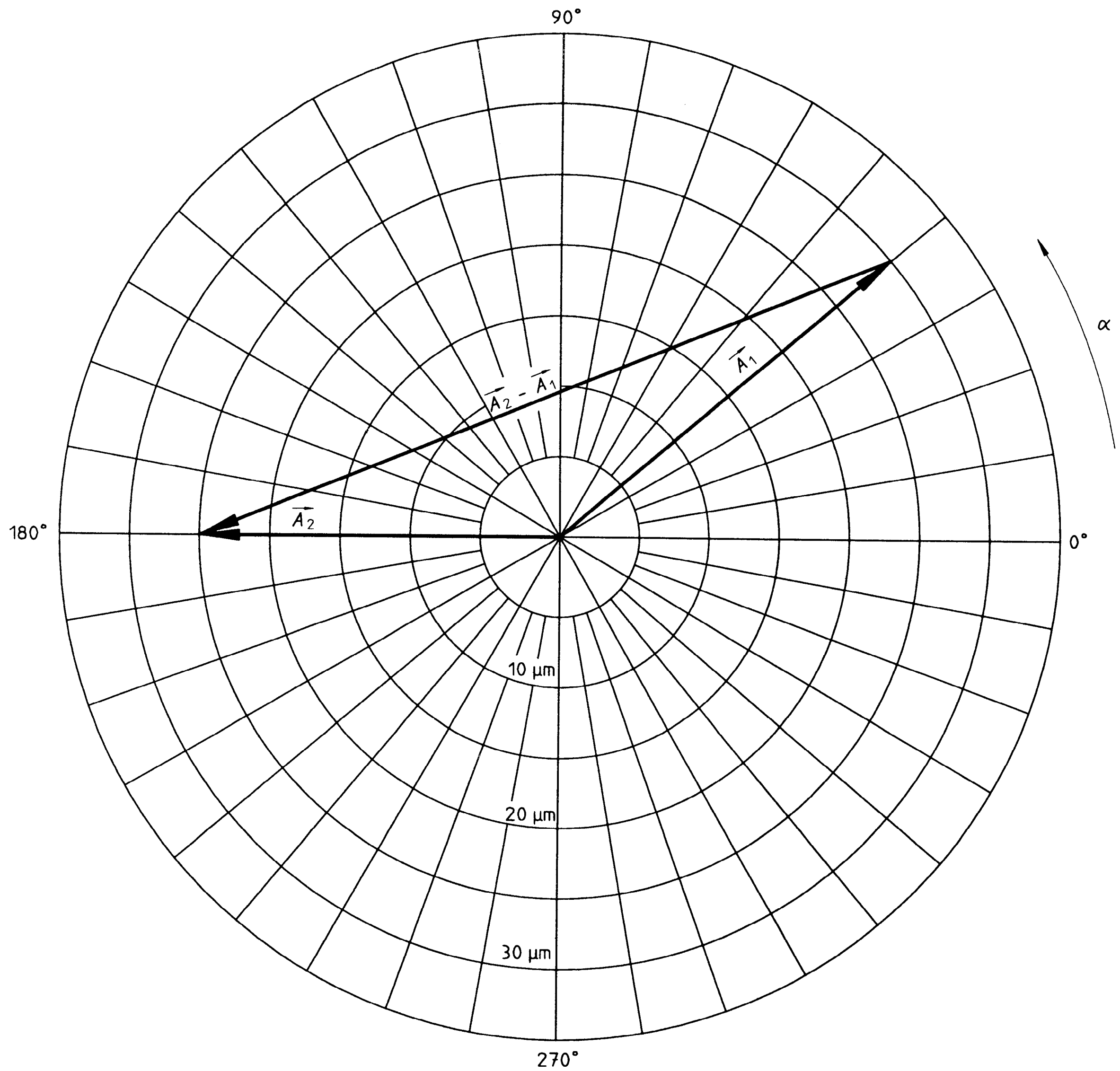
Figure D.1 is a polar diagram which is used to display simultaneously in vector form the amplitude and phase of one of the frequency components of a complex vibration signal.

Vector $\vec{A_1}$ describes the initial steady-state vibration condition; i.e. in this condition the vibration magnitude is 30 µm with a phase angle of 40°. Vector $\vec{A_2}$ describes the steady-state vibration condition after some change has occurred to the machine; i.e. the vibration magnitude is now 25 µm with a phase angle of 180°. Hence, although the vibration magnitude has decreased by 5 µm, the true change of vibration is represented by the vector $\vec{A_2} - \vec{A_1}$, which has a magnitude of 52 µm, over 10 times that indicated by comparing the vibration magnitude alone.

This example illustrates the limitations of basing a criterion for change of vibration on vibration magnitude alone.

D.3 Monitoring vector changes

The example given above clearly illustrates the importance of identifying the vector change in a vibration signal. However, it is necessary to appreciate that, in general, the overall vibration signal is composed of a number of individual frequency components, each of which may register a vector change. Furthermore, an unacceptable change in one particular frequency component may be within acceptable limits for a different component. Consequently, it is not possible at this time to define criteria for vector changes in individual frequency components that are compatible with the context of this part of ISO 7919, which is aimed primarily at normal operational monitoring of overall vibration by non-vibration specialists.



Initial steady-state vector	$\left\|\vec{A}_1\right\| = 30$ µm, $\alpha = 40°$
Steady-state vector after change	$\left\|\vec{A}_2\right\| = 25$ µm, $\alpha = 180°$
Change in vibration magnitude	$\left\|\vec{A}_2\right\| - \left\|\vec{A}_1\right\| = -5$ µm
Vector of change	$\left\|\vec{A}_2 - \vec{A}_1\right\| = 52$ µm

Figure D.1 — Comparison of vector change and change in magnitude for a discrete frequency component

Annex E
(informative)

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1) To be published.

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