

# Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts —

## Part 1: General guidelines

ICS 17.160

# National foreword

This British Standard is the UK implementation of ISO 10816-1:1995+A1:2009. It supersedes BS 7854-1:1996 which is withdrawn.

The start and finish of text introduced or altered by amendment is indicated in the text by tags. Tags indicating changes to ISO text carry the number of the ISO amendment. For example, text altered by ISO amendment 1 is indicated by **A1** **A1**.

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## Introduction

This part of ISO 10816 is a basic document which establishes general guidelines for the measurement and evaluation of mechanical vibration of machinery, as measured on the non-rotating (and, where applicable, non-reciprocating) parts of complete machines, such as bearing housings. Recommendations for measurements and evaluation criteria pertaining to specific machine types are provided in additional parts of ISO 10816.

For many machines, measurements made on non-rotating parts are sufficient to characterize adequately their running conditions with respect to trouble-free operation. However, there are some machines, such as those containing flexible rotors, for which measurements on non-rotating parts may not be totally adequate. In such cases, it may be necessary to monitor the machine using measurements on both the rotating and non-rotating parts, or on the rotating parts alone. For such machines, the guidelines presented in this part of ISO 10816 are complemented by those given for shaft vibration in ISO 7919-1. If the procedures of both standards are applicable, the one which is more restrictive generally applies.

Vibration measurements can be used for a number of purposes including routine operational monitoring, acceptance tests and diagnostic and analytical investigations. This part of ISO 10816 is designed to provide guidelines for operational monitoring and acceptance tests only.

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

## 1 Scope

This part of ISO 10816 establishes general conditions and procedures for the measurement and evaluation of vibration using measurements made on non-rotating and, where applicable, non-reciprocating parts of complete machines. The general evaluation criteria, which are presented in terms of both vibration magnitude and change of vibration, relate to both operational monitoring and acceptance testing. They have been provided primarily with regard to securing reliable, safe, long-term operation of the machine, while minimizing adverse effects on associated equipment. Guidelines are also presented for setting operational limits.

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

This part of ISO 10816 does not include any consideration of torsional vibration.

## 2 Normative reference

**A1)** 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, *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria — Part 1: General guidelines* **A1**

## 3 Measurements

This clause describes the measurements, procedures and operating conditions recommended for assessing machine vibration. The guidelines given will permit the evaluation of vibration in accordance with the general criteria and principles given in clause 5.

### 3.1 Measurement parameters

#### 3.1.1 Frequency range

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

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

Guidelines for instrumentation frequency ranges for specific machine classes will be given in the appropriate parts of ISO 10816.

NOTE 1 In the past, vibration severity was often related to broad-band vibration velocity [mm/s (r.m.s.)] in the range 10 Hz to 1 000 Hz. However, different frequency ranges and measurement quantities may apply for different machine types.

#### 3.1.2 Measurement quantity

For the purposes of this part of ISO 10816, the following can be used:

- a) vibration displacement, measured in micrometres;

<sup>1)</sup> To be published. (Revision of ISO 7919-1:1986)

- b) vibration velocity, measured in millimetres per second;
- c) vibration acceleration, measured in metres per square second.

The use, application and limitations of these quantities are discussed further in clause 5.

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

In order to avoid confusion and to ensure correct interpretation, it is important at all times to identify clearly the measurement units [e.g.  $\mu\text{m}$  (p-p), mm/s (r.m.s.)].

### 3.1.3 *Vibration magnitude*

The result of measurements made with an instrument which complies with the requirements of clause 4 is called the vibration magnitude at a specific measuring position and direction.

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

### 3.1.4 *Vibration severity*

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

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

## 3.2 *Measuring positions*

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

To define the vibrational behaviour at each measuring position, it is necessary to take measurements in three mutually perpendicular directions. The full complement of measurements represented in Figure 1 to Figure 5 is generally only required for acceptance testing. The requirement for operational monitoring is usually met by performing one or both measurements in the radial direction (i.e. normally in the horizontal-transverse and/or vertical directions). These can be supplemented by a measurement of axial vibration. The latter is normally of prime significance at thrust bearing locations where direct axial dynamic forces are transmitted.

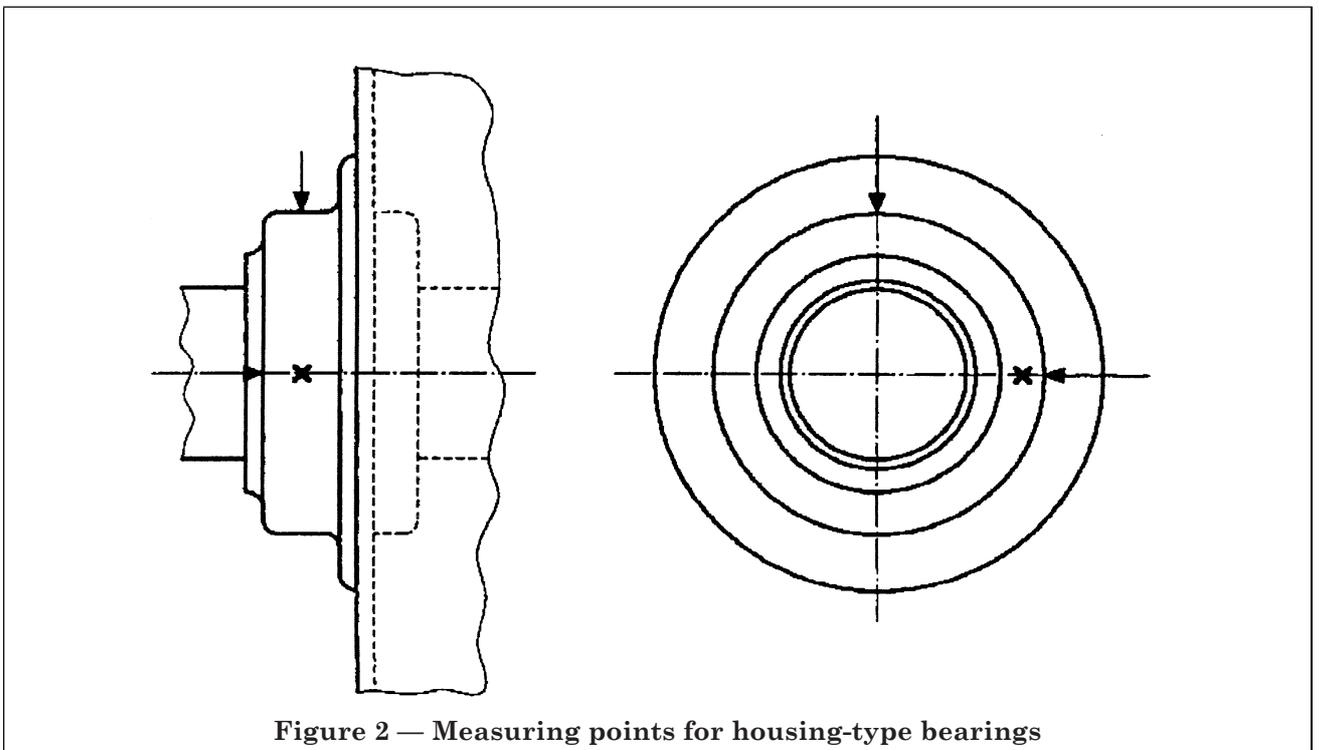
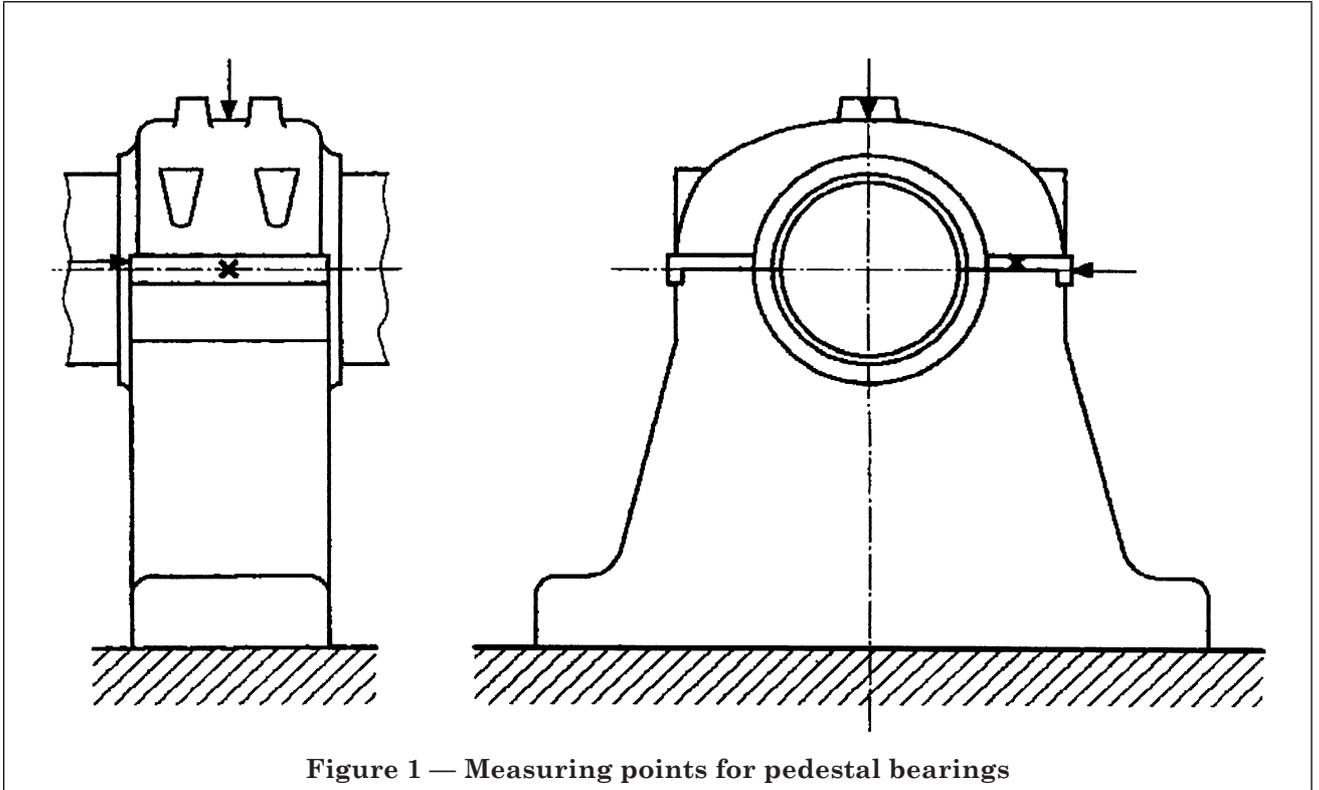
Detailed recommendations for specific machine types are provided in the additional parts of ISO 10816.

## 3.3 *Machine support structure for acceptance testing*

### 3.3.1 *In situ tests*

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

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



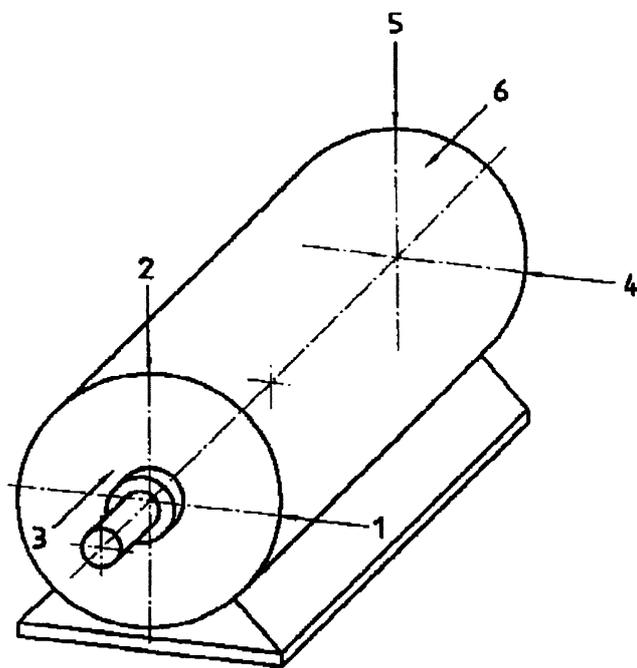


Figure 3 — Measuring points for small electrical machines

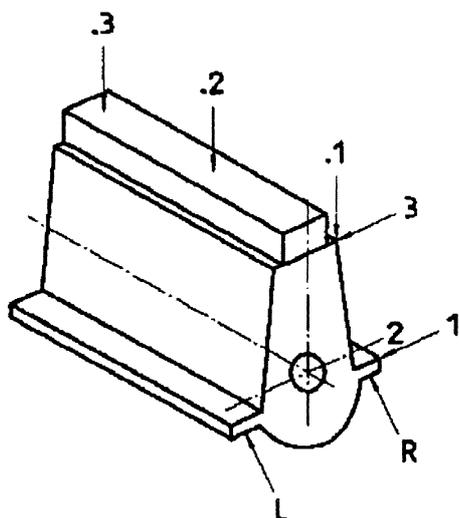


Figure 4 — Measuring points for reciprocating engines

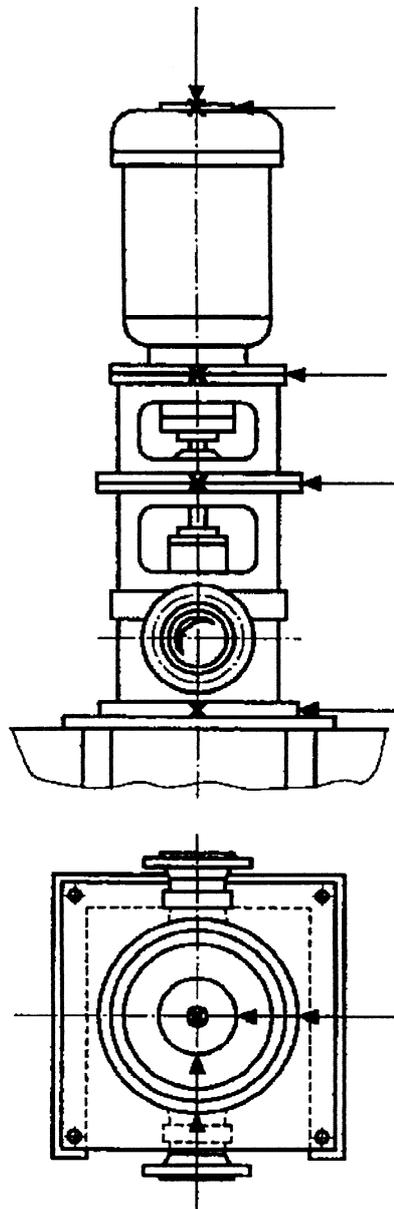


Figure 5 — Measuring points for vertical machine sets

### 3.3.2 In a test facility

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

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

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

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

### 3.4 Machine support structure for operational monitoring

Operational monitoring is carried out on fully installed machines *in situ* (i.e. on their final support structure).

### 3.5 Machine operating conditions

Vibration measurements shall be made after achieving agreed normal operating conditions. Additional vibration measurements that may be taken under other conditions are not applicable for evaluation in accordance with clause 5.

### 3.6 Environmental vibration evaluation

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

## 4 Instrumentation

The instrumentation used shall be designed to operate satisfactorily in the environment for which it is to be used, for example with respect to temperature, humidity, etc. Particular attention shall be given to ensuring that the vibration transducer is correctly mounted and that its presence does not affect the vibration response characteristics of the machine.

Two instrument systems presently in common use to monitor broad-band vibration are acceptable, namely:

- a) instruments which incorporate r.m.s. detector circuits and display the r.m.s. values;
- b) instruments which incorporate either r.m.s. or averaging detector circuits, but are scaled to read peak-to-peak or peak values. The scaling is based on an assumed sinusoidal relationship between r.m.s., average, peak-to-peak and peak values.

If the vibration evaluation is based on more than one measurement quantity (i.e. displacement, velocity, acceleration), the instrumentation used shall be able to characterize all the relevant quantities.

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 General

This clause specifies general criteria and principles for the evaluation of machine vibration. The evaluation criteria relate to both operational monitoring and acceptance testing, and they apply only to the vibration produced by the machine itself and not to vibration transmitted from outside. For certain classes of machinery, the guidelines presented in this part of ISO 10816 are complemented by those given for shaft vibration in ISO 7919-1. If the procedures of both standards are applicable, 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 10816 as they are developed.

## 5.2 Criteria

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

### 5.3 Criterion I: Vibration magnitude

This criterion is concerned with defining limits for absolute vibration magnitude consistent with acceptable dynamic loads on the bearings and acceptable vibration transmission into the support structure and foundation. The maximum vibration magnitude observed at each bearing or pedestal is assessed against four evaluation zones established from international experience. This maximum magnitude of vibration measured is defined as the vibration severity (see 3.1.4).

#### 5.3.1 Evaluation zones

**A1)** The following typical evaluation zones are defined to permit a qualitative assessment of the vibration on a given machine and to provide guidelines on possible actions. Different categorization and number of zones may apply for specific machine types. These are provided in additional parts of ISO 10816, which also define specific values for the zone boundary limits. For those machine types for which no specific part has been developed, see Annex B. **A1)**

**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.

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 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.

#### 5.3.2 Evaluation zone limits

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

In most cases, it has been found that vibration velocity is sufficient to characterize the severity of vibration over a wide range of machine operating speeds. However, it is recognised that the use of a single value of velocity, regardless of frequency, can lead to unacceptably large vibration displacements. This is particularly so for machines with low operating speeds when the once-per-revolution vibration component is dominant. Similarly, constant velocity criteria for machines with high operating speeds, or with vibration at high frequencies generated by machine component parts can lead to unacceptable accelerations.

Consequently, acceptance criteria based on velocity will take the general form of Figure 6. This indicates the upper and lower frequency limits  $f_u$  and  $f_l$  and shows that below a defined frequency  $f_x$  and above a defined frequency  $f_y$  the allowable vibration velocity is a function of the vibration frequency (see also annex C). However, for vibration frequencies between  $f_x$  and  $f_y$ , a constant velocity criterion applies. The r.m.s. velocities listed in annex B refer to this constant velocity region. The precise nature of the acceptance criteria and the values of  $f_l$ ,  $f_u$ ,  $f_x$  and  $f_y$  for specific machine types will be given in the additional parts of ISO 10816.

For many machines, the broad-band vibration consists primarily of a single frequency component, often shaft rotational frequency. In this case, the allowable vibration is obtained from Figure 6 as the vibration velocity corresponding to that frequency.

For less-common machines, where there may be significant vibratory energy beyond the breakpoints  $f_x$  and  $f_y$  of Figure 6, a number of different approaches are possible. Examples are the following.

a) In addition to the usual, broad-band velocity, broad-band displacement may be measured when there is significant energy below  $f_x$ . Similarly, broad-band acceleration may be measured when there is significant energy above  $f_y$ . The allowable vibration displacement and acceleration should be consistent with the velocity corresponding to the sloped portions of Figure 6.

b) The velocity, displacement or acceleration at each significant component throughout the frequency spectrum may be determined using a frequency analyser. The equivalent broad-band velocity can be obtained using equation (A.2) after applying appropriate weighting factors, consistent with Figure 6, for those components whose frequencies are below  $f_x$  or above  $f_y$ . This value should then be evaluated relative to the constant velocity between  $f_x$  and  $f_y$ . It should be noted that, except for the case when the broad-band vibration consists primarily of a single frequency component, a direct comparison of the frequency spectrum components with the curves of Figure 6 would yield misleading results.

c) A composite broad-band measurement encompassing the entire spectrum may be carried out using an instrument incorporating weighting networks consistent with the shape of Figure 6. This value should then be evaluated relative to the constant velocity between  $f_x$  and  $f_y$ .

The evaluation criteria for specific machine types will be given in the additional parts of ISO 10816 as they become available. Annex C provides additional guidance. For certain machine types, it may be necessary to define further criteria beyond those described by Figure 6 (see for example, 5.6.3).

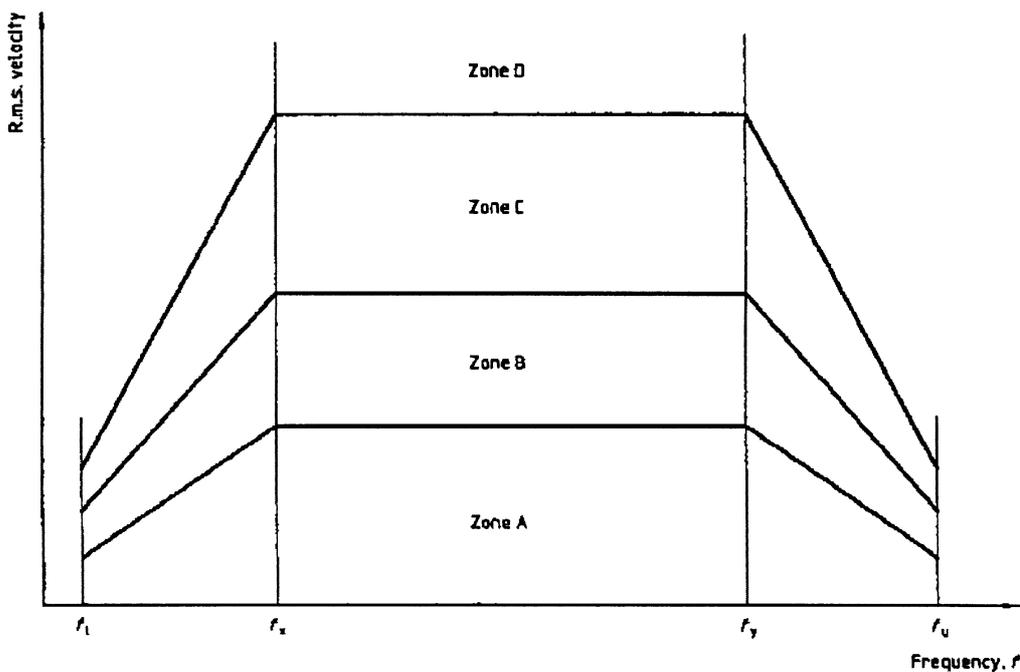


Figure 6 — General form of vibration velocity acceptance criteria

#### 5.4 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 of broad-band vibration for monitoring purposes are given in the additional parts of ISO 10816. However, it should be noted that some changes may not be detected unless the discrete frequency components are monitored (see 5.6.1).

#### 5.5 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 the additional parts of ISO 10816.

#### 5.5.1 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. Guidelines for specific machine types are given in the additional parts of ISO 10816.

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, a 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.

#### 5.5.2 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.

#### 5.6 Additional factors

##### 5.6.1 Vibration frequencies and vectors

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. In some cases, these changes would go undetected when using broad-band vibration measurements. This is demonstrated in annex D.

The specification of criteria for vector changes is beyond the present scope of this part of ISO 10816.

#### **5.6.2 *Vibration sensitivity of the machine***

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.

#### **5.6.3 *Special techniques for rolling element bearings***

Alternative approaches other than broad-band vibration measurements are continuing to be evolved for assessing the conditions of rolling element bearings. These are discussed further in annex E. The definition of evaluation criteria for such additional methods is beyond the scope of this part of ISO 10816.

## Annex A (informative) Vibratory waveform relationships

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

From measured vibration velocity versus time records, the r.m.s. value of the velocity may be calculated as follows:

$$v_{\text{r.m.s.}} = \sqrt{\frac{1}{T} \int_0^T v^2(t) dt} \quad \dots \text{(A.1)}$$

where

- $v(t)$  is the time-dependent vibration velocity;
- $v_{\text{r.m.s.}}$  is the corresponding r.m.s. velocity;
- $T$  is the sampling time, which is longer than the period of any of the major frequency components of which  $v(t)$  is composed.

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

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

$$\begin{aligned} \text{A1} \quad v_{\text{r.m.s.}} &= \pi \times 10^{-3} \sqrt{\frac{1}{2} [(s_1 f_1)^2 + (s_2 f_2)^2 + \dots + (s_n f_n)^2]} \quad \text{A1} \\ &= \sqrt{v_1^2 + v_2^2 + \dots + v_n^2} \\ &= \frac{10^3}{2\pi} \sqrt{\left(\frac{a_1}{f_1}\right)^2 + \left(\frac{a_2}{f_2}\right)^2 + \dots + \left(\frac{a_n}{f_n}\right)^2} \\ &\dots \text{(A.2)} \end{aligned}$$

NOTE 2 According to ISO 2041, the frequency  $f$  may also be called cyclic frequency  $f$ .

In the case where the vibration consists of only two significant frequency components giving beats of r.m.s. value,  $v_{\text{min}}$  and  $v_{\text{max}}$ ,  $v_{\text{r.m.s.}}$  may be determined approximately from the relationship:

$$v_{\text{r.m.s.}} = \sqrt{\frac{1}{2} (v_{\text{max}}^2 + v_{\text{min}}^2)} \quad \dots \text{(A.3)}$$

The operation of interchanging vibration acceleration, velocity or displacement values can be accomplished only for single-frequency harmonic components using, for example, Figure A.1. If the vibration velocity of a single-frequency component is known, the peak-to-peak displacement may be evaluated from the relationship:

$$s_i = \frac{450 v_i}{f_i} \quad \dots \text{(A.4)}$$

where

- $s_i$  is the peak-to-peak displacement value, in micrometres;
- $v_i$  is the r.m.s. value of the vibration velocity, in millimetres per second, of the component with frequency  $f_i$ , in hertz.

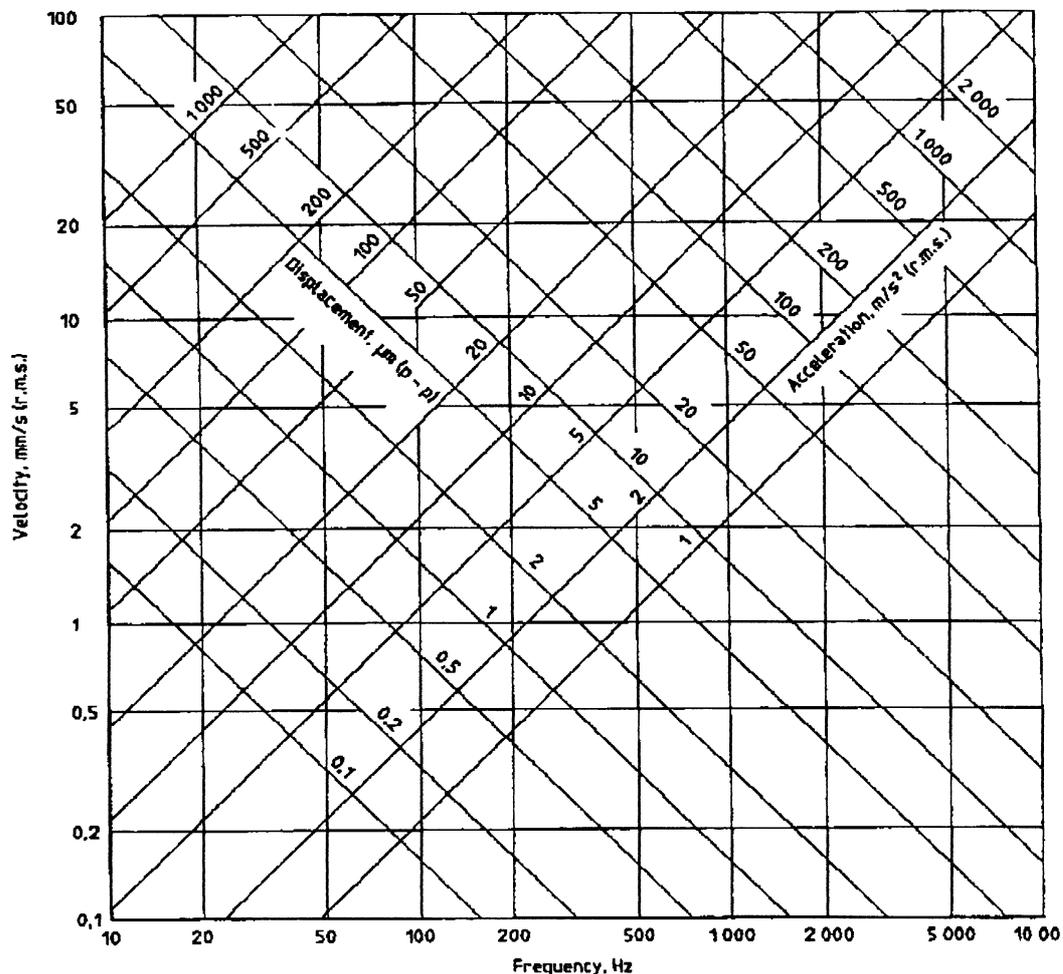


Figure A.1 — Relationship between acceleration, velocity and displacement for single-frequency harmonic components

**Annex B (informative)**  
**Informative guidelines for setting zone boundary limits**

This part of ISO 10816 is a basis document which establishes general procedures for the measurement and evaluation of mechanical vibration of machines, as measured on non-rotating parts. It does not define specific evaluation criteria. These are provided for many common machine types in other parts of ISO 10816.

The evaluation criteria for machine types for which no specific parts have been developed are normally based on successful operating experience with machines of similar design and should be subject to agreement between the supplier and the purchaser of the machine. Factors which should be taken into account include the position and direction of measurement, frequency range, support flexibility and operating conditions.

In those cases where there is no suitable experience or part of ISO 10816 available, a range of typical values for the zone A/B, B/C and C/D boundaries, respectively (see 5.3.1), are given in Table B.1.

In general, the zone boundary limits

- a) for small machines (e.g. electric motors with power up to 15 kW) tend to lie at the lower end of the range, and
- b) for larger machines (e.g. prime movers with flexible supports in the direction of measurement) tend to lie at the upper end of the range.

These values provide a basis for facilitating discussion and agreement between the supplier and the purchaser and should ensure that in most cases gross deficiencies or unrealistic requirements are avoided.

Caution should be exercised when applying the values given in Table B.1 as there may be specific features associated with a particular machine which would require the use of different values.

**Table B.1 — Range of typical values for the zone A/B, B/C and C/D boundaries**

Range of typical zone boundary values r.m.s. vibration velocity mm/s				
0,28				0,28
0,45				0,45
0,71				0,71
1,12	Zone boundary A/B 0,71 to 4,5			1,12
1,8				1,8
2,8				2,8
4,5		Zone boundary B/C 1,8 to 9,3		4,5
7,1				7,1
9,3				9,3
11,2			Zone boundary C/D 4,5 to 14,7	11,2
14,7				14,7
18				18
28				28
45				45
<p>NOTE 1 This table only applies to machines for which specific parts of ISO 10816 have not been developed and for which there is no past satisfactory experience available.</p> <p>NOTE 2 Acceptance criteria should be subject to agreement between the supplier and the purchaser of the machine.</p> <p>NOTE 3 The values selected should take into account the measurement position and the support flexibility/resilience.</p> <p>NOTE 4 Small machines (e.g. electric motors with power up to 15 kW) tend to lie at the lower end of the range and larger machines (e.g. prime movers with flexible supports in the direction of measurement) tend to lie at the upper end of the range.</p>				

**A1**

## Annex C (informative) General guidelines for specification of criteria

The velocity criteria shown in Figure 6 can be represented by the following general equation:

$$v_{r.m.s.} = v_A \cdot G \cdot (f_z/f_x)^k (f_y/f_w)^m \quad \dots (C.1)$$

where

$v_{r.m.s.}$  is the allowable r.m.s. velocity, in millimetres per second;

$v_A$  is the constant r.m.s. velocity, in millimetres per second, which applies between  $f_x$  and  $f_y$  for zone A;

$G$  is a factor which defines the zone boundaries (e.g. the limit of zone A could be obtained by setting  $G = 1$ , the limit of zone B by setting  $G = 2,56$  and the limit of zone C by setting  $G = 6,4$ ); this factor may be a function of the machine speed or any other relevant machine operating quantity (e.g. load, pressure, flow);

$f_x$  and  $f_y$  are the defined frequencies, in hertz, between which it is assumed that a constant velocity criterion applies (see 5.3.2);

$$f_w = f_y \text{ for } f \leq f_y$$

$$f_w = f \text{ for } f > f_y$$

$$f_z = f \text{ for } f < f_x$$

$$f_z = f_x \text{ for } f \geq f_x$$

$f$  is the frequency, in hertz, for which  $v_{r.m.s.}$  is defined;

$k$  and  $m$  are defined constants for a given machine type.

For special groups of machines, single values of r.m.s. velocity can be specified instead of curves of the type shown in Figure 6.

NOTE 3 The frequencies  $f_u$  and  $f_l$  which are shown in Figure 6 are the upper and lower frequency limits for broad-band measurements.

## Annex D (informative) Vector analysis of change in vibration

### Introduction

Evaluation criteria are defined in terms of the normal steady-running value of broad-band vibration and any changes that may occur in the magnitude of these steady values. The latter criterion has limitations because some changes may only be identified by vector analysis of the individual frequency components. The development of this technique for other than synchronous vibration components is still in its infancy and criteria cannot be defined in this part of ISO 10816 at present.

### D.1 General

The broad-band steady vibration signal measured on a machine 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 not necessarily reflected to the same degree in the broad-band vibration and, hence, the criterion based on changes of broad-band vibration magnitude only may require supplementary phase measurements.

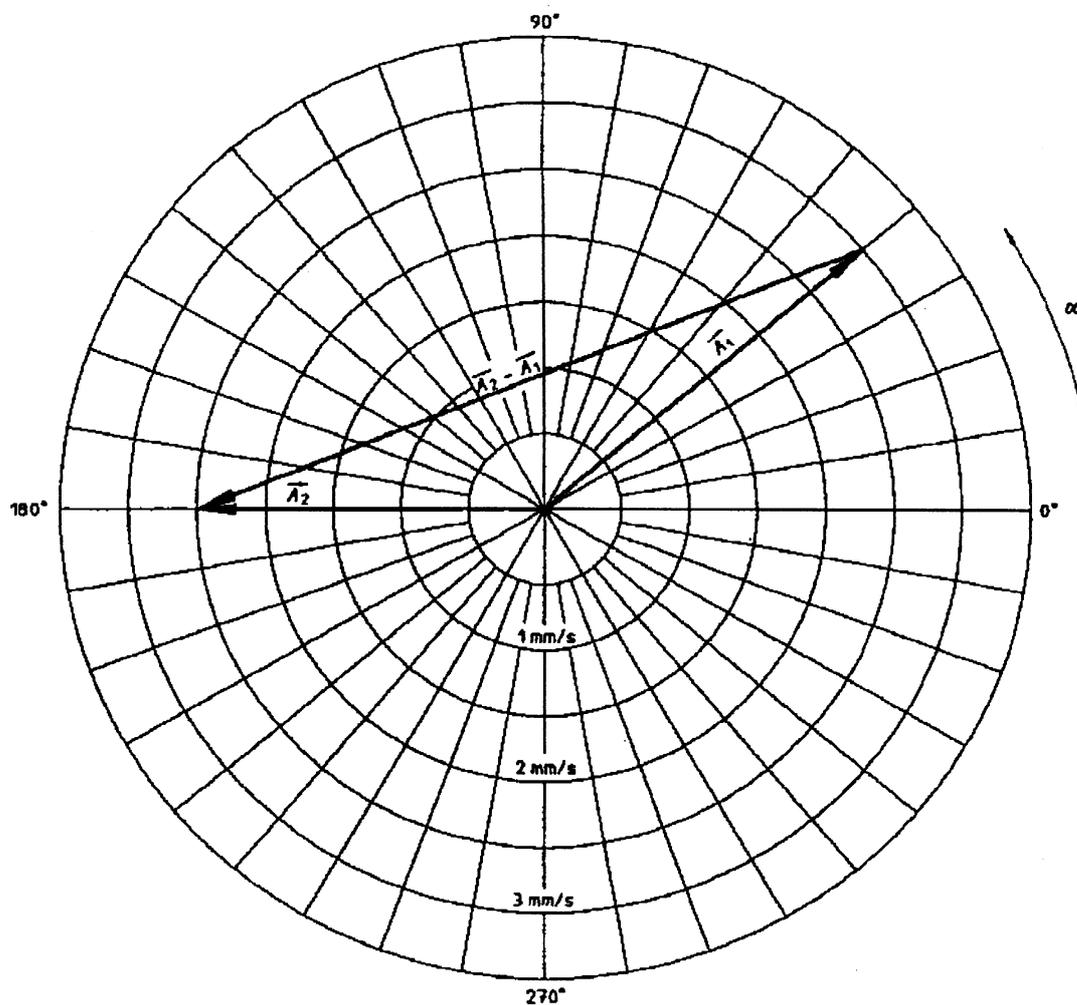
### D.2 Importance of vector changes

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

The vector  $\vec{A}_1$  describes the initial steady-state vibration condition; i.e. in this condition the magnitude of vibration is 3 mm/s (r.m.s.) with a phase angle of  $40^\circ$ . The vector  $\vec{A}_2$  describes the steady-state vibration condition after some change has occurred to the machine; i.e. the magnitude of the vibration is now 2,5 mm/s (r.m.s.) with a phase angle of  $180^\circ$ . Hence, although the vibration magnitude has decreased by 0,5 mm/s (r.m.s.) from 3 mm/s to 2,5 mm/s, the true change of vibration is represented by the vector  $\vec{A}_2 - \vec{A}_1$ , which has a magnitude of 5,2 mm/s (r.m.s.). This is over ten times that indicated by comparing the vibration magnitude alone.

### D.3 Monitoring vector changes

The example given in D.2 clearly illustrates the importance of identifying the vector change in a vibration signal. However, it is necessary to appreciate that, in general, the broad-band 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 10816, which is aimed primarily at normal operational monitoring of broad-band vibration by non-vibration specialists.



Initial steady-state vector	$ \vec{A}_1  = 3 \text{ mm/s (r.m.s.)}, \alpha = 40^\circ$
Steady-state vector after change	$ \vec{A}_2  = 2,5 \text{ mm/s (r.m.s.)}, \alpha = 180^\circ$
Change in vibration magnitude	$ \vec{A}_2  -  \vec{A}_1  = -0,5 \text{ mm/s (r.m.s.)}$
Vector of change	$ \vec{A}_2 - \vec{A}_1  = 5,2 \text{ mm/s (r.m.s.)}$

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

## Annex E (informative)

### Specialist measurement and analysis techniques for detection of problems in rolling-element bearings

#### Introduction

The use of a simple broad-band measuring technique on the raw acceleration data from a rolling-element bearing housing, as described in the main text of this part of ISO 10816, often provides sufficient information to give guidance on the running conditions of that particular bearing, it is realized that this simple technique is not successful in all circumstances. In particular, errors in assessment may arise if there are significant resonance effects in the bearings or its housing within the measurement frequency range, or if significant vibration signals are transmitted to the bearing from other sources such as gear-meshing vibration.

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

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

Clauses **E.1** to **E.4** briefly mention some of the available measuring equipment and analysis techniques which have been shown to have had some success in selected applications. However, insufficient information is available on suitable evaluation criteria values to permit any of the techniques to be incorporated in International Standards at this stage.

#### E.1 Raw data analysis (overall vibration measurements)

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

These alternatives are:

- a) measurement of peak acceleration values;
- b) measurement of the peak-to-r.m.s. ratio (crest factor) of the acceleration;
- c) computation of the product of r.m.s. and peak acceleration measurements.

#### E.2 Frequency analysis

The individual frequency components of a complex vibration signal can be identified with a variety of filtering arrangements or by spectrum analysis. If sufficient data are available about the particular bearing, its characteristic frequencies for a variety of defects can be calculated and compared with the frequency components of the vibration signal. This, therefore, can give not only recognition that a bearing is giving concern, but can also identify the nature of the defect.

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

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

#### E.3 Shock-pulse techniques

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

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

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

#### **E.4 Alternative techniques**

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

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See national foreword.

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